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COMMENT

# A trustworthy reference in a familiar package

As the world spends more and more time online staring at a screen, there are many readers who still believe in the value of the printed page. We publish and distribute content digitally on a daily basis, but still hear of a demand for ink and paper. For those readers, we offer the 2022 **COMPRESSOR**TECH² **SOURCING GUIDE**.

This is a popular product, one which comes from an ongoing demand for updated relevant information in one place. The publication has become the industry go-to guide for all players in the gas compression industry. Think of it as a who's who of the gas compression world.

The **COMPRESSOR**TECH² **SOURCING GUIDE** is still widely used as a reference and teaching tool at colleges, tech schools and educational conferences that offer compression training programs. (To get a physical copy of the guide for education or training programs, email us at CTSG.info@KHL.com.)

Print media has survived into the digital age and by some standards, it is thriving. Publishers moved aggressively into the digital space a couple of decades ago, but print continues to have a place.

By its very nature, print forces readers to engage. They flip pages, thumb through an issue or carry it around as a reference. The ongoing longevity of the printed word is akin to the revival of vinyl records.

Some readers absorb complex information more readily when it is in print. For many readers, the tangibility of print forces them to engage more actively and they recall content more vividly. They recall the content, its location on the page and its location in the overall text.

For some, the heft of a given publication gives its contents additional gravitas that the ephemeral word on a screen does not carry. Print is a means of communication that does not require a separate device and internet access. No booting up, no log-ins and no annoying cookies or pop-ups.

To be sure, digital has its benefits. It's easy to do a quick search for a specific piece of information. It can be updated more easily. It can be sent to another location with a few strokes on the keyboard.

And happily, the **COMPRESSOR**TECH² **SOURCING GUIDE** is available in that format as well. It's your choice as to how you wish to access the information.

Whatever format you choose, we hope you enjoy the 2022 edition of the **COMPRESSOR**TECH² **SOURCING GUIDE**. **CT2**

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*Compressors, Gas*
*Compressors, Oil-Free*
*Compressors, Oil-Injected*
*Compressors, Piston*
*Compressors, Reciprocating*
*Compressors, Stationary*

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- For product listing see **KoHo (Köhler & Hörter GmbH)**

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**Leobersdorfer Maschinenfabrik GmbH ................................................................... 112, 113**
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*Compressors, Air*
*Compressors, Gas*
*Compressors, Oil-Free*
*Compressors, Oil-Injected*

*Compressors, Piston*
*Compressors, Reciprocating*
*Compressors, Rotary Screw*
*Compressors, Screw*
*Compressors, Single Screw*
*Compressors, Skid-Mounted*
*Compressors, Stationary*
*Services, Engineering*
*Services, Gas Turbine Overhaul & Repair*
*Steam Turbines*

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*Compressor Sets, Electrically Engine-Driven*
*Compressor Sets, Gas Turbine-Driven*
*Compressors, Air*
*Compressors, Axial*
*Compressors, Centrifugal*
*Compressors, Gas*
*Compressors, Integral*
*Compressors, Oil-Free*
*Compressors, Rotary Screw*
*Compressors, Screw*
*Compressors, Skid-Mounted*
*Compressors, Stationary*
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*Drives, Compressor*
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*Expanders*
*Monitors, Compressor Systems*
*Monitors, Engine System*
*Packages, Engine Compressor*
*Packages, Motor Compressor*
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*Protective Controls*
*Reactor Systems*
*Rotors, Turbomachinery*
*Service Systems & Training, Gas Turbines*
*Services & Training*
*Services, Compressors Overhaul & Repair*

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*Compressors, Reconditioned*
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*Monitors, Compressor Systems*
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*Monitors, Pressure*
*Monitors, Temperature*
*Monitors, Vibration*
*Pistons, Reconditioning*
*Project Management*
*Rings, Packing*
*Rings, Piston*
*Service Tools & Equipment*
*Services & Training*
*Services, Compressors Overhaul & Repair*
*Services, Diagnostics*
*Services, Engineering*
*Services, Failure-Analysis*
*Services, Field Pulsation, Vibration-Analysis*
*Services, Laser Measurements*
*Services, Shutdown*
*Valves, Check*
*Valves, Compressor*
*Valves, Overhaul*

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*Cylinders, Reconditioning*
*Monitors, Compressor Systems*
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*Services, Field Pulsation, Vibration-Analysis*
*Services, Laser Measurements*
*Services, Shutdown*
*Valves, Check*
*Valves, Compressor*
*Valves, Overhaul*

**NEA Al-Hotta Industrial Est. .............92, 93**
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*Compressor Sets, Electrically Engine-Driven*

*Packages, Engineering & Design*
*Packages, Piping, Structural Analysis*
*Packing Assemblies*
*Pistons, Reconditioning*
*Project Management*
*Rings, Packing*
*Rings, Piston*
*Service Tools & Equipment*
*Services & Training*
*Services, Compressors Overhaul & Repair*
*Services, Diagnostics*
*Services, Engineering*
*Services, Failure-Analysis*
*Services, Field Pulsation, Vibration-Analysis*
*Services, Laser Measurements*
*Services, Shutdown*
*Valves, Check*
*Valves, Compressor*
*Valves, Overhaul*

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*Compressors, Air*
*Compressors, Gas*
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*Compressors, Oil-Injected*
*Compressors, Piston*
*Compressors, Reciprocating*
*Compressors, Skid-Mounted*
*Rings, Packing*
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*Services, Compressors Overhaul & Repair*

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*Services, Engineering*
*Service provider*
*Consultant*
*Projecting of Biogas Plants*
*Biogas Treatment Facilities*
*Biogas Production / Biomethane Feeding into the Grid*

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*Natural Gas and Methanol Reformers*

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*Compressor Sets, Gas Turbine-Driven*
*Compressor Sets, Natural Gas Engine-Driven*
*Compressors, Air*
*Compressors, Gas*
*Compressors, Oil-Free*
*Compressors, Oil-Injected*
*Compressors, Piston*
*Compressors, Reciprocating*
*Compressors, Reconditioned*
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*Services & Training*
*Services, Compressors Overhaul & Repair*
*Services, Engineering*
*Services, Failure-Analysis*

Solar® Turbines
*A Caterpillar Company*

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*Compressors, Gas*
*Engines, Gas Turbine*
*Packages, Engine Compressor*
*Packages, Foundation, Platform Deck, FPSO Module Design*

## T

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Termomeccanica Group

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www.tmic.termomeccanica.com
*Compressors, Air*
*Compressors, Gas*
*Compressors, Oil-Injected*
*Compressors, Screw*

# COMPRESSOR HORSEPOWER SELECTION CHART

(Brake Horsepower Per Million Cu. Ft.)

| SUCTION PRESSURE | DISCHARGE PRESSURE (PSIG) | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 25 | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 550 | 600 | 650 | 700 | 750 | 800 | 850 | 900 | 950 | 1000 | 1050 | 1100 | 1150 | 1200 | |
| 0 | 65 | 99 | 128 | 144 | 156 | 168 | 178 | 187 | 203 | 218 | 233 | 233 | 241 | 248 | 254 | 260 | 266 | 272 | 277 | 282 | 286 | 291 | 295 | 299 | 303 | 307 | 311 | 315 | 3 STAGE |
| 10 | 35 | 63 | 85 | 104 | 121 | 131 | 140 | 149 | 163 | 175 | 186 | 196 | 205 | 214 | 223 | 231 | 228 | 233 | 237 | 242 | 245 | 250 | 253 | 257 | 260 | 264 | 267 | 270 | |
| 20 | | 43 | 62 | 78 | 92 | 106 | 118 | 126 | 139 | 151 | 160 | 170 | 178 | 186 | 193 | 199 | 206 | 212 | 218 | 225 | 231 | 226 | 229 | 232 | 236 | 239 | 242 | 245 | |
| 30 | | 29 | 47 | 62 | 74 | 85 | 96 | 107 | 123 | 133 | 143 | 152 | 159 | 167 | 173 | 179 | 185 | 191 | 196 | 201 | 206 | 211 | 216 | 221 | 226 | 230 | 224 | 227 | |
| 40 | | | 36 | 50 | 61 | 72 | 81 | 90 | 107 | 121 | 130 | 138 | 145 | 152 | 158 | 164 | 170 | 175 | 180 | 185 | 190 | 194 | 198 | 202 | 206 | 210 | 214 | 218 | 2 STAGE |
| 50 | | | 26 | 41 | 52 | 61 | 70 | 78 | 93 | 106 | 119 | 127 | 134 | 141 | 147 | 153 | 158 | 163 | 168 | 173 | 177 | 181 | 185 | 189 | 193 | 196 | 200 | 203 | |
| 60 | | | | 32 | 44 | 53 | 61 | 69 | 83 | 95 | 108 | 118 | 125 | 131 | 137 | 143 | 148 | 153 | 158 | 162 | 166 | 170 | 174 | 178 | 182 | 185 | 188 | 192 | |
| 70 | | | | 25 | 37 | 46 | 54 | 61 | 74 | 86 | 97 | 109 | 117 | 123 | 129 | 135 | 140 | 145 | 149 | 153 | 157 | 161 | 165 | 169 | 172 | 176 | 179 | 182 | |
| 80 | | | | | 30 | 40 | 47 | 54 | 67 | 78 | 89 | 98 | 109 | 117 | 122 | 127 | 132 | 137 | 142 | 146 | 150 | 153 | 157 | 161 | 164 | 167 | 171 | 174 | |
| 90 | | | | | 24 | 34 | 42 | 49 | 61 | 72 | 81 | 91 | 100 | 109 | 116 | 121 | 126 | 131 | 135 | 139 | 143 | 147 | 150 | 154 | 157 | 160 | 163 | 166 | |
| 100 | | | | | | 28 | 37 | 44 | 55 | 66 | 75 | 84 | 92 | 100 | 109 | 116 | 120 | 125 | 129 | 133 | 137 | 141 | 144 | 148 | 151 | 154 | 157 | 160 | |
| 125 | | | | | | | 25 | 32 | 44 | 54 | 63 | 71 | 78 | 85 | 92 | 99 | 106 | 113 | 117 | 121 | 124 | 128 | 131 | 134 | 137 | 140 | 143 | 146 | |
| 150 | | | | | | | | 22 | 35 | 45 | 53 | 60 | 67 | 74 | 80 | 86 | 92 | 98 | 103 | 110 | 114 | 118 | 121 | 124 | 127 | 130 | 133 | 135 | |
| 175 | | | | | | | | | 27 | 37 | 45 | 52 | 57 | 60 | 71 | 76 | 82 | 87 | 92 | 97 | 102 | 107 | 112 | 115 | 118 | 121 | 123 | 126 | |
| 200 | | | | | | | | | | 30 | 38 | 45 | 52 | 58 | 63 | 68 | 73 | 78 | 83 | 88 | 92 | 96 | 101 | 105 | 110 | 113 | 116 | 119 | |
| 250 | | | | | | | | | | | 26 | 33 | 40 | 46 | 51 | 56 | 60 | 65 | 69 | 73 | 77 | 81 | 85 | 88 | 92 | 95 | 99 | 102 | 1 STAGE |
| 300 | | | | | | | | | | | | 23 | 30 | 36 | 41 | 46 | 50 | 54 | 58 | 62 | 66 | 69 | 73 | 76 | 79 | 83 | 86 | 89 | |
| 350 | | | | | | | | | | | | | 21 | 27 | 33 | 38 | 42 | 46 | 50 | 53 | 57 | 60 | 63 | 67 | 70 | 73 | 75 | 78 | |
| 400 | | | | | | | | | | | | | | | 25 | 30 | 35 | 39 | 43 | 46 | 50 | 53 | 56 | 59 | 60 | 64 | 67 | 70 | |
| 450 | | | | | | | | | | | | | | | | 23 | 28 | 32 | 36 | 40 | 43 | 46 | 49 | 52 | 55 | 58 | 60 | 63 | |
| 500 | | | | | | | | | | | | | | | | | 22 | 26 | 30 | 34 | 38 | 41 | 44 | 46 | 49 | 52 | 54 | 57 | |
| 550 | | | | | | | | | | | | | | | | | | 20 | 25 | 20 | 32 | 36 | 39 | 41 | 44 | 46 | 49 | 51 | |
| 600 | | | | | | | | | | | | | | | | | | | | 23 | 27 | 30 | 34 | 37 | 39 | 42 | 44 | 46 | |
| 650 | | | | | | | | | | | | | | | | | | | | | 22 | 26 | 29 | 32 | 35 | 38 | 40 | 42 | |
| 700 | | | | | | | | | | | | | | | | | | | | | | 22 | 25 | 28 | 30 | 33 | 36 | 38 | |
| 750 | | | | | | | | | | | | | | | | | | | | | | | 20 | 24 | 27 | 29 | 32 | 34 | |

**NOTE: 1** MMSCFD MEASURED 14.7 AND 60°F NOT CORRECTED FOR COMPRESSIBILITY **2** "N"=1.26 **3** SUCTION TEMPERATURE 100°F **4** NATURAL GAS

## A



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## R

# CONVERSION FACTORS
## SI - METRIC/DECIMAL SYSTEM

## ABBREVIATIONS

| | |
|---|---|
| abs | absolute |
| ata | atmosphere absolute |
| Btu | British thermal unit |
| Btu/hr | British thermal unit/ hour |
| °C | Celsius |
| cfm | cubic foot/minute |
| cm | centimeter |
| $cm^2$ | square centimeter |
| $cm^3$ | cubic centimeter |
| cu.ft. | cubic foot |
| °F | Fahrenheit |
| ft/sec | foot/second |
| ft-lb | foot-pound |
| gal | gallon |
| hp | horsepower |
| in | inch |
| in. Hg | inch mercury |
| in. $H_2O$ | inch water |
| kcal | kilocalorie |
| kg | kilogram |
| kJ | kilojoule |
| kPa | kilopascal |
| kW | kilowatt |
| L | liter |
| m | meter |
| mm | millimeter |
| $m^2$ | square meter |
| $m^3$ | cubic meter |
| $m^3$/min | cubic meter/minute |
| mph | mile per hour |
| N | Newton |
| $N/m^2$ | Pascal |
| $Nm^3$/hr | normal* cubic meter/hour |
| psi | pound/square inch |
| psia | pound/square inch absolute |
| psig | pound/square inch gage |
| scf | standard* cubic foot |
| scfm | standard* cubic foot/minute |
| sq | square |

* "Normal" = 0°C and 1.01325 x $10^5$ Pascals
* "Standard" = 59°F and 14.73 psia

## MILLIMETERS (mm) TO INCHES (in)
(1 millimeter = 0.03937 inch)

| mm | in | mm | in | mm | in | mm | in | mm | in |
|---|---|---|---|---|---|---|---|---|---|
| **1** | 0.039 | **21** | 0.827 | **41** | 1.614 | **61** | 2.402 | **81** | 3.189 |
| **2** | 0.079 | **22** | 0.866 | **42** | 1.654 | **62** | 2.441 | **82** | 3.228 |
| **3** | 0.118 | **23** | 0.906 | **43** | 1.693 | **63** | 2.480 | **83** | 3.268 |
| **4** | 0.157 | **24** | 0.945 | **44** | 1.732 | **64** | 2.520 | **84** | 3.307 |
| **5** | 0.197 | **25** | 0.984 | **45** | 1.772 | **65** | 2.559 | **85** | 3.346 |
| **6** | 0.236 | **26** | 1.024 | **46** | 1.811 | **66** | 2.598 | **86** | 3.386 |
| **7** | 0.276 | **27** | 1.063 | **47** | 1.850 | **67** | 2.638 | **87** | 3.425 |
| **8** | 0.315 | **28** | 1.102 | **48** | 1.890 | **68** | 2.677 | **88** | 3.465 |
| **9** | 0.354 | **29** | 1.142 | **49** | 1.929 | **69** | 2.717 | **89** | 3.504 |
| **10** | 0.394 | **30** | 1.181 | **50** | 1.968 | **70** | 2.756 | **90** | 3.543 |
| **11** | 0.433 | **31** | 1.220 | **51** | 2.008 | **71** | 2.795 | **91** | 3.583 |
| **12** | 0.472 | **32** | 1.260 | **52** | 2.047 | **72** | 2.835 | **92** | 3.622 |
| **13** | 0.512 | **33** | 1.299 | **53** | 2.087 | **73** | 2.874 | **93** | 3.661 |
| **14** | 0.551 | **34** | 1.339 | **54** | 2.126 | **74** | 2.913 | **94** | 3.701 |
| **15** | 0.591 | **35** | 1.378 | **55** | 2.165 | **75** | 2.953 | **95** | 3.740 |
| **16** | 0.630 | **36** | 1.417 | **56** | 2.205 | **76** | 2.992 | **96** | 3.779 |
| **17** | 0.669 | **37** | 1.457 | **57** | 2.244 | **77** | 3.032 | **97** | 3.819 |
| **18** | 0.709 | **38** | 1.496 | **58** | 2.283 | **78** | 3.071 | **98** | 3.858 |
| **19** | 0.748 | **39** | 1.535 | **59** | 2.323 | **79** | 3.110 | **99** | 3.898 |
| **20** | 0.787 | **40** | 1.575 | **60** | 2.362 | **80** | 3.150 | **100** | 3.937 |

## KILOGRAMS (kg) TO POUNDS (lb)
(1 kilogram = 2.20462 pounds)

| kg | lb | kg | lb | kg | lb | kg | lb | kg | lb |
|---|---|---|---|---|---|---|---|---|---|
| **1** | 2.204 | **21** | 46.297 | **41** | 90.390 | **61** | 134.482 | **81** | 178.574 |
| **2** | 4.409 | **22** | 48.502 | **42** | 92.594 | **62** | 136.687 | **82** | 180.779 |
| **3** | 6.614 | **23** | 50.706 | **43** | 94.799 | **63** | 138.891 | **83** | 182.984 |
| **4** | 8.819 | **24** | 52.911 | **44** | 97.003 | **64** | 141.096 | **84** | 185.188 |
| **5** | 11.023 | **25** | 55.116 | **45** | 99.208 | **65** | 143.300 | **85** | 187.393 |
| **6** | 13.228 | **26** | 57.320 | **46** | 101.413 | **66** | 145.505 | **86** | 189.598 |
| **7** | 15.432 | **27** | 59.525 | **47** | 103.617 | **67** | 147.710 | **87** | 191.802 |
| **8** | 17.637 | **28** | 61.729 | **48** | 105.822 | **68** | 149.914 | **88** | 194.007 |
| **9** | 19.843 | **29** | 63.934 | **49** | 108.026 | **69** | 152.119 | **89** | 196.211 |
| **10** | 22.046 | **30** | 66.139 | **50** | 110.231 | **70** | 154.324 | **90** | 198.416 |
| **11** | 24.251 | **31** | 66.343 | **51** | 112.436 | **71** | 156.528 | **91** | 200.621 |
| **12** | 26.455 | **32** | 70.548 | **52** | 114.640 | **72** | 158.733 | **92** | 202.825 |
| **13** | 28.660 | **33** | 72.753 | **53** | 116.845 | **73** | 160.937 | **93** | 205.030 |
| **14** | 30.865 | **34** | 74.957 | **54** | 119.050 | **74** | 163.142 | **94** | 207.235 |
| **15** | 33.069 | **35** | 77.162 | **55** | 121.254 | **75** | 165.347 | **95** | 209.439 |
| **16** | 35.274 | **36** | 79.366 | **56** | 123.459 | **76** | 167.551 | **96** | 211.644 |
| **17** | 37.479 | **37** | 81.571 | **57** | 125.663 | **77** | 169.756 | **97** | 213.848 |
| **18** | 39.683 | **38** | 83.776 | **58** | 127.868 | **78** | 171.961 | **98** | 216.053 |
| **19** | 41.888 | **39** | 85.980 | **59** | 130.073 | **79** | 174.165 | **99** | 218.258 |
| **20** | 44.093 | **40** | 88.185 | **60** | 132.277 | **80** | 176.370 | **100** | 220.462 |

## CONVERSION FACTORS

| TO CONVERT FROM ENGLISH | TO S.I. METRIC | MULTIPLY BY | TO OLD METRIC | MULTIPLY BY |
|---|---|---|---|---|
| sq. in. | $mm^2$ | **645.16** | $cm^2$ | **6.4516** |
| sq. ft. | $m^2$ | **0.0929** | $m^2$ | **0.0929** |
| lb/cu.ft. | $kg/m^3$ | **16.0185** | $kg/m^3$ | **16.0185** |
| $lb_f$ | N | **4.4482** | N | **4.4482** |
| $lb_f$/ft | N/m | **14.5939** | N/m | **14.5939** |
| Btu | kJ | **1.0551** | kcal | **0.252** |
| Btu/hr | W | **0.2931** | kcal/hr | **0.252** |
| Btu/scf | $kJ/mm^3$ | **37.2590** | $kcal/nm^3$ | **0.1565** |
| in | mm | **25.400** | cm | **2.540** |
| ft | m | **0.3048** | m | **0.3048** |
| yd | m | **0.914** | m | **0.914** |
| lb | kg | **0.4536** | kg | **0.4536** |
| hp | kW | **0.7457** | kW | **0.7457** |
| psi | kPa | **6.8948** | $kg/cm^2$ | **0.070** |
| psia | kPa abs | **6.8948** | bars abs | **0.0716** |
| psig | kPa gage | **6.8948** | ata | **0.070** |
| in. Hg | kPa | **3.3769** | cm Hg | **2.540** |
| in. $H_2O$ | kPa | **0.2488** | cm $H_2O$ | **2.540** |
| °F | °C = | **(°F -32) 5/9** | °C = | **(°F -32) 5/9** |
| °F (Interval) | °C (Interval) | **5/9** | °C (Interval) | **5/9** |
| ft-lb | N • m | **1.3558** | N • m | **1.3558** |
| mph | km/hr | **1.6093** | km/hr | **1.6093** |
| ft/sec | m/sec | **0.3048** | m/sec | **0.3048** |
| cu. ft. | $m^3$ | **0.0283** | $m^3$ | **0.0283** |
| gas (US) | L | **3.7854** | L | **3.7854** |
| cfm | $m^3$/min | **0.0283** | $m^3$/min | **0.0283** |
| scfm | $nm^3$/min | **0.0268** | $nm^3$/hr | **1.61** |

| TO CONVERT FROM OLD METRIC | TO S.I. METRIC | MULTIPLY BY |
|---|---|---|
| $cm^2$ | $mm^2$ | **100.** |
| kcal | kJ | **4.1868** |
| kcal/hr | W | **1.16279** |
| cm | mm | **10.** |
| $kg/cm^2$ | kPa | **98.0665** |
| bars | kPa | **100.** |
| atm | kPa | **101.325** |
| cm Hg | kPa | **1.3332** |
| cm $H_2O$ | kPa | **9.8064** |
| $nm^3$/hr | $nm^3$/min | **0.0176** |

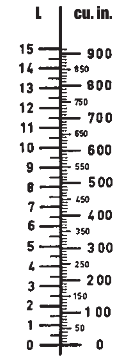
## TEMPERATURE CONVERSION TABLES*

| | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **0 TO 100** | | | 2.78 | **37** | 98.6 | 23.9 | **75** | 167.0 | 93 | **200** | 392 | 299 | **570** | 1058 | 510 | **950** | 1742 |
| -17.8 | **0** | 32 | 3.33 | **38** | 100.4 | 24.4 | **76** | 168.8 | 99 | **210** | 410 | 304 | **580** | 1076 | 516 | **960** | 1760 |
| -17.2 | **1** | 33.8 | 3.89 | **39** | 102.2 | 25.0 | **77** | 170.6 | 100 | **212** | 413 | 310 | **590** | 1094 | 521 | **970** | 1778 |
| -16.7 | **2** | 35.6 | 4.44 | **40** | 104.0 | 25.6 | **78** | 172.4 | 104 | **220** | 428 | 316 | **600** | 1112 | 527 | **980** | 1796 |
| -16.1 | **3** | 37.4 | 5.00 | **41** | 105.8 | 26.1 | **79** | 174.2 | 110 | **230** | 446 | 321 | **610** | 1130 | 532 | **990** | 1814 |
| -15.6 | **4** | 39.2 | 5.56 | **42** | 107.6 | 26.7 | **80** | 176.0 | 116 | **240** | 464 | 327 | **620** | 1148 | 538 | **1000** | 1832 |
| -15.0 | **5** | 41.0 | 6.11 | **43** | 109.4 | 27.2 | **81** | 177.8 | 121 | **250** | 482 | 332 | **630** | 1166 | | | |
| -14.4 | **6** | 42.8 | 6.67 | **44** | 111.2 | 27.8 | **82** | 179.6 | 127 | **260** | 500 | 338 | **640** | 1184 | **1000 TO 1630** | | |
| -13.9 | **7** | 44.9 | 7.22 | **45** | 113.0 | 28.3 | **83** | 181.4 | 132 | **270** | 518 | 343 | **650** | 1202 | 538 | **1000** | 1832 |
| -13.3 | **8** | 46.4 | 7.78 | **46** | 114.8 | 28.9 | **84** | 183.2 | 138 | **280** | 536 | 349 | **660** | 1220 | 543 | **1010** | 1850 |
| -12.8 | **9** | 48.2 | 8.33 | **47** | 116.6 | 29.4 | **85** | 185.0 | 143 | **290** | 554 | 354 | **670** | 1238 | 549 | **1020** | 1868 |
| -12.1 | **10** | 50.0 | 8.89 | **48** | 118.4 | 30.0 | **86** | 186.8 | 149 | **300** | 572 | 360 | **680** | 1256 | 554 | **1030** | 1886 |
| -11.7 | **11** | 51.8 | 9.44 | **49** | 120.0 | 30.6 | **87** | 188.6 | 154 | **310** | 590 | 366 | **690** | 1274 | 560 | **1040** | 1904 |
| -11.1 | **12** | 53.6 | 10.0 | **50** | 122.0 | 31.1 | **88** | 190.4 | 160 | **320** | 608 | 371 | **700** | 1292 | 566 | **1050** | 1922 |
| -10.6 | **13** | 55.4 | 10.6 | **51** | 123.8 | 31.7 | **89** | 192.2 | 166 | **330** | 626 | 377 | **710** | 1310 | 571 | **1060** | 1940 |
| -10.0 | **14** | 57.2 | 11.1 | **52** | 125.6 | 32.2 | **90** | 194.0 | 171 | **340** | 644 | 382 | **720** | 1328 | 577 | **1070** | 1958 |
| -9.44 | **15** | 59.0 | 11.7 | **53** | 127.4 | 32.8 | **91** | 195.8 | 177 | **350** | 662 | 388 | **730** | 1346 | 582 | **1080** | 1976 |
| -8.89 | **16** | 60.8 | 12.2 | **54** | 129.2 | 33.3 | **92** | 197.6 | 182 | **360** | 680 | 393 | **740** | 1364 | 588 | **1090** | 1994 |
| -8.33 | **17** | 62.6 | 12.8 | **55** | 131.0 | 33.9 | **93** | 199.4 | 188 | **370** | 698 | 399 | **750** | 1382 | 593 | **1100** | 2012 |
| -7.78 | **18** | 64.4 | 13.3 | **56** | 132.8 | 34.4 | **94** | 201.2 | 193 | **380** | 716 | 404 | **760** | 1400 | 599 | **1110** | 2030 |
| -7.22 | **19** | 66.2 | 13.9 | **57** | 134.6 | 35.0 | **95** | 203.0 | 199 | **390** | 734 | 410 | **770** | 1418 | 604 | **1120** | 2048 |
| -6.67 | **20** | 68.0 | 14.4 | **58** | 136.4 | 35.6 | **96** | 204.8 | 204 | **400** | 752 | 416 | **780** | 1436 | 610 | **1130** | 2066 |
| -6.11 | **21** | 69.8 | 15.0 | **59** | 138.2 | 36.1 | **97** | 206.6 | 210 | **410** | 770 | 421 | **790** | 1454 | 816 | **1500** | 2732 |
| -5.56 | **22** | 71.6 | 15.6 | **60** | 140.0 | 36.7 | **98** | 208.4 | 216 | **420** | 788 | 427 | **800** | 1472 | 821 | **1510** | 2750 |
| -5.00 | **23** | 73.4 | 16.1 | **61** | 141.8 | 37.2 | **99** | 210.2 | 221 | **430** | 806 | 432 | **810** | 1490 | 827 | **1520** | 2768 |
| -4.44 | **24** | 75.2 | 16.7 | **62** | 143.6 | 37.8 | **100** | 212.0 | 227 | **440** | 824 | 438 | **820** | 1508 | 832 | **1530** | 2786 |
| -3.89 | **25** | 77.0 | 17.2 | **63** | 145.4 | | | | 232 | **450** | 842 | 443 | **830** | 1526 | 838 | **1540** | 2804 |
| -3.33 | **26** | 78.8 | 17.8 | **64** | 147.2 | **100 TO 1000** | | | 238 | **460** | 860 | 449 | **840** | 1544 | 843 | **1550** | 2822 |
| -2.78 | **27** | 80.6 | 18.3 | **65** | 149.0 | 38 | **100** | 212 | 243 | **470** | 878 | 454 | **850** | 1562 | 849 | **1560** | 2840 |
| -2.22 | **28** | 82.4 | 18.9 | **66** | 150.8 | 43 | **110** | 230 | 249 | **480** | 896 | 460 | **860** | 1580 | 854 | **1570** | 2858 |
| -1.67 | **29** | 84.2 | 19.4 | **67** | 152.6 | 49 | **120** | 248 | 254 | **490** | 914 | 466 | **870** | 1598 | 860 | **1580** | 2876 |
| -1.11 | **30** | 86.0 | 20.0 | **68** | 154.4 | 54 | **130** | 266 | 260 | **500** | 932 | 471 | **880** | 1616 | 866 | **1590** | 2894 |
| -0.56 | **31** | 87.8 | 20.6 | **69** | 156.2 | 60 | **140** | 284 | 266 | **510** | 950 | 477 | **890** | 1634 | 871 | **1600** | 2912 |
| 0 | **32** | 89.6 | 21.1 | **70** | 158.0 | 66 | **150** | 302 | 271 | **520** | 968 | 482 | **900** | 1652 | 877 | **1610** | 2930 |
| 0.56 | **33** | 91.4 | 21.7 | **71** | 159.8 | 71 | **160** | 320 | 277 | **530** | 986 | 488 | **910** | 1670 | 882 | **1620** | 2948 |
| 1.11 | **34** | 93.2 | 22.2 | **72** | 161.6 | 77 | **170** | 338 | 282 | **540** | 1004 | 493 | **920** | 1688 | 888 | **1630** | 2966 |
| 1.67 | **35** | 95.0 | 22.8 | **73** | 163.4 | 82 | **180** | 356 | 288 | **550** | 1022 | 499 | **930** | 1706 | | | |
| 2.22 | **36** | 96.8 | 23.3 | **74** | 165.2 | 88 | **190** | 374 | 293 | **560** | 1040 | 504 | **940** | 1724 | | | |

Note: The numbers in **bold** face type refer to the temperature either in degrees Centigrade or Fahrenheit which is desired to convert into the other scale. If converting from Fahrenheit degrees to Centigrade degrees, the equivalent temperatures will be found in the left column; while if converting from degrees Centigrade to degrees Fahrenheit, the answer will be found in the column on the right.

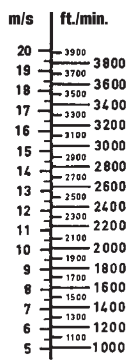
## VOLUME CONVERSION FACTORS

**1 L = 61.02 cu. in.**
**10 cu. in. = 0,164 L**



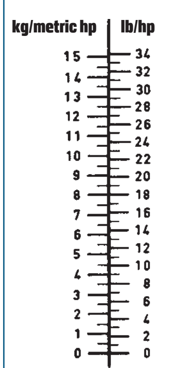

## PISTON SPEED CONVERSION FACTORS

**1 m/s = 196.9 ft./min.**
**100 ft./min. = 0,51 m/s**




## WEIGHT/ HORSEPOWER CONVERSION FACTORS

**1 kg/metric hp = 2.235 lb./hp**
**1 lb/hp = .4474 kg/metric hp**



The system outlined here is the International System of Units (Systeme International d' Unites), for which the abbreviation SI is being used in all languages.

The SI system, which is becoming universally used, is founded on seven base units, these being:

| | | |
|---|---|---|
| Length | meter | m |
| Mass | kilogram | kg |
| Time | second | s |
| Electric current | ampere | A |
| Thermodynamic temperature | Kelvin | K |
| Luminous intensity | candela | cd |
| Amount of substance | mole | mol |

## POWER

The derived SI unit for power is the Watt (W), this being based on the SI unit of work, energy and quantity of heat – the Joule (J). One Watt (1 W) is equal to one Joule per second (1 J/s). One Watt is a very small unit of power, being equivalent to just 0.00134102 horsepower, so for engine ratings the kilowatt (kW) is used, 1 kW being equal to 1.341 hp and 1 hp being the equivalent of 0.7457 kW. The British unit of horsepower is equal to 1.014 metric horsepower (CV, PS, PK, etc.).

1 kW = 1.341 hp = 1.360 metric hp
1 hp = 0.746 kW = 1.014 metric hp
1 metric hp = 0.735 kW = 0.986 hp

## TORQUE

The derived SI unit for torque (or moment of force) is the Newton meter (Nm), this being based on the SI unit of force — the Newton (N) – and the SI unit of length – the meter (m). One Newton (1 N) is equivalent to 0.2248 pound-force (lbf) or 0.10197 kilogram-force (kgf), and one meter is equal to kilogram force (kgf) and one member is equal to 3.28084 feet (ft), so one Newton meter (1 N m) is equal to 0.737562 pound-force (lbf ft). or 0.101972 kilogram-force meter (kgf m).

1 Nm = 0.738 lbf ft = 0.102 kgf m
1 lbf ft = 1.356 Nm = 0.138 kgf m
1 kgf m = 9.807 Nm = 7.233 lbf ft

## PRESSURE AND STRESS

Although it has been decided that the SI derived unit for pressure and stress should be the Pascal (Pa), this is a very small unit, being the same as one Newton per square meter (1 N/m$^2$), which is only 0.000145 lbf/in$^2$ or 0.0000102 kgf/cm$^2$. So many European engine designers favor the bar as the unit of pressure, one bar being 100,000 Pascal (100 kPa), which is the equivalent of 14,504 lbf/in$^2$ or 1.020 kgf/cm$^2$, so being virtually the same as the currently accepted metric equivalent. On the other hand, for engine performance purposes, the millibar seems to be favored to indicate barometric pressure, this unit being one thousandth of a bar. Then again, there is a school that favors the kiloNewton per square meter (kN/m$^2$), this being the same as a kilopascal, and equal to 0.145 lbf/in$^2$ or 0.0102 kgf/cm$^2$.

1 bar = 14.5 lbf/in$^2$ = 1.0197 kgf/cm$^2$
1 lbf/in$^2$ = 0.069 bar
1 kgf/cm$^2$ = 0.98 bar

The American Society of Mechanical Engineers in 1973 published its Performance Test Codes for Reciprocating Internal Combustion engines. Known as PTC 17, this code is intended for tests of all types of reciprocating internal combustion engines for determining power output and fuel consumption. In its Section 2, Description and Definition of Terms, both the FPS and corresponding SI units of meas-urements are given.

## SPECIFIC CONSUMPTION

Fuel consumption measurements will be based on the currently accepted unit, the gram (g), and the Kilowatt Hour (kWh). Also adopted is heat units/power units so that energy consumption of an internal combustion engine referred to net power output, mechanical, is based on low unsaturated heat value of the fuel whether liquid or gaseous type. Thus the SI unit of measurement for net specific energy consumption is expressed: g/kWh.

1 g/kWh = 0.001644 lb/hph =
0.746 g/hph = 0.736 g/metric hph
1 lb/hph = 608.3 g/kWh
1 g/hph = 1.341 k/kWh
1 g/metric hph = 1.36 g/kWh

## HEAT RATE

Heat Rate is a product of Lower Heating Value (LHV) of Fuel (measured in Btu/lb or kJ/g for liquid fuel and Btu/ft$^3$ or kJ/m$^3$ for gas fuel) multiplied times (sfc) specific fuel consumption (measured in lb/hph or g/kWh).

**For Liquid Fuel**

Heat Rate (Btu/hph) = LVH (Btu/lb) X sfc (lb/hph)

**For Gaseous Fuel**

Heat Rate (Btu/hph) = LVH (Btu/ft$^3$) X sfc (ft$^3$/hph)

**To convert these units to SI units:**

Btu/hph X 1.414 = kJ/kWh
Or
Btu/kWh X 1.055 = kJ/kWh

## LUBRICATING-OIL CONSUMPTION

Although the metric liter is not officially an SI unit, its use will continue to be permitted, so measurement of lube-oil consumption will be quoted in liters per hour (liters/h).

1 liter/h = 0.22 Imp gal/h
1 Imp gal/h = 4.546 liters/h

## TEMPERATURES

The SI unit of temperature is Kelvin (K), and the character is used without the degree symbol (°) normally employed with other scales of temperature. A temperature of zero degree Kelvin is equivalent to a temperature of -273.15°C on the Celsius (centigrade) scale. The Kelvin unit is identical in interval to the Celsius unit, so direct conversions can be made by adding or subtracting 273. Use of Celsius is still permitted.

0 K = 273°C; absolute zero K
1°C = 273 K

## WEIGHTS AND LINEAR DIMENSIONS

For indications of "weight" the original metric kilogram (kg) will continue to be used as the unit of mass, but it is important to note that the kilogram will no longer apply for force, for which the SI unit is the Newton (N), which is a kilogram meter per second squared. The Newton is that force which, when applied to a body having a mass of one kilogram, gives it an acceleration of one meter per second squared.

"Weight" in itself will no longer apply, since this is an ambiguous term, so the kilogram in effect should only be used as the unit of mass. Undoubtedly, though, it will continue to be common parlance to use the word "weight" when referring to the mass of an object.

The base SI unit for linear dimensions will be the meter, with a wide range of multiples and sub-multiples ranging from exa ($10^{18}$) to atto ($10^{-18}$): A kilometer is a meter x $10^3$, for example, while a millimeter is a meter x $10^{-3}$.

To give an idea of how currently used units convert to SI units, the tables below give examples.

**KILOWATTS (kW) TO HORSEPOWER (hp) (1 Kw = 1.34102 hp)**

| kW | hp | kW | hp | kW | hp | kW | hp | kW | hp |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.341 | 21 | 28.161 | 41 | 54.982 | 61 | 81.802 | 81 | 108.623 |
| 2 | 2.682 | 22 | 29.502 | 42 | 56.323 | 62 | 83.143 | 82 | 109.964 |
| 3 | 4.023 | 23 | 30.843 | 43 | 57.664 | 63 | 84.484 | 83 | 111.305 |
| 4 | 5.364 | 24 | 32.184 | 44 | 59.005 | 64 | 85.825 | 84 | 112.646 |
| 5 | 6.705 | 25 | 33.526 | 45 | 60.346 | 65 | 87.166 | 85 | 113.987 |
| 6 | 8.046 | 26 | 34.867 | 46 | 61.687 | 66 | 88.507 | 86 | 115.328 |
| 7 | 9.387 | 27 | 36.208 | 47 | 63.028 | 67 | 89.848 | 87 | 116.669 |
| 8 | 10.728 | 28 | 37.549 | 48 | 64.369 | 68 | 91.189 | 88 | 118.010 |
| 9 | 12.069 | 29 | 38.890 | 49 | 65.710 | 69 | 92.530 | 89 | 119.351 |
| 10 | 13.410 | 30 | 40.231 | 50 | 67.051 | 70 | 93.871 | 90 | 120.692 |
| 11 | 14.751 | 31 | 41.572 | 51 | 68.392 | 71 | 95.212 | 91 | 122.033 |
| 12 | 16.092 | 32 | 42.913 | 52 | 69.733 | 72 | 96.553 | 92 | 123.374 |
| 13 | 17.433 | 33 | 44.254 | 53 | 71.074 | 73 | 97.894 | 93 | 124.715 |
| 14 | 18.774 | 34 | 45.595 | 54 | 72.415 | 74 | 99.235 | 94 | 126.056 |
| 15 | 20.115 | 35 | 46.936 | 55 | 73.756 | 75 | 100.577 | 95 | 127.397 |
| 16 | 21.456 | 36 | 48.277 | 56 | 75.097 | 76 | 101.918 | 96 | 128.738 |
| 17 | 22.797 | 37 | 49.618 | 57 | 76.438 | 77 | 103.259 | 97 | 130.079 |
| 18 | 24.138 | 38 | 50.959 | 58 | 77.779 | 78 | 104.600 | 98 | 131.420 |
| 19 | 25.479 | 39 | 52.300 | 59 | 79.120 | 79 | 105.941 | 99 | 132.761 |
| 20 | 26.820 | 40 | 53.641 | 60 | 80.461 | 80 | 107.282 | 100 | 134.102 |

**POUNDS FORCE FEET (lbf ft) TO NEWTON METERS (Nm) (1 lbf ft = 1.35582 Nm)**

| lbf ft | Nm | lbf ft | Nm | lbf ft | Nm | lbf ft | Nm | lbf ft | Nm |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.356 | 21 | 28.472 | 41 | 55.589 | 61 | 82.705 | 81 | 109.821 |
| 2 | 2.712 | 22 | 29.828 | 42 | 56.944 | 62 | 84.061 | 82 | 111.177 |
| 3 | 4.067 | 23 | 31.184 | 43 | 58.300 | 63 | 85.417 | 83 | 112.533 |
| 4 | 5.423 | 24 | 32.540 | 44 | 59.656 | 64 | 86.772 | 84 | 113.889 |
| 5 | 6.779 | 25 | 33.896 | 45 | 61.012 | 65 | 88.128 | 85 | 115.245 |
| 6 | 8.135 | 26 | 35.251 | 46 | 62.368 | 66 | 89.484 | 86 | 116.601 |
| 7 | 9.491 | 27 | 36.607 | 47 | 63.724 | 67 | 90.840 | 87 | 117.956 |
| 8 | 10.847 | 28 | 37.963 | 48 | 65.079 | 68 | 92.196 | 88 | 119.312 |
| 9 | 12.202 | 29 | 39.319 | 49 | 66.435 | 69 | 93.552 | 89 | 120.668 |
| 10 | 13.558 | 30 | 40.675 | 50 | 67.791 | 70 | 94.907 | 90 | 122.024 |
| 11 | 14.914 | 31 | 42.030 | 51 | 69.147 | 71 | 96.263 | 91 | 123.380 |
| 12 | 16.270 | 32 | 43.386 | 52 | 70.503 | 72 | 97.619 | 92 | 124.715 |
| 13 | 17.626 | 33 | 44.742 | 53 | 71.808 | 73 | 98.975 | 93 | 126.001 |
| 14 | 18.981 | 34 | 46.098 | 54 | 73.214 | 74 | 100.331 | 94 | 127.447 |
| 15 | 20.337 | 35 | 47.454 | 55 | 74.570 | 75 | 101.687 | 95 | 128.803 |
| 16 | 21.693 | 36 | 48.810 | 56 | 75.926 | 76 | 103.042 | 96 | 130.159 |
| 17 | 23.049 | 37 | 50.165 | 57 | 77.282 | 77 | 104.398 | 97 | 131.515 |
| 18 | 24.405 | 38 | 51.521 | 58 | 78.638 | 78 | 105.754 | 98 | 132.870 |
| 19 | 25.761 | 39 | 52.877 | 59 | 79.993 | 79 | 107.110 | 99 | 134.226 |
| 20 | 27.116 | 40 | 54.233 | 60 | 81.349 | 80 | 108.466 | 100 | 135.582 |

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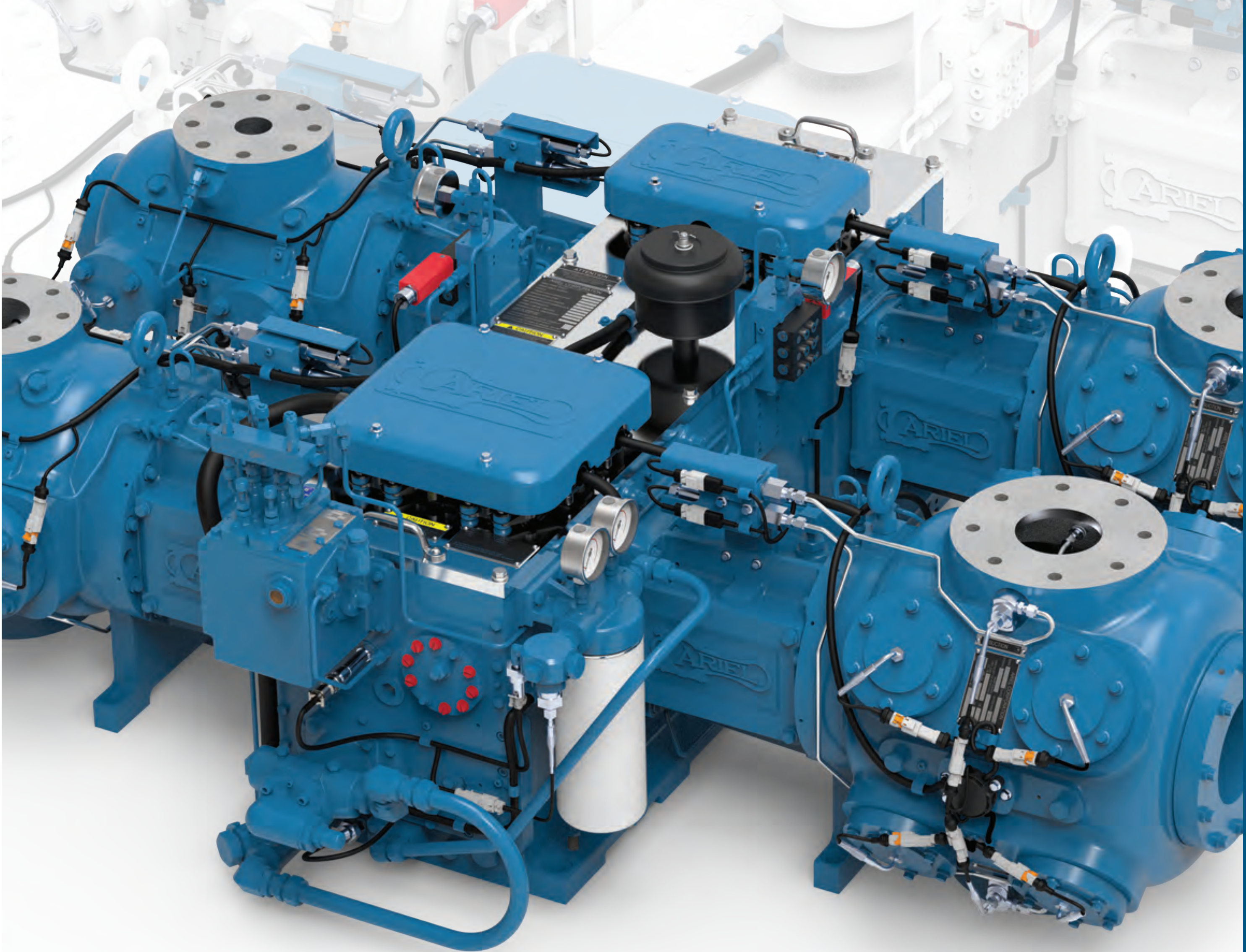



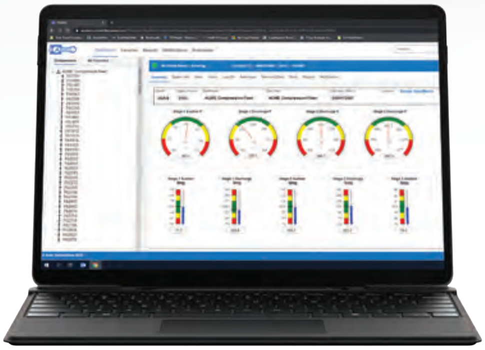

# CENTRIFUGAL COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Axial Flow: Multiple Stage | Axial Flow: Fixed Stator Vanes | Axial Flow: Variable Stator Vanes | Radial Flow: Single Stage | Radial Flow: Multiple Stage | Radial Flow: Horizontally Split | Radial Flow: Vertically Split | Radial Flow: Integral Gear | Radial Flow: Integral Electric | Thermal: Single Stage | Thermal: Multiple Stage | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| AERZENER MASCHINENFABRIK | * | AT and TB Turbo Blowers | | | | X | | | | | | | | | 212 | 19,070 | 6 | 540 | 17.5 | 1.2 | 2 | 871.65 | 650 | 20000 | 45400 |
| ATLAS COPCO GAS AND PROCESS | 70, 71 | GT-Series | | | | X | X | | | X | | | | OF | 140 | 293,100 | 4 | 8300 | 2970 | 205 | 4 | 46500 | 35000 | | 52000 |
| | | T-Series | | | | X | | | | | | | | OF | 6060 | 38,000 | 170 | 1080 | 623.7 | 43 | 1.25 | 13300 | 9900 | | 3600 |
| | | RT-Series | | | | | X | | | | | | | OF | 188,000 | 293,100 | 5330 | 8300 | 101 | 7 | 2.5 | 40000 | 30000 | | 6500 |
| BAKER HUGHES | Inside Front Cover, 65 | AN (Air Service) | X | | X | | | | | | | | | OF | 60,000 | 355,000 | 1600 | 10000 | 362.5 | 25 | | 95200 | 70000 | 3000 | 10000 |
| | | AN (LNG Service) | X | | X | | | | | | | | | OF | 60,000 | 355,000 | 1600 | 10000 | 360 | 25 | | 95200 | 70000 | 3000 | 10000 |
| | | BCL-HP (>350bara) | | | | | X | | X | | | | | OF | 350 | 7060 | 10 | 200 | 14500 | 1000 | | 40800 | 30000 | 7000 | 20000 |
| | | BCL-LP/MP (<350 bara) | | | | | X | | X | | | | | OF | 350 | 100,000 | 10 | 2700 | 5075 | 350 | | 54400 | 40000 | 3000 | 20000 |
| | | MCL | | | | | X | X | | | | | | OF | 3530 | | 100 | 8500 | 870 | 60 | | 95200 | 70000 | 3000 | 20000 |
| | | PCL | | | | X | X | | X | | | | | OF | 2100 | 60,000 | 60 | 1700 | 1890 | 130 | | 54400 | 40000 | 3600 | 18000 |
| | | SRL | | | | X | X | | | X | | | | OF | 1060 | 215,000 | 30 | 6000 | 2900 | 200 | | 43500 | 32000 | 1500 | 30000 |
| | | SRL (Overhung, Single Stage) | | | | X | | | | | | | | OF | 1060 | 60,000 | 30 | 1700 | 1380 | 95 | | 20400 | 32000 | 1500 | 20000 |
| | | ICL | | | | | X | | X | | X | | | OF | 880 | 20,000 | 25 | 550 | 5075 | 350 | | 21500 | 16000 | 1500 | 30000 |
| | | ICL single stage | | | | X | | | X | | X | | | OF | 880 | 20,500 | 25 | 580 | 1740 | 120 | | 19600 | 14600 | 1500 | 30000 |
| | | Blue-C | | | | | X | | X | | X | | | OF | 52,500 | 530,000 | 1500 | 15000 | 2980 | 205 | | 18620 | 14000 | 3000 | 11000 |
| BORSIG ZM COMPRESSION GMBH | 77 | BTC Series | | | | X | X | | | X | | | | OF | 425 | 282,500 | 12 | 8000 | 2900 | 200 | 3 | 33525 | 25000 | | 48000 |
| COMOTI | 99 | CCAE 9-125 | | | | | X | | X | X | X | | X | OF | 2154 | 3072 | 61 | 87 | 116 | 8 | | 939 | 700 | 22900 | 36800 |
| | | CCAE 9-144 | | | | | X | | X | X | X | | X | OF | 2649 | 3531 | 75 | 100 | 116 | 8 | | 939 | 700 | 22900 | 36800 |
| | | CCAE 9-300 | | | | | X | | X | X | X | | X | OF | 5297 | 7345 | 150 | 208 | 131 | 9 | | 1475 | 1100 | 22900 | 36800 |
| | | CCAE 12-300 | | | | | X | | X | X | X | | X | OF | 5297 | 7345 | 150 | 208 | 160 | 11 | | 1944 | 1450 | 16500 | 31000 |
| | | CCAE 21-300 | | | | | X | | X | X | X | | X | OF | 5297 | 7345 | 150 | 208 | 290 | 20 | | 2414 | 1800 | 16500 | 31000 |
| | | CCAE 15-300 | | | | | X | | X | X | X | | X | OF | 5368 | 7593 | 152 | 215 | 203 | 14 | | 2146 | 1600 | 16500 | 31000 |
| | | CCAE 25-350 | | | | | X | | X | X | X | | X | OF | | | 172 | 245 | | 25 | | | 2200 | | |
| CRYOSTAR SAS | * | CM 400 / CM 300 | | | | X | | | X | X | | | | OF | | | 58 | 666 | | 6 | 2 | | 1000 | | 11000 |
| | | CM 2-200 / 300 | | | | | 2, 4, 6 | | X | X | | | | OF | | | 42 | 175 to 100 | | 10 to 25 | 2 | | 700 to 1000 | | 30000 |

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# CENTRIFUGAL COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Axial Flow | | | Radial Flow | | | | | | Thermal | | OF = Oil Free OI = Oil Injected | Inlet Flow Range | | | | MAWP Maximum Allowable Working Pressure | | Compression Ratio (Per Stage) | Maximum Input Power | | Speed Range (rpm) | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Multiple Stage | Fixed Stator Vanes | Variable Stator Vanes | Single Stage | Multiple Stage | Horizontally Split | Vertically Split | Integral Gear | Integral Electric | Single Stage | Multiple Stage | | acfm min | acfm max | m³/min min | m³/min max | psig | bar | | hp | kW | min | max |
| ELLIOTT GROUP | 61, 137, Inside Back Cover | A | X | X | X | | | | | | | | | OF | | 441,000 | | 12500 | 90 | 6.2 | | 175000 | 130000 | | 8025 |
| | | M | | | | | X | X | | | | | | OF | | 896,000 | | 25370 | 1000 | 69 | | 225000 | 170000 | | 20000 |
| | | MB | | | | | X | | X | | | | | OF | | 319,000 | | 9000 | 10000 | 690 | | 225000 | 170000 | | 20000 |
| | | PH | | | | X | | | X | | | | | OF | | 75,000 | | 2100 | 800 | 55 | | 15000 | 11000 | | 13500 |
| | | TC | | | | X | | | X | | | | | OF | | 90,000 | | 2500 | 725 | 50 | | 40000 | 30000 | | 10000 |
| FIMA MASCHINENBAU GMBH | * | F1 Series | | | | X | | | X | X | | | | OF | | | 3 | 5000 | | 100 | 2.5 | 6800 | 5000 | | 35000 |
| | | F3 Series | | | | | X | | X | X | | | | OF | | | 3 | 5000 | | 100 | 2.5 | 6800 | 5000 | | 35000 |
| | | F2 Series | | | | X | X | | X | | | | | OF | | | 3 | 85 | | 240 | 1.6 | 408 | 600 | | 11000 |
| | | F4 Series (Zone 0) | | | | X | | | X | | | | | OF | | | 3 | 150 | | 1.3 | 1.2 | 74 | 55 | | 7700 |
| FS-ELLIOTT | * | PAP Series | | | | X | X | X | | X | | | | OF | 900 to 15,000 | 2200 to 24,500 | 25 to 425 | 60 to 695 | 175 to 450 | 12.1 to 31 | up to 3.1 | 500 to 6000 | 375 to 4475 | 1450 to 2950 | 3600 |
| | | Polaris Series | | | | X | X | X | | X | | | | OF | 900 to 5500 | 2200 to 12,000 | 25 to 155 | 60 to 340 | 150 | 10.5 | up to 3.0 | 450 to 2600 | 335 to 2600 | 2950 | 3600 |
| GARO S.P.A. | * | VC | | | | X | X | | X | | | | | OF | | | 17 | 833 | | 2 | 1.2 | 536 | 400 | 3000 | 6000 |
| | | VAP | | | | X | X | | X | | | | | OF | | | 17 | 833 | | 90 | 1.2 | 2682 | 2000 | 3000 | 6000 |
| | | CC (Galileo) | | | | X | X | | X | X | | | | OF | | | 17 | 833 | | 90 | 3 | 2682 | 2000 | 4000 | 42000 |
| HANWHA POWER SYSTEMS | * | SM3000 | | | | | X | X | | X | | | | OF | 1950 | 3100 | 55 | 88 | 264 | 18 | | 913 | 680 | | 3600 |
| | | SM4000 | | | | | X | X | | X | | | | OF | 3100 | 4950 | 88 | 140 | 264 | 18 | | 1350 | 1010 | | 3600 |
| | | SM5000 | | | | | X | X | | X | | | | OF | 4950 | 8850 | 140 | 250 | 264 | 18 | | 1800 | 1540 | | 3600 |
| | | SM6000 | | | | | X | X | | X | | | | OF | 8850 | 12,400 | 250 | 350 | 264 | 18 | | 3150 | 2350 | | 3600 |
| | | SM2100 | | | | | X | X | | X | | | | OF | 700 | 1950 | 20 | 55 | 164 | 12 | | 450 | 335 | | 3600 |
| | | SM3100 | | | | | X | X | | X | | | | OF | 1950 | 3250 | 55 | 92 | 187 | 13 | | 780 | 580 | | 3600 |
| | | SM4100 | | | | | X | X | | X | | | | OF | 3250 | 5300 | 92 | 150 | 187 | 13 | | 1200 | 930 | | 3600 |
| | | SM5100 | | | | | X | X | | X | | | | OF | 5300 | 8850 | 150 | 250 | 187 | 13 | | 2010 | 1500 | | 3600 |
| | | SM6100 | | | | | X | X | | X | | | | OF | 8850 | 14,400 | 250 | 408 | 187 | 13 | | 3350 | 2500 | | 3600 |
| | | SM7100 | | | | | X | X | | X | | | | OF | 14,400 | 18,800 | 408 | 533 | 187 | 13 | | 4155 | 3100 | | 3600 |
| | | SE-32 | | | | X | X | X | | X | | | | OF | | | | 150 | | 60 | | 1000 | 800 | | 3600 |
| | | SE-45 | | | | X | X | X | | X | | | | OF | | | | 400 | | 60 | | 4000 | 3000 | | 3600 |
| *continued* | | SE-65 | | | | X | X | X | | X | | | | OF | | | | 800 | | 60 | | 6700 | 5000 | | 1800 |

# CENTRIFUGAL COMPRESSORS

## 2022 BASIC SPECIFICATIONS

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|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| HANWHA POWER SYSTEMS | * | SE-82 | | | | X | X | X | | X | | | | OF | | | | 1200 | | 60 | | 10700 | 8000 | | 1800 |
| | | SE-90 | | | | X | X | X | | X | | | | OF | | | | 1700 | | 60 | | 17400 | 13000 | | 1800 |
| | | SE-110 | | | | X | X | X | | X | | | | OF | | | | 2500 | | 60 | | 28000 | 21000 | | 1800 |
| | | SE-130 | | | | X | X | X | | X | | | | OF | | | | 5000 | | 60 | | 28000 | 21000 | | 1800 |
| HITACHI, LTD. | * | 2BCH | | | | | X | | X | | | | | OF | | | 3 to 8 | 1700 to 6000 | | 45 to 750 | 5 to 10 | | 30000 to 50000 | 2500 to 3500 | 14000 to 18000 |
| HOWDEN | 104, 105 | Howden Periflow | | | | X | | X | | | X | | | OF | 60 | 3026 | 2 | 86 | 3000 | 200 | 3 | 1340 | 1000 | 500 | 6000 |
| | | Howden ČKD RL | | | | X | X | X | X | | | | | OF | 2500 | 24,000 | 70 | 633 | 20 | 1.4 | 2 | 1600 | 1200 | 3000 | 27000 |
| | | Howden ČKD RLV | | | | X | X | X | X | | | | | OF | 1000 | 13,000 | 30 | 370 | 23 | 1.6 | 2 | 1350 | 1000 | 12000 | 27000 |
| | | Howden ČKD RD | | | | X | X | X | X | | | | | OF | 50,000 | 280,000 | 1400 | 8000 | 20 | 1.4 | 1 | 13500 | 10000 | 3000 | 27000 |
| | | Howden ČKD RK Series | | | | X | X | | | X | | | | OF | 5300 | 81,000 | 140 | 2100 | 2320 | 160 | 2 | 17200 | 12800 | 3000 | 37000 |
| | | Howden ČKD RS Series | | | | X | X | | X | | | | | OF | 450 | 18,000 | 12 | 470 | 1400 | 96 | 2 | 13000 | 9900 | 3000 | 27000 |
| | | Howden ČKD RM Series | | | | X | X | X | | | | | | OF | 4000 | 216,000 | 100 | 5700 | 502 | 35 | 2 | 34000 | 25400 | 3000 | 27000 |
| | | Howden ČKD RP Series | | | | | X | X | | | | | | OF | 2450 | 37,000 | 65 | 1000 | 310 | 21.4 | 2 | 4000 | 3000 | 3000 | 27000 |
| | | SG26 | | | | X | | | | X | | | | OF | 4238 | 6357 | 120 | 180 | 29 | 2 | 3 | 268 | 200 | 5000 | 33000 |
| | | SG30 | | | | X | | X | | X | | | | OF | 4238 | 8476 | 120 | 240 | 29 | 2 | 3 | 603 | 450 | 5000 | 33000 |
| | | SG35 | | | | X | | X | | X | | | | OF | 4238 | 10,595 | 120 | 300 | 29 | 2 | 3 | 671 | 500 | 5000 | 33000 |
| | | SG40 | | | | X | | X | | X | | | | OF | 6357 | 14,832 | 180 | 420 | 29 | 2 | 3 | 1073 | 800 | 5000 | 33000 |
| | | SG45 | | | | X | | X | | X | | | | OF | 8476 | 19,070 | 240 | 540 | 29 | 2 | 3 | 1341 | 1000 | 5000 | 33000 |
| | | SG52 | | | | X | | X | | X | | | | OF | 12,713 | 25,427 | 360 | 720 | 29 | 2 | 3 | 2146 | 1600 | 5000 | 33000 |
| | | SG60 | | | | X | | X | | X | | | | OF | 14,832 | 31,784 | 420 | 900 | 29 | 2 | 3 | 2414 | 1800 | 5000 | 33000 |
| | | SG65 | | | | X | | X | | X | | | | OF | 16,951 | 33,903 | 480 | 960 | 29 | 2 | 3 | 2682 | 2000 | 5000 | 33000 |
| | | SG70 | | | | X | | X | | X | | | | OF | 21,189 | 40,259 | 600 | 1140 | 29 | 2 | 3 | 3487 | 2600 | 5000 | 33000 |
| | | SG80 | | | | X | | X | | X | | | | OF | 25,427 | 52,973 | 720 | 1500 | 29 | 2 | 3 | 4023 | 3000 | 5000 | 33000 |
| | | SG92 | | | | X | | X | | X | | | | OF | 33,903 | 63,567 | 960 | 1800 | 29 | 2 | 3 | 5364 | 4000 | 5000 | 33000 |
| | | SG105 | | | | X | | X | | X | | | | OF | 44,497 | 84,757 | 1260 | 2400 | 29 | 2 | 3 | 6705 | 5000 | 5000 | 33000 |
| *continued* | | KK&K SF (2.8 - 14) | | | | X | | | X | | | | | OF | 2600 | 250,000 | 75 | 7000 | 45 | 3 | 3.5 | 22000 | 16000 | 2500 | 40000 |

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# CENTRIFUGAL COMPRESSORS

2022 BASIC SPECIFICATIONS

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|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **HOWDEN** | 104, 105 | **KK&K SF (18 - 22.4)** | | | | X | | X | | | | | | OF | 140,000 | 420,000 | 4000 | 12000 | 30 | 2 | 2.3 | 22000 | 16000 | 1800 | 4000 |
| | | **KK&K SF (HP)** | | | | X | | | X | | | | | OF | 2600 | 105,000 | 75 | 3000 | 370 | 50 | 3 | 13500 | 10000 | 3600 | 40000 |
| | | **KK&K SFG** | | | | X | | | X | X | | | | OF | 1800 | 175,000 | 50 | 5000 | 45 | 3 | 3.5 | 11000 | 8000 | 3600 | 40000 |
| | | **KK&K SFG (HP)** | | | | X | | | X | X | | | | OF | 1800 | 105,000 | 50 | 3000 | 370 | 50 | 3 | 11000 | 10000 | 3600 | 40000 |
| | | **KK&K SL** | | | | X | | X | | | | | | OF | 9000 | 237,000 | 250 | 6700 | 30 | 2 | 2.3 | 11000 | 8000 | 2800 | 15000 |
| | | **KK&K R** | | | | X | | | X | | | | | OF | 40 | 200,000 | 10 | 6000 | 30 | 2 | 1.7 | 7000 | 5000 | 1200 | 15000 |
| | | **KK&K R (HP)** | | | | X | | | X | | | | | OF | 40 | 30,000 | 10 | 800 | 370 | 25 | 1.5 | 3500 | 2500 | 1800 | 15000 |
| | | **KK&K ST** | | | | X | | | X | | | | | OF | 2600 | 85,000 | 75 | 2400 | 45 | 3 | 3.5 | 8000 | 6000 | 8000 | 30000 |
| | | **KK&K ST (HP)** | | | | X | | | X | | | | | OF | 2600 | 85,000 | 75 | 2400 | 370 | 50 | 2.8 | 8000 | 6000 | 8000 | 25000 |
| | | **Roots OIB** | | | | X | | X | | | | | | OF | 3000 | 230,000 | 85 | 6500 | 25 | 1.72 | 2.75 | 18000 | 13500 | 2500 | 30000 |
| | | **Roots H** | | | | | X | X | | | | | | OF | 5000 | 90,000 | 140 | 2550 | 25 | 1.72 | 1.8 | 18000 | 13500 | 3000 | 20000 |
| | | **ExVel Xr** | | | | X | | | X | | | | | OF | 5000 | 350,000 | 140 | 10000 | 1450 | 10 | 2 | 7000 | 5000 | 1800 | 6000 |
| | | **HV-TURBO / Turblex KA2** | | | | X | | | X | X | | | | OF | 883 | 2354 | 25 | 67 | 29 | 2 | 3 | 215 | 160 | 19500 | 40500 |
| | | **HV-TURBO / Turblex KA5** | | | | X | | | X | X | | | | OF | 2060 | 4708 | 58 | 133 | 33 | 2.3 | 3.3 | 536 | 400 | 13000 | 31500 |
| | | **HV-TURBO / Turblex KA10** | | | | X | | | X | X | | | | OF | 3531 | 8828 | 100 | 250 | 33 | 2.3 | 3.3 | 872 | 650 | 10000 | 23000 |
| | | **HV-TURBO / Turblex KA22** | | | | X | | | X | X | | | | OF | 6474 | 14,124 | 183 | 400 | 29 | 2 | 3 | 1207 | 900 | 7800 | 17800 |
| | | **HV-TURBO / Turblex KA44** | | | | X | | | X | X | | | | OF | 11,770 | 23,540 | 333 | 667 | 27 | 1.9 | 2.9 | 2548 | 1900 | 7500 | 14200 |
| | | **HV-TURBO / Turblex KA66** | | | | X | | | X | X | | | | OF | 18,832 | 38,253 | 533 | 1083 | 26 | 1.8 | 2.8 | 3889 | 2900 | 5000 | 11300 |
| | | **HV-TURBO / Turblex KA80** | | | | X | | | X | X | | | | OF | 29,425 | 54,142 | 833 | 1533 | 23 | 1.6 | 2.6 | 3353 | 2500 | 4300 | 8700 |
| | | **HV-TURBO / Turblex KA100** | | | | X | | | X | X | | | | OF | 44,138 | 64,735 | 1250 | 1833 | 17 | 1.2 | 2.2 | 3487 | 2600 | 3600 | 6500 |
| **INGERSOLL RAND** | * | **Centac Series** | | | | X | X | | X | X | | | | OF | 1300 to 12,500 | 2100 to 30,000 | 42 to 350 | 60 to 850 | 35 to 610 | 3 to 42 | maxim of 3 | 350 to 6000 | 270 to 4500 | | 1800 to 3600 |
| | | **TA Series** | | | | | X | X | | X | | | | OF | 1300 to 12,500 | 6000 to 24,000 | 14 to 339 | 48 to 679 | 150 to 1160 | 10 to 80 | maxim of 3 | 350 to 5500 | 250 to 4100 | | 3600 |
| | | **MSG Series** | | | | | X | X | | X | | | | OF | 2500 to 50,000 | 10,000 to 135,000 | 70 to 1416 | 283 to 3823 | 125 to 1450 | 8 to 100 | maxim of 3 | 4000 to 25,000 | 3000 to 19,000 | | 1800 to 3600 |
| **KOBELCO** | * | **VH Series** | | | | | X | | X | | | | | OF | 1059 | 58,951 | 30 | 1670 | 5000 | 350 | | 28000 | 20000 | 5000 | 18000 |
| | | **VGS/VGSP Series** | | | | X | X | | | X | | | | OF | 1765 | 264,750 | 50 | 7500 | 1420 | 100 | | 67000 | 50000 | 3000 | 40000 |
| *continued* | | **DH Series** | | | | X | | | X | | | | | OF | 1059 | 105,900 | 30 | 3000 | 1300 | 90 | | 33500 | 25000 | 1000 | 18000 |

# CENTRIFUGAL COMPRESSORS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Axial Flow | | | Radial Flow | | | | | | Thermal | | OF = Oil Free OI = Oil Injected | Inlet Flow Range | | | | MAWP Maximum Allowable Working Pressure | | Compression Ratio (Per Stage) | Maximum Input Power | | Speed Range (rpm) | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Multiple Stage | Fixed Stator Vanes | Variable Stator Vanes | Single Stage | Multiple Stage | Horizontally Split | Vertically Split | Integral Gear | Integral Electric | Single Stage | Multiple Stage | | acfm min | acfm max | m³/min min | m³/min max | psig | bar | | hp | kW | min | max |
| KOBELCO | * | V-VS-VSS Series | | | | X | X | X | | | | | | OF | 1059 | 200,151 | 30 | 5670 | 700 | 50 | | 21000 | 15000 | 5000 | 18000 |
| | | VG / VGP Series | | | | X | X | | | X | | | | OF | 1765 | 264,750 | 50 | 7500 | 1420 | 100 | | 67000 | 50000 | 3000 | 60000 |
| MAN ENERGY SOLUTIONS - Turbo Compressors | 118, 119 | A, AV | X | X | X | | | | | | | | | OF | 29,000 | 900,000 | 835 | 25450 | 362 | 25 | | | 120000 | | |
| | | AG, AK, AKF | X | X | X | | | | | | | | | OF | 29,000 | 900,000 | 835 | 25450 | 220 | 15 | | | 125000 | | |
| | | AR | X | X | X | X | X | | | | | | | OF | 29,000 | 900,000 | 835 | 25450 | 362 | 25 | | | 165000 | | |
| | | HOFIM | | | | | X | | X | | X | | | OF | 95 | 18,000 | 3 | 500 | 4350 | 300 | | 24000 | 18000 | 3000 | 16000 |
| | | RB | | | | X | X | | X | | | | | OF | 95 | 190,000 | 3 | 5400 | 14500 | 1000 | | 107000 | 80000 | | 24000 |
| | | TURBAIR (RC) | | | | | X | X | | | | | | OF | 5300 | 115,000 | 150 | 3300 | 29 | 2 | | | 5400 | | |
| | | RG | | | | X | X | | X | X | | | | OF | 1000 | 400,000 | 28 | 12000 | 3600 | 250 | | | 80000 | | 50000 |
| | | RH | | | | X | X | X | | | | | | OF | 450 | 417,000 | 13 | 11800 | 1160 | 80 | | | 80000 | | |
| | | RIK, RIKT, RIO | | | | | X | X | | | | | | OF | 10,500 | 440,000 | 300 | 12500 | 300 | 21 | | | 100000 | 15000 | 3000 |
| | | MOPICO (RM) | | | | X | X | | X | | X | | | OF | 530 | 18,000 | 15 | 500 | 2200 | 150 | | 24000 | 18000 | 3000 | 16000 |
| | | RV | | | | X | X | | X | | | | | OF | 450 | 50,000 | 13 | 1400 | 2175 | 150 | | 67000 | 50000 | | |
| | | RV-F | | | | X | | | X | | | X | | OF | 285 | 88,750 | 8 | 2500 | 14500 | 100 | | | 20000 | 3000 | 30000 |
| | | subsea HOFIM | | | | | X | | X | | X | | | OF | 150 | 18,000 | 4 | 500 | 4350 | 300 | | 24000 | 18000 | 3100 | 16000 |
| MITSUBISHI HEAVY INDUSTRIES COMPRESSOR INTERNATIONAL | * | H-Type | | | | X | X | X | | | | | | OF/OI | | 530,000 | | 15000 | 870 | 60 | 2 | 134100 | 100000 | 1500 | 20000 |
| | | V-Type | | | | X | X | | X | | | | | OF/OI | | 212,000 | | 6000 | 14500 | 1000 | 2 | 134100 | 100000 | 3000 | 20000 |
| | | Integrally Geared | | | | X | X | | | X | | | | OI | | 600,000 | | 17000 | 2900 | 200 | 2.5 | 93900 | 70000 | 1500 | 100000 |
| MITSUI E&S | * | H Series | | | | | X | X | | | | | | OI | 353 | 177,000 | 10 | 5000 | 725 | 50 | | 42900 | 32000 | | |
| | | V Series | | | | | X | | X | | | | | OI | 353 | 35,300 | 10 | 1000 | 9400 | 650 | | 42900 | 32000 | | |
| | | MA Series | X | | X | | | | | | | | | OI | | 353,000 | | 9200 | 145 | 10 | 7 | 134100 | 100000 | | |
| SIEMENS ENERGY | * | STC-GV | | | | X | X | | X | X | | | | OF | 880 | 590,000 | 25 | 16700 | 2900 | 200 | | 80500 | 60000 | | 45000 |
| | | STC-SV | | | | X | X | | X | | | | | OF | 140 | 283,000 | 4 | 8000 | 14500 | 1000 | | 134000 | 100000 | | 20000 |
| | | STC-SH | | | | X | X | X | | | | | | OF | 140 | 353,000 | 4 | 10000 | 1450 | 100 | | 134000 | 100000 | | 20000 |
| *continued* | | STC-GC | | | | | X | | | X | | | | OF | 5900 | 236,000 | 167 | 6670 | 290 | 20 | | 40200 | 30000 | | 45000 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# CENTRIFUGAL COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Axial Flow: Multiple Stage | Axial Flow: Fixed Stator Vanes | Axial Flow: Variable Stator Vanes | Radial Flow: Single Stage | Radial Flow: Multiple Stage | Radial Flow: Horizontally Split | Radial Flow: Vertically Split | Radial Flow: Integral Gear | Radial Flow: Integral Electric | Thermal: Single Stage | Thermal: Multiple Stage | OF = Oil Free, OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| SIEMENS ENERGY | * | STC-SX | X | X | X | | | X | | | | | | OF | 29,400 | 766,000 | 833 | 21700 | 102 | 7 | | 53600 | 40000 | | 9000 |
| | | STC-SR | X | X | X | X | X | X | | | | | | OF | 29,400 | 766,000 | 833 | 21700 | 232 | 16 | | 134000 | 100000 | | 9000 |
| | | STC-GVT | | | | | X | | | X | | | | OF | 880 | 283,000 | 25 | 8000 | 870 | 60 | | 40200 | 30000 | | 45000 |
| | | DATUM | | | | X | X | X | X | | | | | OF | | 470,900 | | 13300 | 14500 | 1000 | | 181000 | 135000 | | 26500 |
| | | Axial | X | X | X | | | | | | | | | OF | 75,000 | 700,000 | 2120 | 20000 | 80 | 5.5 | | 125000 | 93250 | | 8000 |
| | | RFA, RFBB | | | | X | X | | X | | | | | OF | 12,710 | 62,400 | 360 | 1775 | 2250 | 155 | | 75000 | 56000 | | 13800 |
| SOLAR TURBINES INCORPORATED | Prime Movers Tab | C16 | | | | X | X | | | | | | | | 200 | 2200 | 4 | 60 | 4500 | 310 | | 13100 | 9800 | | 23800 |
| | | C31 | | | | X | X | | X | | | | | | 500 | 4000 | 15 | 113 | 5000 | 344 | | 20000 | 14900 | | 16000 |
| | | C33 | | | | X | X | | X | | | | | | 800 | 9500 | 23 | 270 | 2700 | 186 | | 17270 | 12900 | | 19000 |
| | | C40 | | | | X | X | | X | | | | | | 600 | 9000 | 17 | 255 | 2500 | 172 | | 29500 | 21700 | | 14300 |
| | | C41 | | | | X | X | | X | | | | | | 750 | 18,000 | 21 | 510 | 3750 | 259 | | 41976 | 31300 | | 14300 |
| | | C41D | | | | X | X | | X | | | | | | 750 | 18,000 | 21 | 510 | 3750 | 259 | | 41976 | 31300 | | 14300 |
| | | C50 | | | | X | X | | X | | | | | | 2000 | 20,000 | 57 | 565 | 1500 | 103 | | 31915 | 23800 | | 14000 |
| | | C51 | | | | X | X | | X | | | | | | 2000 | 25,000 | 57 | 710 | 3000 | 207 | | 52333 | 39000 | | 12000 |
| | | C51D | | | | X | X | | X | | | | | | | 12,000 | | 339 | 3000 | 207 | | | | | 12000 |
| | | C61 | | | | X | X | | X | | | | | | 2800 | 35,000 | 79 | 990 | 3000 | 207 | | 87640 | 65400 | | 10200 |
| | | C40 Pipeline | | | | X | X | | X | | | | | | 1500 | 11,000 | 42 | 300 | 1600 | 186 | | 16223 | 12100 | | 15500 |
| | | C45 Pipeline | | | | X | X | | X | | | | | | 3800 | 18,500 | 108 | 525 | 2250 | 124 | | 35206 | 26300 | | 12000 |
| | | C65 Pipeline | | | | X | X | | X | | | | | | 5000 | 24,000 | 142 | 680 | 1600 | 110 | | 34968 | 26100 | | 10500 |
| | | C75 Pipeline | | | | X | X | | X | | | | | | 2420 | 30,000 | 68 | 850 | 2250 | 155 | | 76138 | 56730 | | 8860 |
| | | C85 Pipeline | | | | X | X | | X | | | | | | 10,000 | 45,000 | 283 | 1275 | 1600 | 110 | | 77707 | 57900 | | 7000 |
| SUNDYNE CORPORATION | * | LMC | | | | X | | | X | X | | | | OF | 50 | 3600 | 85 | 6120 | 2160 | 149 | 4 | 550 | 410 | 2950 | 34000 |
| | | BMC | | | | X | | | X | X | | | | OF | 50 | 3600 | 85 | 6120 | 2160 | 149 | 4 | 550 | 410 | 2950 | 34000 |
| | | LF-2000 | | | | X | X | | X | X | | | | OF | 100 | 10,200 | 170 | 17300 | 4400 | 304 | 4 | 10000 | 7500 | 5000 | 50000 |
| YORK/FRICK (JCI) | * | M Series | | | | | X | X | | | | | | OF | 400 to 14,200 | 1800 to 23,000 | 11 to 402 | 51 to 651 | 450 to 600 | 31 to 41 | | 2500 to 17,300 | 1865 to 12,900 | | 9410 to 24,980 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# TURBOEXPANDERS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Axial Flow: Multiple Stage | Axial Flow: Fixed Stator Vanes | Axial Flow: Variable Stator Vanes | Radial Flow: Single Stage | Radial Flow: Multiple Stage | Radial Flow: Horizontally Split | Radial Flow: Vertically Split | Radial Flow: Integral Gear | Radial Flow: Integral Electric | Thermal: Single Stage | Thermal: Multiple Stage | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | Mass Flow lb/hr min | Mass Flow lb/hr max | Mass Flow kg/h min | Mass Flow kg/h max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | Temp (°C) | Expansion Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| ATLAS COPCO GAS AND PROCESS | 70, 71 | Radial Inflow Expanders Frame 1 to 10 EC,ECM,EG, ETH | | | X | X | X | | X | X | | | | OF, OI | 50 | 40,000 | 1 | 1200 | | | | | 3000 | 200 | -253 to +550 | 1.2 to 18 | 30,000 | 23000 | 3000 | 105000 |
| BAKER HUGHES | Inside Front Cover, 65 | HIPER | | | | X | | | X | | X | | | OF | | 412 | | 12 | | | | | 1232.5 | 85 | 0 to 100 | | 402 | 300 | | 30000 |
| | | EC 10 | | | | X | | | X | | | | | OF, OI | | 647 | | 18 | | | | | 3335 | 230 | -270 to 315 | | 268 | 200 | | 110000 |
| | | EC 15 | | | | X | | | X | | | | | OF, OI | | 1295 | | 37 | | | | | 3335 | 230 | -270 to 315 | | 1072 | 800 | | 70000 |
| | | EC 20 | | | | X | | | X | | | | | OF, OI | | 2648 | | 75 | | | | | 3335 | 230 | -270 to 315 | | 2010 | 1500 | | 51000 |
| | | EC 25 | | | | X | | | X | | | | | OF, OI | | 3531 | | 100 | | | | | 3335 | 230 | -270 to 315 | | 4020 | 3000 | | 35000 |
| | | EC 30 | | | | X | | | X | | | | | OF, OI | | 4473 | | 127 | | | | | 3335 | 230 | -270 to 315 | | 6700 | 5000 | | 25000 |
| | | EC 40 | | | | X | | | X | | | | | OF, OI | | 7062 | | 200 | | | | | 3335 | 230 | -270 to 315 | | 10050 | 7500 | | 22000 |
| | | EC 50 | | | | X | | | X | | | | | OF, OI | | 10,299 | | 292 | | | | | 2030 | 140 | -270 to 315 | | 14,740 | 11000 | | 19000 |
| | | EC 60 | | | | X | | | X | | | | | OF, OI | | 13,241 | | 375 | | | | | 2030 | 140 | -270 to 315 | | 18,090 | 13500 | | 15000 |
| | | EC 80 | | | | X | | | X | | | | | OF, OI | | 17,067 | | 483 | | | | | 1305 | 90 | -270 to 315 | | 24,120 | 18000 | | 12000 |
| | | EC 100 | | | | X | | | X | | | | | OF, OI | | 22,363 | | 633 | | | | | 1305 | 90 | -110 to 315 | | 26,800 | 20000 | | 7000 |
| | | EC 130 | | | | X | | | X | | | | | OF, OI | | 29,425 | | 833 | | | | | 725 | 50 | -110 to 315 | | 33,500 | 25000 | | 5500 |
| | | EC 160 | | | | X | | | X | | | | | OF, OI | | 44,726 | | 1267 | | | | | 725 | 50 | -110 to 315 | | 46,900 | 35000 | | 4000 |
| | | EC 180 | | | | X | | | X | | | | | OF, OI | | 64,735 | | 1833 | | | | | 725 | 50 | -110 to 315 | | 53,600 | 40000 | | 3000 |
| | | EG 10 | | | | X | X | | X | X | | | | OI | | 647 | | 18 | | | | | 2030 | 140 | -270 to 315 | | 268 | 200 | | 38000 |
| | | EG 15 | | | | X | X | | X | X | | | | OI | | 1295 | | 37 | | | | | 2030 | 140 | -270 to 315 | | 804 | 600 | | 38000 |
| | | EG 20 | | | | X | X | | X | X | | | | OI | | 2648 | | 75 | | | | | 2030 | 140 | -270 to 315 | | 1340 | 1000 | | 38000 |
| | | EG 25 | | | | X | X | | X | X | | | | OI | | 3531 | | 100 | | | | | 2030 | 140 | -270 to 315 | | 2010 | 1500 | | 35000 |
| | | EG 30 | | | | X | X | | X | X | | | | OI | | 4473 | | 127 | | | | | 2030 | 140 | -270 to 315 | | 4020 | 3000 | | 26000 |
| | | EG 40 | | | | X | X | | X | X | | | | OI | | 7062 | | 200 | | | | | 2030 | 140 | -270 to 315 | | 6968 | 5200 | | 20000 |
| | | EG 50 | | | | X | X | | X | X | | | | OI | | 10299 | | 292 | | | | | 2030 | 140 | -270 to 315 | | 10050 | 7500 | | 17000 |
| | | EG 60 | | | | X | X | | X | X | | | | OI | | 13241 | | 375 | | | | | 2030 | 140 | -270 to 315 | | 14070 | 10500 | | 13000 |
| | | EG 80 | | | | X | X | | X | X | | | | OI | | 17067 | | 483 | | | | | 1305 | 90 | -270 to 315 | | 17420 | 13000 | | 8000 |
| continued | | EG 100 | | | | X | X | | X | X | | | | OI | | 22363 | | 633 | | | | | 1305 | 90 | -110 to 315 | | 24120 | 18000 | | 5000 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# TURBOEXPANDERS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Axial Flow: Multiple Stage | Axial Flow: Fixed Stator Vanes | Axial Flow: Variable Stator Vanes | Radial Flow: Single Stage | Radial Flow: Multiple Stage | Radial Flow: Horizontally Split | Radial Flow: Vertically Split | Radial Flow: Integral Gear | Radial Flow: Integral Electric | Thermal: Single Stage | Thermal: Multiple Stage | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | Mass Flow lb/hr min | Mass Flow lb/hr max | Mass Flow kg/h min | Mass Flow kg/h max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | Temp (°C) | Expansion Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| BAKER HUGHES | Inside Front Cover, 65 | EG 130 | | | | X | X | | X | X | | | | OI | | 29425 | | 833 | | | | | 725 | 50 | -110 to 315 | | 31490 | 23500 | | 4000 |
| | | EG 160 | | | | X | | | X | | | | | OI | | 44726 | | 1267 | | | | | 725 | 50 | -110 to 315 | | 38860 | 29000 | | 3000 |
| | | EG 180 | | | | X | | | X | | | | | OI | | 64735 | | 1833 | | | | | 725 | 50 | -110 to 315 | | 46900 | 35000 | | 1500 |
| CRYOSTAR SAS | * | TP Series | | | | X | | | X | | | X | | OI | 30 to 300 | 260 to 8100 | 0.75 to 8 | 7.5 to 230 | 6500 to 100,000 | 66,000 to 2,777,821 | 3000 to 45,000 | 30,000 to 1,260,000 | 1015 to 1450 | 70 to 100 | -196 to 60 | 1.05 - 18 | 268 to 9655 | 200 to 7200 | 1400 to 10,500 | 8200 to 66,000 |
| | | MTC 200 | | | | X | | | X | | | X | | OF | 30 to 300 | 630 to 8100 | 1 to 8 | 18 to 230 | 6500 to 100,000 | 158,732 to 2,777,821 | 3000 to 45,000 | 72000 to 1,260,000 | 870 to 1450 | 60 to 100 | -196 to 60 | 1.05 - 18 | 1234 to 10,862 | 920 to 8100 | 4800 to 21,000 | 10,850 to 37,600 |
| | | TG Series | | | | X | X | | X | X | | X | X | OI | 30 to 360 | 630 to 9712 | 1 to 10 | 18 to 275 | 6500 to 175,000 | 158,733 to 4,900,000 | 3000 to 80,000 | 72,000 to 2,200,000 | 1015 | 70 | -196 to 60 | 1.05 - 18 | 916 | 683 | 11,100 | 66,000 |
| ELLIOTT GROUP | 61, 137, Inside Back Cover | TH-85 | X | X | | | | | | | | | | | | | | | | 467,000 | | 211,830 | 50 / 5 | 3.4 / 0.34 | 760 / 649 | | 18,000 | 13,500 | | 6700 |
| | | TH-100 | X | X | | | | | | | | | | | | | | | | 718,000 | | 325,680 | 50 / 5 | 3.4 / 0.34 | 760 / 649 | | 25,000 | 19,000 | | 5800 |
| | | TH-120 | | X | | | | | | | | | | | | | | | | 1,105,000 | | 501,220 | 50 / 5 | 3.4 / 0.34 | 760 / 649 | | 40,000 | 30,000 | | 4700 |
| | | TH-140 | | X | | | | | | | | | | | | | | | | 1,700,000 | | 771,110 | 50 / 5 | 3.4 / 0.34 | 760 / 649 | | 60,000 | 45,000 | | 4000 |
| L.A. TURBINE | * | L Series | | | | | | | | | | | | | | 350 to 9400 | | | | | | | 3000 | 206 | -195 to 260 | | 1070 to 18,700 | 800 to 14,000 | | 15,000 to 105,000 |
| MAN ENERGY SOLUTIONS | 118, 119 | EN | X | | X | X | | X | | | | | | | | | | | | 1,320,000 | | 600,000 | 218 | 16 | 540 | 15 | 80,000 | 60,000 | | 20,000 |
| | | EH | X | X | | | | | | | | | | | | | | | | 1,320,000 | | 600,000 | 44 | 4 | 760 | 10 | 60,000 | 45,000 | | 24,000 |
| | | ER | | | | X | X | | X | X | | | | | | | | | | 1,230,000 | | 560,000 | 348 | 24 | to 500 | 5 | 67,000 | 50,000 | | 50,000 |
| SIEMENS ENERGY | * | STC-GT | | | | X | X | | X | X | | | X | OI | 5297 | 353,146 | 150 | 10,000 | | | | | 217.5 | 15 | 550 | 16 | 60,345 | 45,000 | 4400 | 25,000 |
| | | E Series | X | X | | | | | | | | | | | | | | | 58,000 to 780,000 | 20,000 to 1,609,000 | 26,500 to 354,000 | 45,000 to 730,000 | 35 to 217 | 3 to 15 | 760 | 15 | 10,000 to 65,000 | 7500 to 48,500 | | 3600 to 10,930 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free, OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| ABC COMPRESSORS | * | HA Series | X | X | | | X | | | | | | | | | OI/OF | 345 to 2075 | 650 to 5550 | 9.9 to 59.3 | 18.6 tp 159 | 2456 | 300 | 7800 | 35,000 | 4 | 100 to 830 | 75 to 610 | 370 | 700, 1000 |
| | | HG Series | X | X | | | X | | | | | | | | | OI/OF | 480 to 2880 | 900 to 7710 | 13.7 to 82.3 | 25.7 to 220 | 2456 | 200 | 11,000 | 49,000 | 4 | 135 to 1145 | 100 to 840 | 370 | 700, 1000 |
| | | HP-1 | X | X | | | X | | | | | | | | | OI/OF | 1745 to 10,470 | 2180 to 18,660 | 50 to 299 | 62.3 to 533 | 2456 | 200 | 22,000 | 98,000 | 4 | 340 to 2800 | 250 to 2060 | 370 | 460, 660 |
| | | HX-1 | X | X | | | X | | | | | | | | | OI/OF | 3535 to 21,195 | 4410 to 37,800 | 101 to 605 | 126 to 1080 | 2456 | 200 | 44,000 | 196,000 | 4 | 680 to 5645 | 500 to 4150 | 370 | 460, 660 |
| AERZENER MASCHINENFABRIK | * | VMY Series | X | X | | | | | | X | | | | | | OI | 63 to 1404 | 315 to 7020 | 2 to 40 | 9 to 199 | 232 to 508 | 16 to 35 | | | 16 to 25 | | | | 3,600 |
| | | VML | X | | | | | | | X | | | | | | OF | 177 | 8830 | 5 | 250 | 30 | 2 | | | 3 | | | | 3600 |
| | | VM | X | X | | | | | | X | | | | | | OF | 70 | 6000 | 2 | 170 | 145 | 10 | | | 6 | | | | 3600 |
| | | VRa | X | X | | | | | | X | | | | | | OF | 335 | 89,500 | 10 | 2535 | 754 | 52 | | | 12 | | | 2000 | 18,000 |
| | | Delta Hybrid | X | | | | | | | X | | | | | | OF | 71 | 5297 | 2 | 150 | 29 | 2 | | | 3 | | | 2100 | 15,000 |
| | | GM Series | X | X | | | | | X | | | | | | | OF | 18 to 5200 | 190 to 46,600 | 0.5 to 148 | 1320 | 15 | 1 to 64 | | | 2 | | | 150 to 1000 | 610 to 4800 |
| | | GR (2-lobe) | X | X | | | | | X | | | | | | | OF | 60 | 29,400 | 2 | 833 | 363 | 25 | | | 2 | | | 500 | 4000 |
| | | GQ (2-lobe) | X | X | | | | | X | | | | | | | OF | 880 | 58,900 | 25 | 1667 | 87 | 6 | | | 3 | | | 500 | 1200 |
| ARROW ENGINE CO. | * | VRU1 | X | | | X | | | | | | | | | | | | | | | 350 | 24.1 | 10,000 | | | 15 | | 250 | 600 |
| | | VR Series, Multi-Stage | X | X | | X | X | | | | | | | | | | | | | | 350 | 24.1 | 10,000 | | | 30 | | 250 | 900 |
| | | VRC Series | X | X | | X | X | | | | | | | | | | | | | | 6000 | 413.7 | 14,000 to 20,000 | | | 150 to 550 | | 900 | 1800 |
| ARIEL CORPORATION | Compressor Tab | JGM:P | X | X | | X | X | | | | | | | | | OI | | | | | 9000 | 621 | 7000 | 31,138 | | 170 | 127 | 750 | 1800 |
| | | JGN:Q | X | X | | X | X | | | | | | | | | OI | | | | | 9000 | 621 | 11,000 | 48,930 | | 280 | 209 | 750 | 1800 |
| | | JG:A | X | X | | X | X | | | | | | | | | OI/OF | | | | | 9000 | 621 | 11,000 | 48,930 | | 840 | 627 | 750 | 1800 |
| | | JGR:J | X | X | | X | X | | | | | | | | | OI/OF | | | | | 6100 | 421 | 23,000 | 102,309 | | 1860 | 1388 | 600 | 1800 |
| | | JGH:E | X | X | | X | X | | | | | | | | | OI | | | | | 10,000 | 690 | 32,000 | 142,343 | | 3210 | 2395 | 600 | 1500 |
| | | JGK:T | X | X | | X | X | | | | | | | | | OI/OF | | | | | 10,000 | 690 | 40,000 | 177,929 | | 3900 | 2909 | 600 | 1500 |
| | | KBK:T | X | X | | X | X | | | | | | | | | OI/OF | | | | | 10,000 | 690 | 50,000 | 222,411 | | 5520 | 4118 | 600 | 1500 |
| | | JGC:D:F | X | X | | X | X | | | | | | | | | OI/OF | | | | | 10,000 | 690 | 60,000 | 266,893 | | 6210 | 4633 | 500 | 1400 |
| | | KBC:D:F | X | X | | X | X | | | | | | | | | OI/OF | | | | | 10,000 | 690 | 69,000 | 306,927 | | 7200 | 5369 | 500 | 1400 |
| | | KBU:Z | X | X | | X | X | | | | | | | | | OI/OF | | | | | 10,000 | 690 | 80,000 | 355,858 | | 7800 | 5819 | 500 | 1200 |
| | | KBB:V | X | X | | X | X | | | | | | | | | OI/OF | | | | | 6700 | 462 | 100,000 | 444,822 | | 10,000 | 7460 | 360 | 900 |
| ATLAS COPCO GAS AND PROCESS | 70, 71 | GZ75 - VSD to GZ900 - VSD | | | | | | | | | | | | | | OF | | | | | 392 | 27 | | | | 1,207 | 900 | | |
| BAKER HUGHES | Inside Front Cover, 65 | HA | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 32,557 | 144,700 | | 2,840 | 2120 | | 1000 |
| | | HB | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 53,190 | 236,400 | | 7,397 | 5520 | | 800 |
| *continued* | | HD | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 72,562 | 322,500 | | 13,936 | 10,400 | | 700 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

†optimized for direct drive at 3000/3600 rpm, but higher speeds are allowable with VFD. Provided flow rates assume 3600 rpm drive speed.

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| BAKER HUGHES | Inside Front Cover, 65 | HE | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 150,748 | 670,000 | | 38,458 | 28,700 | | 800 |
| | | HF | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 256,500 | 1,140,000 | | 46,360 | 34,600 | | 514 |
| | | HG | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 444,960 | 1,980,000 | | 56,322 | 66,820 | | 514 |
| | | OA | X | X | | X | | | | | | | | | | OF/OI | | | | | | | 26,325 | 117,000 | | 580 | 435 | | 800 |
| | | PK | | X | | X | X | | | | | | | | | OI | | | | | | | 652,500 | 2,900,000 | | 96,480 | 72,000 | | 310 |
| | | PH | | X | | X | X | | | | | | | | | OI | | | | | | | 326,245 | 1,450,000 | | 48,226 | 36,000 | | 310 |
| | | SHM | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 80,565 | 358,065 | | 10,690 | 7,980 | | 1200 |
| | | SHMB | X | X | | X | X | | | | | | | | | OF/OI | | | | | | | 80,565 | 358,065 | | 7,130 | 5320 | | 1200 |
| BAUER KOMPRESSOREN GMBH, GERMANY | * | VERTICUS Series (air cooled) | | X | X | | | | | | | | | | | OI | 3 | 33 | 0.1 | 0.9 | 7250 | 500 | | | | 25 | 19 | 985 | 1485 |
| | | PE-VE series (air cooled) | | X | | | | | | | | | | | | OI | 3 | 21 | 0.1 | 0.6 | 5300 | 365 | | | | 20 | 15 | 985 | 1785 |
| | | VERTICUS Booster Series (air cooled) | | X | X | | | | | | | | | | | OI | 7 | 33 | 0.2 | 0.9 | 5300 | 365 | | | | 20 | 15 | 985 | 1485 |
| | | K22 to K28 Series (air cooled) | | X | X | | | | | | | | | | | OI | 30 | 240 | 0.9 | 6.8 | 7250 | 500 | | | | 148 | 110 | 985 | 1485 |
| | | IB23 Series (air- / water cooled) | | X | X | | | | | | | | | | | OI | 26 | 64 | 0.7 | 1.8 | 5300 | 365 | | | | 50 | 37 | 985 | 1485 |
| | | GIB23 Booster Series (air- / water cooled) | | X | X | | | | | | | | | | | OI | 47 | 203 | 1.3 | 5.7 | 6100 | 420 | | | | 50 | 37 | 985 | 1485 |
| | | I26 Series (water cooled) | | X | X | | | | | | | | | | | OI | 85 | 190 | 2.4 | 5.4 | 6100 | 420 | | | | 120 | 90 | 985 | 1485 |
| | | GIB26 Booster Series (water cooled) | | X | X | | | | | | | | | | | OI | 184 | 403 | 5.2 | 11.4 | 7500 | 520 | | | | 215 | 160 | 985 | 1485 |
| | | I52 Series (water cooled) | | X | X | | | | | | | | | | | OI | 160 | 240 | 4.8 | 6.6 | 6100 | 420 | | | | 215 | 160 | 985 | 1485 |
| | | GIB52 Booster Series (water cooled) | | X | X | | | | | | | | | | | OI | 371 | 805 | 10.5 | 22.8 | 7500 | 520 | | | | 422 | 315 | 985 | 1485 |
| | | GIB26-SP Series (water cooled) | | X | | | | | | | | | | | | OI | 367 | 530 | 10.4 | 15 | 7500 | 520 | | | | 422 | 315 | 1485 | 1485 |
| BLACKMER | * | HD Series | X | X | | | | | | | | | | | | OF | 2 to 52 | 8 to 125 | | | 335 to 600 | 23 to 41 | 2650 to 7000 | | 5 | 5 to 50 | | 350 | 825 |
| | | HD Series (Watercooled) | X | X | | | | | | | | | | | | OF | 4 to 52 | 8 to 125 | | | 335 to 985 | 23 to 68 | 2650 to 7000 | | 5 | 15 to 50 | | 350 | 825 |
| | | NG Series | X | X | | | | | | | | | | | | OF | 2 to 52 | 8 to 125 | | | 335 to 600 | 23 to 41 | 2650 to 7000 | | 5 | 5 to 50 | | 350 | 825 |
| | | NGH100 | X | X | | | | | | | | | | | | OF | 30 | 345 | | | 1500 | 103 | 7000 | | 5 | 100 | | 500 | 1800 |
| BORSIG ZM COMPRESSION GMBH | 77 | BX15 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 1450 | 100 | | | 5 | 8046 | 6000 | 800 | 1200 |
| | | BX22 | X | X | | | X | | | | | | | | | OF/OI | | | | | 14,500 | 1000 | | | 5 | 1340 | 1000 | 400 | 750 |
| *continued* | | BX32 | X | X | | | X | | | | | | | | | OF/OI | | | | | 14,500 | 1000 | | | 5 | 4023 | 3000 | 270 | 600 |

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| BORSIG ZM COMPRESSION GMBH | 77 | BX40 | X | X | | | X | | | | | | | | | OF/OI | | | | | 14,500 | 1000 | | | 5 | 9387 | 7000 | 200 | 450 |
| | | BX45 | X | X | | | X | | | | | | | | | OF/OI | | | | | 14,500 | 1000 | | | 5 | 16,092 | 12,000 | 150 | 400 |
| | | BX50 | X | X | | | X | | | | | | | | | OF/OI | | | | | 14,500 | 1000 | | | 5 | 28,161 | 21,000 | 110 | 350 |
| | | PV90 Vertical Series | X | X | | | | | | | | | | | | OF | | | | | 10,150 | 700 | | | 5 | 335 | 250 | 450 | 1500 |
| | | PV110 Vertical Series | X | X | | | | | | | | | | | | OF | | | | | 10,150 | 700 | | | 5 | 600 | 450 | 360 | 1200 |
| | | PV140 Vertical Series | X | X | | | | | | | | | | | | OF | | | | | 10,150 | 700 | | | 5 | 940 | 700 | 295 | 900 |
| | | PV180 Vertical Series | X | X | | | | | | | | | | | | OF | | | | | 10,150 | 700 | | | 5 | 2010 | 1500 | 230 | 750 |
| | | PV220 Vertical Series | X | X | | | | | | | | | | | | OF | | | | | 10,150 | 700 | | | 5 | 3350 | 2500 | 180 | 600 |
| BURCKHARDT COMPRESSION AG *PROCESS GAS COMPRESSOR API 618* | Back Cover | BY | X | X | | | X | | | | | | | | | OF/OI | | 2300 | | 60 | 14,500 | 1000 | 22,500 | 100,000 | 4 | 1000 | 800 | 425 | 850 |
| | | CY | X | X | | | | | | | | | | | | OF/OI | | 2300 | | 60 | 14,500 | 1000 | 22,500 | 100,000 | 4 | 1000 | 800 | 425 | 850 |
| | | BF | X | X | | | | | | | | | | | | OF/OI | | | | | 14,500 | 1000 | 32,600 | 145,000 | 4 | 3000 | 2200 | 300 | 600 |
| | | BS | X | X | | | X | | | | | | | | | OF/OI | | 7100 | | 200 | 14,500 | 1000 | 45,000 | 200,000 | 4 | 3200 | 2400 | 300 | 600 |
| | | CS | X | X | | | | | | | | | | | | OF/OI | | 4700 | | 130 | 14,500 | 1000 | 45,000 | 200,000 | 4 | 3200 | 2400 | 300 | 600 |
| | | BX | X | X | | | X | | | | | | | | | OF/OI | | 10,600 | | 300 | 14,500 | 1000 | 78,500 | 350,000 | 4 | 7200 | 5400 | 260 | 520 |
| | | BA | X | X | | | X | | | | | | | | | OF/OI | | 15,900 | | 450 | 14,500 | 1000 | 124,000 | 550,000 | 4 | 12,700 | 9,500 | 250 | 500 |
| | | BC | X | X | | | X | | | | | | | | | OF/OI | | 19,400 | | 550 | 14,500 | 1000 | 202,000 | 900,000 | 4 | 21,700 | 16,000 | 300 | 450 |
| | | BE | X | X | | | X | | | | | | | | | OI | | 23,000 | | 650 | 14,500 | 1000 | 382,000 | 1,700,000 | 4 | 42,100 | 31,000 | 300 | 429 |
| | | D/M 6.5 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 14,612 | 65,000 | 4 | 805 | 600 | 300 | 520 |
| | | D/M 10 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 22,480 | 100,000 | 4 | 1475 | 1100 | 300 | 520 |
| | | D/M 12 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 26,977 | 120,000 | 4 | 1743 | 1300 | 300 | 512 |
| | | D/M 16 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 35,969 | 160,000 | 4 | 2145 | 1600 | 300 | 450 |
| | | D/M 20 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 44,992 | 200,000 | 4 | 2,682 | 2,000 | 300 | 450 |
| | | D/M 25 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 56,202 | 250,000 | 4 | 4,291 | 3,200 | 300 | 450 |
| | | D/M 32 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 71,938 | 320,000 | 4 | 7242 | 5400 | 300 | 420 |
| | | D/M 45 | X | X | | | X | | | | | | | | | OF/OI | | | | | 2900 | 200 | 101,164 | 450,000 | 4 | 7445 | 5550 | 300 | 400 |
| | | D/M 80 | X | X | | | X | | | | | | | | | OI | | | | | 2900 | 200 | 179,847 | 800,000 | 4 | 11,165 | 8325 | 300 | 360 |
| | | D/M VL | X | X | | | X | | | | | | | | | OI | | | | | 2900 | 200 | 224,808 | 1,000,000 | 4 | 13,955 | 10,406 | 300 | 360 |
| | | D/M HE | X | X | | | X | | | | | | | | | OI | | | | | 2900 | 200 | 281,011 | 1,250,000 | 4 | 21,725 | 16,200 | 300 | 333 |
| *LABY COMPRESSORS* | | 1D130 | X | | | | | | | | | | | | | OF | | 300 | | 10 | 4640 | 300 | | | 4 | 100 | 76 | 450 | 750 |
| | | 1D150 | X | | | | | | | | | | | | | OF | | 300 | | 10 | 730 | 50 | | | 4 | 160 | 120 | 360 | 600 |
| | | 2D100 | X | X | | | | | | | | | | | | OF | | 300 | | 10 | 730 | 50 | | | 4 | 160 | 120 | 600 | 1000 |
| | | 2D140 | X | X | | | | | | | | | | | | OF | | 400 | | 10 | 290 | 20 | | | 4 | 233 | 174 | 450 | 750 |
| *continued* | | 2D160 | X | X | | | | | | | | | | | | OF | | 700 | | 20 | 1160 | 80 | | | 4 | 407 | 304 | 450 | 750 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **BURCKHARDT COMPRESSION AG** *LABY COMPRESSORS* | Back Cover | **2D200** | X | X | | | | | | | | | | | | OF | | 1000 | | 30 | 750 | 52 | | | 4 | 643 | 480 | 360 | 600 |
| | | **2DL200** | X | X | | | | | | | | | | | | OF | | 2200 | | 60 | 290 | 20 | | | 4 | 643 | 480 | 360 | 600 |
| | | **2D205** | X | X | | | | | | | | | | | | OF | | 800 | | 20 | 3630 | 250 | | | 4 | 938 | 700 | 360 | 600 |
| | | **2D250** | X | X | | | | | | | | | | | | OF | | 2500 | | 70 | 1740 | 120 | | | 4 | 2279 | 1700 | 312 | 520 |
| | | **2DL250** | X | X | | | | | | | | | | | | OF | | 2700 | | 80 | 360 | 25 | | | 4 | 2370 | 1770 | 312 | 520 |
| | | **3D200** | X | X | | | | | | | | | | | | OF | | 1900 | | 50 | 1020 | 70 | | | 4 | 657 | 490 | 360 | 600 |
| | | **4D225** | X | X | | | | | | | | | | | | OF | | 2700 | | 80 | 810 | 56 | | | 4 | 973 | 726 | 360 | 600 |
| | | **4D250** | X | X | | | | | | | | | | | | OF | | 4700 | | 130 | 3050 | 210 | | | 4 | 1374 | 1025 | 312 | 520 |
| | | **4D300** | X | X | | | | | | | | | | | | OF | | 4000 | | 110 | 1280 | 88 | | | 4 | 2055 | 1533 | 270 | 450 |
| | | **4D375** | X | X | | | | | | | | | | | | OF | | 4800 | | 140 | 730 | 50 | | | 4 | 2755 | 2055 | 228 | 380 |
| | | **6D375** | X | X | | | | | | | | | | | | OF | | 5400 | | 150 | 870 | 60 | | | 4 | 2755 | 2055 | 228 | 380 |
| | | **2K70** | X | X | | | | | | | | | | | | OF | | 300 | | 10 | 261 | 18 | | | 4 | 95 | 71 | 1080 | 1800 |
| | | **2K90** | X | X | | | | | | | | | | | | OF | | 300 | | 10 | 610 | 42 | | | 4 | 154 | 115 | 600 | 1000 |
| | | **2KL90** | X | X | | | | | | | | | | | | OF | | 300 | | 10 | 610 | 16 | | | 4 | 154 | 115 | 600 | 1000 |
| | | **2K105** | X | X | | | | | | | | | | | | OF | | 500 | | 10 | 1160 | 80 | | | 4 | 252 | 188 | 600 | 1000 |
| | | **2K140** | X | X | | | | | | | | | | | | OF | | 800 | | 20 | 730 | 50 | | | 4 | 406 | 303 | 450 | 750 |
| | | **2KL140** | X | X | | | | | | | | | | | | OF | | 800 | | 20 | 232 | 16 | | | 4 | 406 | 303 | 450 | 750 |
| | | **2K158** | X | X | | | | | | | | | | | | OF | | 700 | | 20 | 460 | 32 | | | 4 | 665 | 485 | 450 | 750 |
| | | **2K160** | X | X | | | | | | | | | | | | OF | | 1200 | | 30 | 2180 | 150 | | | 4 | 665 | 485 | 450 | 750 |
| | | **2K250** | X | X | | | | | | | | | | | | OF | | 800 | | 20 | 1670 | 115 | | | 4 | 2226 | 1660 | 300 | 500 |
| | | **3K140** | X | X | | | | | | | | | | | | OF | | 700 | | 20 | 580 | 90 | | | 4 | 665 | 485 | 450 | 750 |
| | | **3K160** | X | X | | | | | | | | | | | | OF | | 1800 | | 950 | 640 | 44 | | | 4 | 665 | 485 | 450 | 750 |
| | | **4K165** | X | X | | | | | | | | | | | | OF | | 2000 | | 60 | 960 | 66 | | | 4 | 1397 | 1042 | 450 | 750 |
| *HYPER COMPRESSORS* | | **H** | X | X | | | X | | | | | | | | | OI | | 14,200 | | 400 | 50,760 | 3500 | | | 4 | 10,700 | 8000 | 154 | 257 |
| | | **F** | X | X | | | X | | | | | | | | | OI | | 42,600 | | 1210 | 50,760 | 3500 | | | 4 | 26,850 | 20,000 | 139 | 231 |
| | | **K** | X | X | | | X | | | | | | | | | OI | | 85,100 | | 2410 | 50,760 | 3500 | | | 4 | 51,000 | 38,000 | 129 | 215 |
| *LABY-GI COMPRESSORS* | | **LP250** | X | X | | | X | | | | | | | | | OF/OI | | | | 110 | 14500 | 1000 | 72 | 322 | 4 | 5320 | 4000 | 312 | 520 |
| *STANDARD HIGH-PRESSURE COMPRESSORS* | | **CB** | | X | | | | | | | | | | | | OI | | | | 0.8 | 2900 | 200 | | | 5 | 24 | 18 | | 1500 |
| | | **CC** | | X | | | | | | | | | | | | OI | | | | 1.7 | 5100 | 350 | | | 5 | 60 | 45 | | 1365 |
| | | **CU** | | X | | | | | | | | | | | | OI | | | | 13.3 | 5800 | 400 | | | 5 | 150 | 110 | | 1230 |
| | | **CT** | | X | | | | | | | | | | | | OI | | | | 25 | 5100 | 350 | | | 5 | 270 | 200 | | 1045 |
| *MARINE HIGH-PRESSURE COMPRESSORS* | | **MHP-A-310** | | X | | | | | | | | | | | | OI | | | | 25 | 4500 | 310 | | | 5 | 225 | 168 | | 1180 |
| *DIAPHRAGM COMPRESSORS* | | **MD2.5** | | X | | | X | X | | | | | | | | OF | | | | 0.6 | 8000 | 550 | 6 | 25 | 5 | 16 | 12 | | |
| *continued* | | **MD5** | | X | | | X | X | | | | | | | | OF | | | | 1.6 | 8000 | 550 | 11 | 50 | 5 | 39 | 29 | | |

# RECIPROCATING AND ROTARY COMPRESSORS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Reciprocating: Single Stage | Multiple Stages | Integral Engine Driven | Separable | Balanced/Opposed | Diaphragm | Rotary: Straight Lobe | Helical Lobe (Screw) | Single Screw | Sliding Vane | Liquid-Ring | Trochoidal | Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | acfm max | m³/min min | m³/min max | MAWP Maximum Allowable Working Pressure psig | bar | MARL Maximum Allowable Rod Load lb | Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | kW | Speed Range (rpm) min | max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| BURCKHARDT COMPRESSION AG *DIAPHRAGM COMPRESSORS* | Back cover | MD6.5 | | X | | | X | X | | | | | | | | OF | | | | 11.5 | 8000 | 550 | 15 | 65 | 5 | 225 | 168 | | |
| | | MD10 | | X | | | X | X | | | | | | | | OF | | | | 19 | 8000 | 550 | 22 | 100 | 5 | 350 | 261 | | |
| | | MD12 | | X | | | X | X | | | | | | | | OF | | | | 23 | 8000 | 550 | 27 | 120 | 5 | 424 | 316 | | |
| Clauger-Technofrigo S.p.A. | * | GEA Grasso LT & M Series | X | | | | | | | X | | | | | | OI | 164 | 8,150 | 4.6 | 229.3 | 913.0 | 63 | | | 20 | 5,623 | 4193 | 1400 | 3600 |
| | | Howden WRV | X | | | | | | | X | | | | | | OI | 328 | 17,068 | 9.2 | 483.3 | 653.0 | 45 | | | 22 | 11,400 | 8500 | 750 | 3600 |
| COMOTI | 99 | ECS 2.5/10 | | | | | | | | X | | | | | | OI | | | 1 | 3 | | 10 | | | 10 | | 30 | 1450 | 3600 |
| | | ECS 10/10 | | | | | | | | X | | | | | | OI | | | 4 | 8 | | 10 | | | 10 | | 37 | 1050 | 2000 |
| | | ECS 15/10 | | | | | | | | X | | | | | | OI | | | 4 | 11 | | 10 | | | 10 | | 75 | 1050 | 3200 |
| | | ECS 20/10 | | | | | | | | X | | | | | | OI | | | 4 | 13 | | 10 | | | 10 | | 90 | 1050 | 3700 |
| | | ECS 1/10 | | | | | | | | X | | | | | | OI | | | 0 | 1 | | 10 | | | 10 | | 15 | 1450 | 3000 |
| | | ECS 2/10 | | | | | | | | X | | | | | | OI | | | 1 | 1 | | 10 | | | 10 | | 22 | 1450 | 3000 |
| | | ECS 25/10 | | | | | | | | X | | | | | | OI | | | 7 | 17 | | 10 | | | 10 | | 110 | 1000 | 1700 |
| | | ECS 30/10 | | | | | | | | X | | | | | | OI | | | 7 | 23 | | 10 | | | 10 | | 132 | 1000 | 2100 |
| | | ECS 60/10 | | | | | | | | X | | | | | | OI | | | 13 | 42 | | 10 | | | 10 | | 250 | 600 | 1700 |
| | | ECS 75/10 | | | | | | | | X | | | | | | OL | | | 42 | 52 | | 6 | | | 6 | | 315 | 2000 | 3000 |
| | | ECS 2.5/16 | | | | | | | | X | | | | | | OI | | | 1 | 3 | | 16 | | | 15 | | 40 | 1450 | 3600 |
| | | ECS 20/25 | | | | | | | | X | | | | | | OI | | | 2 | 7 | | 25 | | | 15 | | 75 | 1700 | 5100 |
| | | ECS 15/30 | | | | | | | | X | | | | | | OI | | | 3 | 10 | | 30 | | | 15 | | 132 | 1200 | 3600 |
| | | ECS 20/30 | | | | | | | | X | | | | | | OI | | | 2 | 6 | | 30 | | | 15 | | 132 | 1600 | 4800 |
| | | ECS 30/30 | | | | | | | | X | | | | | | OI | | | 2 | 5 | | 30 | | | 15 | | 160 | 1600 | 4600 |
| | | ECS 35/30 | | | | | | | | X | | | | | | OI | | | 17 | 25 | | 30 | | | 15 | | 250 | 2000 | 3000 |
| | | ECS 60/30 | | | | | | | | X | | X | | | | OI | | | 27 | 42 | | 30 | | | 15 | | 450 | 2000 | 3000 |
| | | ECS 80/30 | | | | | | | | X | | | | | | OI | | | 40 | 56 | | 30 | | | 15 | | 500 | 2000 | 3000 |
| | | ECS 5/40 B | | | | | | | | X | | | | | | OL | | | 2 | 7 | | 40 | | | 14 | | 90 | 2000 | 3000 |
| | | CU64GM | | | | | | | | X | | | | | | OL | | | 1 | 4 | | 26 | | | 15 | | 52 | 2296 | 9182 |
| | | CU 90GM | | | | | | | | X | | | | | | OL | | | 2 | 8 | | 26 | | | 15 | | 98 | 1632 | 6529 |
| | | CU 128G | | | | | | | | X | | | | | | OL | | | 4 | 16 | | 26 | | | 15 | | 193 | 1148 | 4591 |
| | | CU 64 GM | | | | | | | | X | | | | | | OL | | | 1 | 4 | | 45 | | | 15 | | 100 | 2296 | 9182 |
| | | CU 90 GM | | | | | | | | X | | | | | | OL | | | 2 | 8 | | 45 | | | 15 | | 175 | 1632 | 6529 |
| | | CU 200 | | | | | | | | X | | | | | | OL | | | 7 | 35 | | 26 | | | 15 | | 550 | 735 | 3673 |
| CORKEN, INC. | * | HG600/THG600 Horizontal Series | X | X | | | X | | | | | | | | | OI/OF | | | | | 1650 | 20.7 | 7500 | 33,361 | 5:1 | 75 | 56 | 400 | 1200 |
| | | Model 91 Vertical Series | X | | | | | | | | | | | | | OF | | | | | 335 | 23.1 | 3600 | 1633 | 5:1 | 8 | 6 | 400 | 825 |
| *continued* | | Model 291 Vertical Series | X | | | | | | | | | | | | | OF | | | | | 335 | 23 | 3600 | 1633 | 5:1 | 15 | 11 | 400 | 825 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| CORKEN, INC. | * | Model 491 Vertical Series | X | | | | | | | | | | | | | OF | | | | | 335 | 23 | 4000 | 1814 | 5:1 | 15 | 11 | 400 | 825 |
| | | Model 491-3 Vertical Series | X | | | | | | | | | | | | | OF | | | | | 600 | 41.4 | 4000 | 1814 | 5:1 | 15 | 11 | 400 | 825 |
| | | Model 691 Vertical Series | X | | | | | | | | | | | | | OF | | | | | 335 | 23 | 7000 | 3175 | 5:1 | 35 | 26 | 400 | 825 |
| | | Model 691-4 Vertical Series | X | | | | | | | | | | | | | OF | | | | | 600 | 41 | 7000 | 3175 | 5:1 | 35 | 26 | 400 | 825 |
| | | FD891 Vertical | X | | | | | | | | | | | | | | | | | | 450 | 31 | 7000 | 3175 | 5:1 | 45 | 34 | 400 | 825 |
| | | FT891 Vertical | X | | | | | | | | | | | | | OF | | | | | 450 | 31 | 7000 | 3175 | 5:1 | 15 | 11 | 400 | 825 |
| | | Model 151 Vertical Series | | X | | | | | | | | | | | | OF | | | | | 1200 | 83 | 3600 | 1633 | 5:1 | 15 | 11 | 400 | 825 |
| | | Model 191 Vertical Series | | X | | | | | | | | | | | | OF | | | | | 600 | 41 | 3600 | 1633 | 5:1 | 15 | 11 | 400 | 825 |
| | | Model 351 Vertical Series | | X | | | | | | | | | | | | OF | | | | | 1200 | 83 | 4000 | 1814 | 5:1 | 15 | 11 | 400 | 825 |
| | | Model 391 Vertical Series | | X | | | | | | | | | | | | OF | | | | | 600 | 41 | 4000 | 1814 | 5:1 | 15 | 11 | 400 | 825 |
| | | Model 551 Vertical Series | | X | | | | | | | | | | | | OF | | | | | 1000 | 69 | 7000 | 3175 | 5:1 | 35 | 26 | 400 | 825 |
| | | Model 591 Vertical Series | | X | | | | | | | | | | | | OF | | | | | 600 | 41 | 7000 | 3175 | 5:1 | 35 | 26 | 400 | 825 |
| | | FD791 Vertical | | X | | | | | | | | | | | | | | | | | 600 | 41 | 7000 | 3175 | 5:1 | 45 | 34 | 400 | 825 |
| | | FT791 Vertical | | X | | | | | | | | | | | | OF | | | | | 600 | 41 | 7000 | 3175 | 5:1 | 45 | 34 | 400 | 825 |
| COMPACT COMPRESSION | * | HCG Series | X | | | X | | | | | | | | | | OF | 0 | 19 | 0 | 0.4 to 1.5 | 1500 | 103.4 | | | 20 | 15, 50 | 11, 37 | | |
| | | WGC Series | X | X | | X | | | | | | | | | | OF | 0 | 34 to 63 | 0 | 0.96 to 2.8 | 1500 | 103.4 | 4000 | 17,793 | 20 | 20, 30 | 22 | 30 | 225 |
| COOPER MACHINERY SERVICES | * | AJAX Series | X | X | X | | | | | | | | | | | OI | | | | | 5500 | 379 | 30,000 | 133,446 | | 148 to 300 | 110 to 224 | 260, 300 | 440, 455 |
| | | M301 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 6,000 | 26,689 | | 60 | 45 | 900 | 1800 |
| | | M302 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 6,000 | 26,689 | | 120 | 89 | 900 | 1800 |
| | | H301 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 10,000 | 44,482 | | 100 | 75 | 900 | 1800 |
| | | H302 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 10,000 | 44,482 | | 200 | 149 | 900 | 1800 |
| | | H304 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 10,000 | 44,482 | | 400 | 298 | 900 | 1800 |
| | | A351 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 14,500 | 64,499 | | 200 | 149 | 900 | 1800 |
| | | A352 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 14,500 | 64,499 | | 400 | 298 | 900 | 1800 |
| | | A354 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 6000 | 414 | 14,500 | 64,499 | | 800 | 596 | 900 | 1800 |
| | | CFA 32 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 5808 | 400 | 13,500 | 60,051 | | 290 | 216 | 600 | 1800 |
| | | CFA 34 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 5808 | 400 | 13,500 | 60,051 | | 580 | 433 | 600 | 1800 |
| | | CFH62 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,000 | 689 | 65,000 | 289,134 | | 1360 | 1014 | 250 | 1200 |
| | | CFH64 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,000 | 689 | 65,000 | 289,134 | | 2720 | 2028 | 250 | 1200 |
| | | CFR52 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 4700 | 324 | 55,000 | 244,652 | | 1700 | 1268 | 250 | 1500 |
| continued | | CFR54 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 4700 | 324 | 55,000 | 244,652 | | 3400 | 2535 | 250 | 1500 |

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Reciprocating: Single Stage | Reciprocating: Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| COOPER MACHINERY SERVICES | * | RAM52 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 2200 | 152 | 44,000 | 195,722 | | 1188 | 886 | 250 | 1500 |
| | | RAM54 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 2200 | 152 | 44,000 | 195,722 | | 2375 | 1771 | 250 | 1500 |
| | | MH62 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,000 | 689 | 56,000 | 249,100 | | 1800 | 1342 | 250 | 1200 |
| | | MH64 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,000 | 689 | 56,000 | 249,100 | | 3600 | 2685 | 250 | 1200 |
| | | MH66 | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,000 | 689 | 56,000 | 249,100 | | 5400 | 4027 | 250 | 1200 |
| | | WH Series | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,000 | 689 | 70,000 | 311,376 | | 1700 to 5400 | 1268 to 4027 | 250 | 1000, 1200 |
| | | WG Series | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,000 | 689 | 90,000 | 400,340 | | 2500 to 9000 | 2237 to 6711 | 200 | 1000, 1200 |
| | | Cooper-Bessemer Series | X | X | X | | | | | | | | | | | OF, OI | | | | | 15,000 | 1034 | 150,000 | 667,230 | | 3500 to 9300 | 2600 to 6900 | 264 | 330 |
| FLSMIDTH INC. FUL-VANE | * | CC Series, Single Stage | X | | | | | | | | | X | | | | OI | 45 to 943 | 121 to 2176 | 1.3 to 26.7 | 3.4 to 61.6 | 150 | 10.3 | | | 6 | 50 to 500 | 37 to 375 | 325 to 725 | 750 to 1940 |
| | | C Series, Single Stage | X | | | | | | | | | X | | | | OI | 1102 to 1943 | 2203 to 3368 | 31.2 to 55 | 62.4 to 95.4 | 150 | 10.3 | | | 5, 6 | 650, 500 | 375, 485 | 300, 325 | 520, 650 |
| | | CC Series, Multi-Stage | | X | | | | | | | | X | | | | OI | 106 to 943 | 271 to 2176 | 3 to 26.7 | 7.7 to 61.6 | 300 | 20 | | | 5 | 75 to 500 | 55 to 375 | 325 to 725 | 750 to 1850 |
| | | C350-350H | | X | | | | | | | | X | | | | OI | 1102 | 2203 | 31.2 | 62.4 | 300 | 20 | | | 5 | 500 | 375 | 325 | 650 |
| | | CB Series | X | | | | | | | | | X | | | | OI | 38 to315 | 97 to 726 | 1.1 to 8.9 | 2.8 to 20.6 | 300 | 20 | | | 7 | 75 to 500 | 55 to 375 | 325 to 725 | 750 to 1850 |
| | | B350 | X | | | | | | | | | X | | | | OI | 368 | 736 | 10.4 | 20.8 | 300 | 20 | | | 7 | 500 | 375 | 325 | 650 |
| | | V Series, Single Stage | X | | | | | | | | | X | | | | OI | 106 to 1102 | 271 to 2203 | 3 to 31.2 | 7.7 to 62.4 | -13.2 | -0.9 | | | | 75 to 500 | 55 to 375 | 325 to 725 | 650 to 1500 |
| | | V Series, Multi-Stage | | X | | | | | | | | X | | | | OI | 106 to 1102 | 271 to 2203 | 3 to 31.2 | 7.7 to 62.4 | -14.6 | -1 | | | | 75 to 500 | 55 to 375 | 325 to 725 | 650 to 1500 |
| FORNOVOGAS SRL | * | SA200 | X | X | | X | | | | | | | | | | OF | | | | | | 308 | | 15,000 | | | 55 | 500 | 1800 |
| | | DA300 | X | X | | X | | | | | | | | | | OF | | | | | | 375 | | 50,000 | | | 400 | 500 | 1500 |
| | | DA500 | X | X | | X | X | | | | | | | | | OF/OI | | | | | | 275 | | 125,000 | | | 1500 | 650 | 1800 |
| GARO S.P.A. | * | AM, ASM, AB | X | X | | | | | | | | | X | | | OF | 140 | 3000 | 4 | 83 | 190 | 13 | | | 5 | 1340 | 1000 | 580 | 3600 |
| HAUG SAUER KOMPRESSOREN AG | * | HAUG.Pluto | X | | X | | X | | | | | | | | | OF | | 30 | | 1 | 870 | 60 | | | | 3 | 2.2 | 970 | 1740 |
| | | HAUG.Mercure | X | X | X | | X | | | | | | | | | OF | | 41 | | 1 | 1450 | 100 | | | | 5.5 | 4 | 970 | 1470 |
| | | HAUG.Neptune | X | X | | | X | | | | | | | | | OF | | 59 | | 2 | 1450 | 100 | | | | 10 | 7.5 | 970 | 1470 |
| | | HAUG.Sirius | X | X | | | X | | | | | | | | | OF | | 590 | | 17 | 1450 | 100 | | | | 41 | 30 | 970 | 1470 |
| | | HAUG.Sirius NanoLoc | X | X | | | X | | | | | | | | | OF | | 35 | | 1 | 6527 | 450 | | | | 41 | 30 | 970 | 1470 |
| | | HAUG.Titan | X | X | | | X | | | | | | | | | OF | | 1177 | | 33 | 1450 | 100 | | | | 150 | 110 | 450 | 900 |
| | | HAUG.Cygnus | X | X | | | X | | | | | | | | | OF | | 7 | | 1 | 435 | 30 | | | | 3 | 2.2 | 1450 | 3400 |
| | | HAUG.Taurus | X | X | | | X | | | | | | | | | OF | | 27 | | 1 | 870 | 60 | | | | 15 | 11 | 970 | 1470 |
| | | HAUG.Orion | X | X | | | X | | | | | | | | | OF | | 94 | | 3 | 870 | 60 | | | | 41 | 30 | 970 | 1470 |
| HOFER | 92, 93 | MK | X | X | | X | X | X | | | | | | | | OF | 0.35 | 1200 | 0.01 | 35.00 | 72.000 | 5000 | 31,500 | 140,000 | 8 | 335 | 250 | 250 | 720 |
| | | TKH | X | X | | X | X | | | | | | | | | OF | 0.35 | 15.00 | 0.01 | 0.40 | 72.000 | 5000 | 78,500 | 350,000 | 8 | 270 | 200 | 8 | 40 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Reciprocating: Single Stage | Reciprocating: Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| HOWDEN | 104, 105 | Howden Thomassen P Series | X | X | | | X | | | | | | | | | OF | 4 | 20,000 | 0.1 | 568 | 5100 | 350 | 49,500 | 220,000 | 4 | 3400 | 2500 | 200 | 1000 |
| | | Howden Burton Corblin D Series | X | X | | | | X | | | | | | | | OF | 0 | 357 | 0 | 10 | 43,500 | 3000 | 49,500 | 220,000 | 12 | 1766 | 1300 | 200 | 750 |
| | | Howden HPD Hybrid Series | | X | | | X | X | | | | | | | | OF | 35 | 2925 | 1 | 83 | 14,500 | 1000 | 49,500 | 220,000 | 3 | 2649 | 1950 | 200 | 500 |
| | | Howden Thomassen C-7 | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 29,225 | 130,000 | 5 | 1090 | 800 | 300 | 600 |
| | | Howden Thomassen C-12 | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 41,600 | 185,000 | 5 | 3130 | 2300 | 300 | 600 |
| | | Howden Thomassen C-25 | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 67,500 | 300,000 | 5 | 7620 | 5600 | 300 | 600 |
| | | Howden Thomassen C-35 | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 123,700 | 550,000 | 5 | 14,000 | 10,300 | 250 | 500 |
| | | Howden Thomassen C-45 | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 185,500 | 825,000 | 5 | 20,950 | 15,400 | 250 | 500 |
| | | Howden Thomassen C-85 | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 281,000 | 1,250,000 | 5 | 33,720 | 24,800 | 190 | 375 |
| | | Howden Thomassen C-85H | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 404,600 | 1,800,000 | 5 | 44,870 | 33,000 | 300 | 375 |
| | | Howden Thomassen CHS | X | X | | | X | | | | | | | | | OF/OI | | | | | 8700 | 600 | 55,000 | 245,000 | 5 | 6254 | 4600 | 500 | 1200 |
| | | XRV127 | | | | | | | | X | | | | | | OI | | 342 | | 9.9 | 305 | 21 | | | 10 | 200 | 150 | 1800 | 5000 |
| | | XRV163 | | | | | | | | X | | | | | | OI | | 504 | | 14.3 | 305 | 21 | | | 10 | 350 | 260 | 1800 | 3600 |
| | | XRV204 | | | | | | | | X | | | | | | OI | | 976 | | 27.6 | 305 | 21 | | | 10 | 600 | 450 | 1800 | 3600 |
| | | M127 | | | | | | | | X | | | | | | OI | | 410 | | 11.8 | 305 | 21 | | | 10 | 200 | 150 | 1800 | 5000 |
| | | GTV 228 | | | | | | | | X | | | | | | OI | | 683 | | 19.3 | 870 | 60 | | | 3 | 4690 | 3500 | 1250 | 4000 |
| | | WRV(H)163 | | | | | | | | X | | | | | | OI | | 601 | | 17.3 | 350 | 24 | | | 12 | 470 | 350 | 1500 | 4500 |
| | | WRV(H)204 | | | | | | | | X | | | | | | OI | | 718 | | 33.6 | 350 | 24 | | | 12 | 1028 | 766 | 1500 | 4500 |
| | | WRVi255 | | | | | | | | X | | | | | | OI | | 1905 | | 54 | 350 | 24 | | | 12 | 1542 | 1150 | 1500 | 3600 |
| | | WRV255 | | | | | | | | X | | | | | | OI | | 2258 | | 63.9 | 200 | 13.8 | | | 8 | 1542 | 1150 | 1500 | 3600 |
| | | WRVi321 | | | | | | | | X | | | | | | OI | | 3811 | | 108 | 350 | 24 | | | 12 | 2741 | 2044 | 1500 | 3600 |
| | | WRV321 | | | | | | | | X | | | | | | OI | | 4517 | | 128 | 200 | 13.8 | | | 8 | 2741 | 2044 | 1500 | 3600 |
| | | WRVi365 | | | | | | | | X | | | | | | OI | | 5580 | | 158.1 | 350 | 24 | | | 12 | 5814 | 4335 | 1500 | 3600 |
| | | WRVT510 | | | | | | | | X | | | | | | OI | | 13,430 | | 380 | 350 | 24 | | | 12 | 6700 | 5000 | 700 | 2600 |
| | | WRVTi580 | | | | | | | | X | | | | | | OI | | 17,070 | | 483 | 350 | 24 | | | 12 | 11,265 | 8400 | 750 | 2600 |
| | | WRVTF510 | | | | | | | | X | | | | | | OI | | 12,400 | | 350 | 203 | 14 | | | 12 | 6700 | 5000 | 700 | 3000 |
| | | H127 | | | | | | | | X | | | | | | OF | | 882 | | 25 | 126 | 8.7 | | | 4 | 286 | 213 | 7500 | 15,000 |
| | | HP204 | | | | | | | | X | | | | | | OF | | 1490 | | 42 | 201 | 13.8 | | | 4 | 800 | 600 | 4750 | 9500 |
| | | H204 | | | | | | | | X | | | | | | OF | | 2236 | | 63 | 126 | 8.7 | | | 4 | 800 | 600 | 4750 | 9500 |
| *continued* | | HP255 | | | | | | | | X | | | | | | OF | | 2300 | | 65 | 201 | 13.8 | | | 4 | 1250 | 935 | 4000 | 7500 |

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range $m^3$/min min | Inlet Flow Range $m^3$/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| HOWDEN | 104, 105 | H255 | | | | | | | | X | | | | | | OF | | 3450 | | 98 | 126 | 8.7 | | | 4 | 1250 | 935 | 4000 | 7500 |
| | | HP408 | | | | | | | | X | | | | | | OF | | 5899 | | 167 | 201 | 13.8 | | | 4 | 3050 | 2275 | 2300 | 4700 |
| | | H408 | | | | | | | | X | | | | | | OF | | 8845 | | 250 | 126 | 8.7 | | | 4 | 3050 | 2275 | 2300 | 4700 |
| | | HP510 | | | | | | | | X | | | | | | OF | | 8266 | | 234 | 200 | 15 | | | 4 | 4000 | 2985 | 2000 | 3750 |
| | | H510 | | | | | | | | X | | | | | | OF | | 12,397 | | 351 | 130 | 10 | | | 4 | 4000 | 2985 | 2000 | 3750 |
| | | ČKD JSKB | X | | | | X | | | | | | | | | OF | 113 | 1561 | 3 | 44 | 435 | 30 | 88,185 | 392,266 | 6 | 362 | 270 | 300 | 1000 |
| | | ČKD DSKB | | X | | | X | | | | | | | | | OF | 99 | 1137 | 3 | 32 | 435 | 30 | 88,185 | 392,266 | 5 | 367 | 274 | 300 | 1000 |
| | | ČKD TSKB | | X | | | X | | | | | | | | | OF | 60 | 1737 | 2 | 49 | 2391 | 165 | 88,185 | 392,266 | 4 | 624 | 465 | 300 | 1000 |
| | | ČKD JSKM | X | | | | X | | | | | | | | | OI | 441 | 1811 | 13 | 51 | 652 | 45 | 88,185 | 392,266 | 5 | 282 | 210 | 300 | 1000 |
| | | ČKD DSKM | | X | | | X | | | | | | | | | OI | 162 | 2147 | 5 | 61 | 652 | 45 | 88,185 | 392,266 | 5 | 456 | 340 | 300 | 1000 |
| | | ČKD (T)SKM | | X | | | X | | | | | | | | | OI | 11 | 706 | 1 | 20 | 3623 | 250 | 88,185 | 392,266 | 4 | 429 | 320 | 300 | 1000 |
| | | Howden Roots URAI | X | | | | | | X | | | | | | | OF | 6 | 2350 | 0.2 | 66.5 | 25 | 1.7 | | | 2 | 75 | 56 | 600 | 5275 |
| | | Howden Roots RAM | X | | | | | | X | | | | | | | OF | 100 | 3300 | 2.8 | 93.4 | 25 | 1.7 | | | 2 | 175 | 130 | 900 | 4000 |
| | | Howden Roots RCS | X | | | | | | X | | | | | | | OF | 400 | 5500 | 11.3 | 155.7 | 25 | 1.7 | | | 2 | 325 | 242 | 950 | 2600 |
| | | Howden Roots Tri-RAM | X | | | | | | X | | | | | | | OF | 300 | 5500 | 8.5 | 155.7 | 25 | 1.7 | | | 2 | 350 | 261 | 1100 | 4750 |
| | | Howden Roots RAS | X | | | | | | X | | | | | | | OF | 600 | 29000 | 17 | 821.2 | 60 | 4.1 | | | 2 | 1000 | 746 | 600 | 1900 |
| | | Howden Roots RGS | X | | | | | | X | | | | | | | OF | 600 | 29000 | 17 | 821.2 | 60 | 4.1 | | | 2 | 1000 | 746 | 600 | 1900 |
| HYCOMP INC. | * | AN3A | X | | | | | | | | | | | | | OF | 2 | 4 | | | 1000 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | AN4A | X | | | | | | | | | | | | | OF | 3 | 6 | | | 1000 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | WN4A | X | | | | | | | | | | | | | OF | 3 | 6 | | | 1000 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | AN6A | X | | | | | | | | | | | | | OF | 4 | 8 | | | 1000 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | AN6 | X | | | | | | | | | | | | | OF | 4 | 8 | | | 600 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | WN07 | X | | | | | | | | | | | | | OF | 0 | 1 | | | 1500 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | AN12 | X | | | | | | | | | | | | | OF | 8 | 17 | | | 200 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | AN6C | X | | | | | | | | | | | | | OF | 4 | 9 | | | 750 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | WN6C | X | | | | | | | | | | | | | OF | 4 | 9 | | | 750 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | AN10C | X | | | | | | | | | | | | | OF | 7 | 14 | | | 600 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | WN10C | X | | | | | | | | | | | | | OF | 7 | 14 | | | 600 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | AN26 | X | | | | | | | | | | | | | OF | 17 | 36 | | | 200 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | WN26 | X | | | | | | | | | | | | | OF | 17 | 36 | | | 200 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | AN12D | X | | | | | | | | | | | | | OF | 8 | 16 | | | 1000 | | 4700 | | 5 | 20 | | 400 | 825 |
| *continued* | | AN17D | X | | | | | | | | | | | | | OF | 11 | 24 | | | 600 | | 4700 | | 5 | 20 | | 400 | 825 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# RECIPROCATING AND ROTARY COMPRESSORS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| HYCOMP INC. | * | AN23D | X | | | | | | | | | | | | | OF | 16 | 32 | | | 600 | | 4700 | | 5 | 20 | | 400 | 825 |
| | | AN44 | X | | | | | | | | | | | | | OF | 29 | 61 | | | 200 | | 4700 | | 5 | 20 | | 400 | 825 |
| | | AN14E | X | | | | | | | | | | | | | OF | 9 | 19 | | | 800 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | WN14E | X | | | | | | | | | | | | | OF | 9 | 19 | | | 800 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | AN20E | X | | | | | | | | | | | | | OF | 13 | 27 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | WN20E | X | | | | | | | | | | | | | OF | 13 | 27 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | AN27E | X | | | | | | | | | | | | | OF | 18 | 37 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | WN27E | X | | | | | | | | | | | | | OF | 18 | 37 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | AN35E | X | | | | | | | | | | | | | OF | 23 | 48 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | AN44E | X | | | | | | | | | | | | | OF | 29 | 61 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | AN72 | X | | | | | | | | | | | | | OF | 48 | 99 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | WN72 | X | | | | | | | | | | | | | OF | 48 | 99 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | WN90 | X | | | | | | | | | | | | | OF | 60 | 124 | | | 600 | | 6300 | | 5 | 40 | | 400 | 825 |
| | | WN28F | X | | | | | | | | | | | | | OF | 19 | 34 | | | 700 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | AN28F | X | | | | | | | | | | | | | OF | 19 | 34 | | | 700 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | AN44F | X | | | | | | | | | | | | | OF | 29 | 51 | | | 650 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | WN44F | X | | | | | | | | | | | | | OF | 29 | 51 | | | 650 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | WN55F | X | | | | | | | | | | | | | OF | 37 | 64 | | | 500 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | WN75F | X | | | | | | | | | | | | | OF | 50 | 88 | | | 400 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | WN98 | X | | | | | | | | | | | | | OF | 60 | 105 | | | 300 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | AN154 | X | | | | | | | | | | | | | OF | 102 | 179 | | | 250 | | 11,700 | | 5 | 66 | | 400 | 700 |
| | | 2AD4A | | X | | | | | | | | | | | | OF | 3 | 6 | | | 1000 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | 2AN6B | | X | | | | | | | | | | | | OF | 4 | 8 | | | 500 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | 2AN8 | | X | | | | | | | | | | | | OF | 6 | 11 | | | 500 | | 2800 | | 5 | 8 | | 400 | 825 |
| | | 2AN3C | | X | | | | | | | | | | | | OF | 2 | 5 | | | 1500 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | 2AN5C | | X | | | | | | | | | | | | OF | 3 | 7 | | | 1500 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | 2AN10C | | X | | | | | | | | | | | | OF | 6 | 12 | | | 500 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | 2WN10C | | X | | | | | | | | | | | | OF | 7 | 14 | | | 500 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | 2AN17 | | X | | | | | | | | | | | | OF | 11 | 23 | | | 500 | | 3700 | | 5 | 11 | | 400 | 825 |
| | | 2AN10D | | X | | | | | | | | | | | | OF | 7 | 14 | | | 750 | | 4700 | | 5 | 23 | | 400 | 825 |
| | | 2AN15D | | X | | | | | | | | | | | | OF | 10 | 21 | | | 750 | | 4700 | | 5 | 23 | | 400 | 825 |
| | | 2AN26 | | X | | | | | | | | | | | | OF | 18 | 36 | | | 500 | | 4700 | | 5 | 23 | | 400 | 825 |
| | | 2AN35 | | X | | | | | | | | | | | | OF | 23 | 47 | | | 500 | | 4700 | | 5 | 23 | | 400 | 825 |
| | | 2WN35 | | X | | | | | | | | | | | | OF | 23 | 47 | | | 500 | | 4700 | | 5 | 23 | | 400 | 825 |
| *continued* | | 2AN40 | | X | | | | | | | | | | | | OF | 26 | 54 | | | 500 | | 4700 | | 5 | 23 | | 400 | 825 |

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| HYCOMP INC. | * | 2WN40 | | X | | | | | | | | | | | | OF | 26 | 54 | | | 500 | | 4700 | | 5 | 23 | | 400 | 825 |
| | | 2WN13E | | X | | | | | | | | | | | | OF | 9 | 18 | | | 1000 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2AN17E | | X | | | | | | | | | | | | OF | 12 | 24 | | | 1000 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2WN17E | | X | | | | | | | | | | | | OF | 12 | 24 | | | 1000 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2AN22E | | X | | | | | | | | | | | | OF | 15 | 30 | | | 750 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2WN22E | | X | | | | | | | | | | | | OF | 15 | 30 | | | 750 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2AN61 | | X | | | | | | | | | | | | OF | 41 | 84 | | | 350 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2WN61 | | X | | | | | | | | | | | | OF | 41 | 84 | | | 350 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2AN76 | | X | | | | | | | | | | | | OF | 51 | 105 | | | 350 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2WN76 | | X | | | | | | | | | | | | OF | 51 | 105 | | | 350 | | 6300 | | 5 | 43 | | 400 | 825 |
| | | 2AN22F | | X | | | | | | | | | | | | OF | 15 | 25 | | | 1000 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2WN22F | | X | | | | | | | | | | | | OF | 15 | 25 | | | 1500 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2WN28F | | X | | | | | | | | | | | | OF | 18 | 32 | | | 1000 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2AN28F | | X | | | | | | | | | | | | OF | 18 | 32 | | | 750 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2WN38F | X | | | | | | | | | | | | | OF | 25 | 44 | | | 750 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2WN49F | X | | | | | | | | | | | | | OF | 33 | 57 | | | 750 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2AN58F | X | | | | | | | | | | | | | OF | 38 | 67 | | | 750 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2AN137 | X | | | | | | | | | | | | | OF | 91 | 159 | | | 200 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2WN150L | X | | | | | | | | | | | | | OF | 101 | 177 | | | 400 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 2WN150H | X | | | | | | | | | | | | | OF | 101 | 177 | | | 500 | | 11,700 | | 5 | 72 | | 400 | 700 |
| | | 3AN44 | X | | | | | | | | | | | | | OF | 29 | 61 | | | 500 | | 4700 | | 5 | 23 | | 400 | 825 |
| INGERSOLL RAND | * | PS-4 | | X | | | X | | | | | | | | | OF | 950 | 1800 | 1500 | 3300 | 650 | 45 | 18,000 | 80,070 | 3 | 750 | 560 | 340 | 514 |
| | | PHE 9 | | X | | | X | | | | | | | | | OF | 900 | 2400 | 1500 | 4000 | 1500 | 100 | 18,000 | 80,070 | 4 | 400 | 300 | 300 | 514 |
| | | PHE 7 | | X | | | X | | | | | | | | | OF | 100 | 800 | 150 | 1400 | 1200 | 83 | 10,000 | 44,480 | 4 | 200 | 150 | 300 | 650 |
| | | R-Series 4-11 | X | | | | | | | | X | | | | | OI | 12 | 58 | 0.3 | 1.6 | 212 | 14.6 | | | 6-8 | 15 | 11 | | |
| | | R-Series 15-22 | X | | | | | | | | X | | | | | OI | 68 | 113 | 1.9 | 3.2 | 212 | 14.6 | | | 6-8 | 30 | 22 | | |
| | | RS 30-37 | X | | | | | | | | X | | | | | OI | 186 | 230 | 5.3 | 6.5 | 212 | 14.6 | | | 6-8 | 50 | 37 | | |
| | | R-Series 45-75 | X | | | | | | | | X | | | | | OI | 263 | 478 | 7.4 | 13.5 | 212 | 14.6 | | | 6-8 | 100 | 75 | | |
| | | R-Series 90-160 | X | X | | | | | | | X | | | | | OI | 590 | 1030 | 16.7 | 29.2 | 212 | 14.6 | | | 6-8 | 225 | 160 | | |
| | | RS-Series 200-250 | X | X | | | | | | | X | | | | | OI | 1255 | 1750 | 35.5 | 49.6 | 212 | 14.6 | | | 6-8 | 225 | 160 | | |
| | | SSR 250-350 | X | X | | | | | | | X | | | | | OI | 1690 | 2450 | 47.9 | 69.4 | 212 | 14.6 | | | 2-4 | 500 | 350 | | |
| | | Sierra / Nirvana Oil Free 50-100 | | X | | | | | | | X | | | | | OF | 200 | 425 | 5.7 | 12 | 150 | 10.3 | | | 2-5 | 100 | 75 | | |
| | | Sierra / Nirvana Oil Free 125-225 | | X | | | | | | | X | | | | | OF | 570 | 920 | 16.1 | 26.1 | 150 | 10.3 | | | 2-6 | 225 | 160 | | |
| | | Sierra / Nirvana Oil Free 250-400 | | X | | | | | | | X | | | | | OF | 1200 | 1550 | 34.0 | 43.9 | 150 | 10.3 | | | 2-4 | 400 | 300 | | |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

## RECIPROCATING AND ROTARY COMPRESSORS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| KOBELCO | * | KR Series | X | X | | | X | | | | | | | | | OF/OI | | | | 400 | 10,150 | 700 | 350,000 | 160,000 | | 40,000 | 30,000 | 250 | 1000 |
| | | Dry Screw | X | X | | | | | | X | | | | | | OF | 140 | 84,000 | 4 | 2,380 | 650 | 45 | | | 10 | 13,500 | 10,000 | 1500 | 24,000 |
| | | Oil-Injected Screw | X | X | | | | | | X | | | | | | OI | 120 | 20,000 | 3.5 | 567 | 1500 | 100 | | | 20 | 13,500 | 10,000 | 1000 | 5500 |
| KÖHLER & HÖRTER GMBH (KOHO) | 81 | TWX 1 | X | X | | | | | | | | | | | | OF / OI | | | | | 2176 | 150 | 9000 | 40,000 | 4 | 41 | 30 | 250 | 800 |
| | | TWX 4 | X | X | | | | | | | | | | | | OF / OI | | | | | 2176 | 150 | 9000 | 40,000 | 4 | 74 | 55 | 250 | 800 |
| | | TWX 5 | X | X | | | | | | | | | | | | OF / OI | | | | | 2176 | 150 | 13,490 | 60,000 | 4 | 121 | 90 | 250 | 800 |
| | | TWX 6 | X | X | | | | | | | | | | | | OF / OI | | | | | 2176 | 150 | 9000 | 40,000 | 4 | 74 | 55 | 250 | 800 |
| | | TWX 7 | X | X | | | | | | | | | | | | OF / OI | | | | | 2611 | 180 | 13,490 | 60,000 | 4 | 121 | 90 | 250 | 800 |
| | | TWX 8 | X | X | | | | | | | | | | | | OF / OI | | | | | 2611 | 180 | 15,740 | 70,000 | 4 | 148 | 110 | 250 | 800 |
| | | TWX 9 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 470 | 350 | 250 | 750 |
| | | TWX 10 | X | X | | | | | | | | | | | | OF / OI | | | | | 2611 | 180 | 15,740 | 70,000 | 4 | 148 | 110 | 250 | 750 |
| | | TWX 11 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 671 | 500 | 250 | 600 |
| | | TWX 12 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 671 | 500 | 250 | 600 |
| | | TWX 13 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 671 | 500 | 250 | 600 |
| | | TWX 14 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 671 | 500 | 250 | 600 |
| | | TWX 15 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 671 | 500 | 250 | 600 |
| | | TWX 16 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 671 | 500 | 250 | 600 |
| | | TWX 17 | X | X | | | | | | | | | | | | OF / OI | | | | | 2176 | 150 | 9000 | 40,000 | 4 | 41 | 30 | 250 | 800 |
| | | TWX 18 | X | X | | | | | | | | | | | | OF / OI | | | | | 4352 | 300 | 20,240 | 90,000 | 4 | 671 | 500 | 250 | 600 |
| LEROI GAS COMPRESSORS | * | HG Series | X | | | | | | | X | | | | | | OI | 0 | 160 to 1040 | 0 | 4.5 to 30 | 350 | 24 | | | 25 | 90 to 300 | 67 to 224 | 500 to 1640 | 5480 |
| | | HHG24XXX | X | | | | | | | X | | | | | | OI | 0 | 1530 | 0 | 43 | 300 | 21 | | | 22 | 350 | 261 | 500 | 5870 |
| | | LG30XXX | X | | | | | | | X | | | | | | OI | 0 | 2840 | 0 | 80 | 250 | 17 | | | 15 | 600 | 447 | 350 | 3470 |
| | | LGL30XXX | X | | | | | | | X | | | | | | OI | 0 | 3750 | 0 | 106 | 150 | 10 | | | 12 | 900 | 671 | 350 | 2400 |
| | | LGT24xxx | | X | | | | | | X | | | | | | OI | 0 | 1550 | 0 | 44 | 150 | 10 | | | 8 | 400 | 298 | 500 | 3400 |
| | | LGT30xxx | | X | | | | | | X | | | | | | OI | 0 | 2900 | 0 | 82 | 150 | 10 | | | 8 | 600 | 448 | 950 | 3300 |
| | | HGT17xxx | | X | | | | | | X | | | | | | OI | 0 | 750 | 0 | 21 | 500 | 34 | | | 15 | 400 | 298 | 950 | 3350 |
| | | HGT24xxx | | X | | | | | | X | | | | | | OI | 0 | 1550 | 0 | 44 | 500 | 34 | | | 15 | 600 | 448 | 500 | 3680 |
| | | EN3000 | X | | | | | | | | | | | | | OI | 0 | 125 | 0 | 3.5 | 205 | 14 | | | 19 | 20 | 15 | 2500 | 10,000 |
| | | E6+000 | X | | | | | | | | | | | | | OI | 0 | 305 | 0 | 8.6 | 205 | 14 | | | 19 | 44 | 33 | 2000 | 8000 |
| | | E6+Gxxx | X | | | | | | | | | | | | | OI | 0 | 305 | 0 | 8.6 | 205 | 14 | | | 19 | 44 | 33 | 2000 | 8000 |
| | | E12+000 | X | | | | | | | | | | | | | OI | 0 | 610 | 0 | 17.3 | 205 | 14 | | | 19 | 67 | 50 | 1000 | 6360 |
| | | E12+Gxxx | X | | | | | | | | | | | | | OI | 0 | 610 | 0 | 17.3 | 205 | 14 | | | 19 | 67 | 50 | 1000 | 6360 |
| | | E12+DGTxxx | X | | | | | | | | | | | | | OI | 0 | 610 | 0 | 17.3 | 205 | 14 | | | 19 | 67 | 50 | 1000 | 6360 |
| | | E25000 | X | | | | | | | | | | | | | OI | 0 | 1050 | 0 | 29.6 | 205 | 14 | | | 19 | 107 | 80 | 1200 | 5600 |

continued

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Reciprocating: Single Stage | Multiple Stages | Integral Engine Driven | Separable | Balanced/Opposed | Diaphragm | Rotary: Straight Lobe | Helical Lobe (Screw) | Single Screw | Sliding Vane | Liquid-Ring | Trochoidal | Scroll | OF = Oil Free, OI = Oil Injected | Inlet Flow Range acfm min | acfm max | m³/min min | m³/min max | MAWP Maximum Allowable Working Pressure psig | bar | MARL Maximum Allowable Rod Load lb | Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | kW | Speed Range (rpm) min | max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| LEROI GAS COMPRESSORS | * | E25Gxxx | X | | | | | | | | | | | | | OI | 0 | 1050 | 0 | 29.6 | 205 | 14 | | | 19 | 107 | 80 | 1200 | 5600 |
| | | 35-LRG9-DP | | X | | X | | | | | | | | | | | 0 | 300 | 0 | 8.5 | 2500 | 172 | 6000 | 26,700 | 8 | 35 | 26 | 560 | 1200 |
| | | 55-LRG9-DP | | X | | X | | | | | | | | | | | 0 | 300 | 0 | 8.5 | 2500 | 172 | 6000 | 26,700 | 8 | 55 | 41 | 560 | 1200 |
| | | VRU30B | | X | | X | | | | | | | | | | | 0 | 150 | 0 | 4.3 | 275 | 6 | 3900 | 17,350 | 6 | 40 | 30 | 850 | 1015 |
| LMF COMPRESSORS | 112, 113 | Process Gas (API 618) | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,150 | 700 | 134,885 | 600,000 | 4 | 8300 | 6200 | 300 | 1200 |
| | | Process Gas (API 11P) | X | X | | X | X | | | | | | | | | OF/OI | | | | | 10,150 | 700 | 98,916 | 440,000 | 4 | 8160 | 6000 | 450 | 1800 |
| | | EcoPET | | X | | | | | | | | | | | | OF | | | | | 580 | 40 | 15,737 | 70,000 | 3 | 778 | 580 | 450 | 1200 |
| | | Mobile Systems | X | X | | | X | | | | | | | | | OF/OI | | | | | 10,150 | 700 | 16,861 | 75,000 | 4 | 1743 | 1300 | 450 | 1800 |
| | | CNG | | X | | | X | | | | | | | | | OF/OI | | | | | 5076 | 350 | 16,861 | 75,000 | 4 | 800 | 1200 | 450 | 1800 |
| | | Industrial Applications | X | X | | | X | | | | | | | | | OF/OI | | | | | 6527 | 450 | 24,729 | 110,000 | 6 | 800 | 1200 | 600 | 1800 |
| | | Electric Rotary Screw | X | X | | | | | | X | | | | | | OI | 15 | 2600 | 0,5 | 74 | 175 | 13 | | | | 475 | 355 | | |
| MAN ENERGY SOLUTIONS | 118, 119 | CP | X | X | | | | | | X | | | | | | OF | 120 | 12,000 | 3 | 340 | 725 | 50 | | | 10 | 13,410 | 10,000 | 3000 | 25,000 |
| | | SKUEL | X | X | | | | | | X | | | | | | OF | 2331 | 60,000 | 66 | 1700 | 230 | 16 | | | 10 | 13,410 | 10,000 | 1500 | 8900 |
| MEHRER COMPRESSION GMBH | * | Separable Style | X | X | | X | | | | | | | | | | OF | | | | | 217 to 2900 | 11 to 200 | | | 4, 5 | 7 to 268 | 5 to 200 | 380, 400 | 750 to 940 |
| | | Diaphram Style | X | X | | | | X | | | | | | | | OF | | | | | 14,504, 17,405 | 1000, 1200 | | | 10 | 15 to 225 | 11 to 160 | 400 | 400 |
| MITSUI E&S | * | C Series | X | X | | | X | | | | | | | | | OF/OI | | 14,125 | | 400 | 21,200 | 600 | | | 7 | 24,138 | 18,000 | | 500 |
| | | MB Series | X | X | | | X | | | | | | | | | OF/OI | | 14,125 | | | 21,200 | 600 | | | 7 | 134 | 1000 | | 700 |
| NATURAL GAS SERVICES GROUP/CIP | * | CiP PHT2 | X | X | | X | X | | | | | | | | | OI | 5 | 585 | 0.1 | 16.6 | 3050 | 210 | 20,000 | 890,000 | 4 | 250/400 | 186/298 | 800 | 1800 |
| | | CiP PXT2 | X | X | | X | X | | | | | | | | | OI | 5 | 585 | 0.1 | 16.6 | 3050 | 210 | 16,000 | 712,000 | 4 | 175/300 | 130/223 | 800 | 1800 |
| | | CiP PVT2 | X | X | | X | X | | | | | | | | | OF/OI | 3 | 270 | 0.1 | 7.7 | 2000 | 138 | 7500 | 34,000 | 4 | 100/150 | 74/111 | 900 | 2200 |
| NEUMAN & ESSER GROUP | 92, 93 | 25 | X | X | | X | | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 5620 | 25,000 | | 200 | 150 | | 900 |
| | | 40 | X | X | | X | | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 8540 | 38,000 | | 160 | 120 | | 1000 |
| | | 30 | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 18,000 | 80,000 | | 1900 | 1440 | | 1200 |
| | | 80hs | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 18,000 | 80,000 | | 1900 | 1440 | | 1800 |
| | | V1 | X | X | | X | | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 24,730 | 110,000 | | 890 | 660 | | 1800 |
| | | 63 | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 33,720 | 150,000 | | 3600 | 2700 | | 1800 |
| | | 150hs | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 33,720 | 150,000 | | 3600 | 2700 | | 1200 |
| | | 130 | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 56,200 | 250,000 | | 6000 | 4500 | | 1000 |
| | | 250hs | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 56,200 | 250,000 | | 6000 | 4500 | | 1000 |
| | | 190 | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 85,400 | 380,000 | | 9200 | 6840 | | 1000 |
| | | 320hs | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 105,700 | 470,000 | | 11,300 | 8460 | | 1200 |
| continued | | 300 | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 125,890 | 560,000 | | 13,500 | 10,080 | | 650 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# RECIPROCATING AND ROTARY COMPRESSORS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Reciprocating: Single Stage | Multiple Stages | Integral Engine Driven | Separable | Balanced/Opposed | Diaphragm | Rotary: Straight Lobe | Helical Lobe (Screw) | Single Screw | Sliding Vane | Liquid-Ring | Trochoidal | Scroll | OF = Oil Free, OI = Oil Injected | Inlet Flow Range acfm min | acfm max | m³/min min | m³/min max | MAWP Maximum Allowable Working Pressure psig | bar | MARL Maximum Allowable Rod Load lb | Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | kW | Speed Range (rpm) min | max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| NEUMAN & ESSER GROUP | 92, 93 | 560hs | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 125,890 | 560,000 | | 10,000 | 7500 | | 1200 |
| | | 320 | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 193,340 | 860,000 | | 20,800 | 15,480 | | 600 |
| | | 700hs | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 193,340 | 860,000 | | 20,800 | 15,480 | | 1200 |
| | | 500 | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 382,180 | 1,700,000 | | 41,000 | 30,600 | | 600 |
| | | 1500hs | X | X | | X | X | X | | | | | | | | OF/OI | | | | | 43,500 | 3000 | 382,180 | 1,700,000 | | 41,000 | 30,600 | | 1200 |
| PEDRO GIL S.l | * | Rotary Piston Blower | X | | | | | | X | | | | | | | OF | 1 | 170 | 0.5 | 150 | 29 | 2.0 | | | | 422 | 315 | | 4800 |
| PETER BROTHERHOOD | * | M | X | X | | | X | | | | | | | | | OI/OF | Dependent on inlet conditions | | | | 5800 | 400 | 24,728 | 110,000 | 3 | 2400 | 1800 | 300 | 750 |
| | | A | X | X | | | X | | | | | | | | | OI/OF | Dependent on inlet conditions | | | | 5800 | 400 | 44,960 | 200,000 | 3 | 4690 | 3500 | 300 | 600 |
| | | B | X | X | | | X | | | | | | | | | OI/OF | Dependent on inlet conditions | | | | 5800 | 400 | 71,936 | 320,000 | 3 | 8715 | 6500 | 300 | 500 |
| | | D | X | X | | | X | | | | | | | | | OI/OF | Dependent on inlet conditions | | | | 5800 | 400 | 105,656 | 470,000 | 3 | 15,420 | 11,500 | 300 | 400 |
| | | E | X | X | | | X | | | | | | | | | OI/OF | Dependent on inlet conditions | | | | 5800 | 400 | 143,872 | 640,000 | 3 | 26,820 | 20,000 | 300 | 360 |
| RO-FLO COMPRESSORS | * | Single-Stage, Sliding Vane | X | | | | | | | | | X | | | | OI | 0.5 to 880 | 30 to 2286 | 0 to 20.7 | 0.9 to 64.7 | 80, 150, 200 | 5, 10, 13 | | | 5, 7 | 15 to 500 | 11 to 373 | 275 to 865 | 640 to 2200 |
| | | Mulit-Stage, Sliding Vane | | X | | | | | | | | X | | | | OI | 15 to 700 | 34 to 2254 | 0.4 to 19.8 | 1 to 63.8 | 150, 200 | 10, 13 | | | 5 | 50 to 600 | 37 to 447 | 275 to 865 | 640 to 1465 |
| ROTORCOMP VERDICHTER GMBH | * | EVO Series Gas/ Geared | X | | | | | | | X | | | | | | OI | | | | 2 to 105 | 217 | 14 to 17 | | | | 15 to 900 | 11 to 670 | 1000 to 2000 | 3000 to 9300 |
| | | EVO2-NK Series | X | | | | | | | X | | | | | | OI | | | | 2 to 8 | 217 | 15 | | | | 15 to 75 | 11 to 55 | 1500, 2000 | 6300 to 9000 |
| | | NK200-Gas/ Geared | X | | | | | | | X | | | | | | OI | | | | 10 | 217 | 15 | | | | 102 | 75 | 1500 | 4000 |
| SAFE SPA | * | S7-S9 | | X | | X | | | | | | | | | | OI | | | | 4 | | 300 | | 22,000 | 4 | | 75 | 550 | 1500 |
| | | SW | X | X | | X | | | | | | | | | | OF/OI | | | | 18 | | 300 | | 50,000 | 4 | | 500 | 550 | 1500 |
| | | ST | X | X | | X | | | | | | | | | | OI | | | | 8 | | 250 | | 35,000 | 4 | | 110 | 550 | 1500 |
| | | Hydraulic series | X | X | | X | | | | | | | | | | OI | | | | 15 | | 300 | | | 4 | | 75 | | |
| SAUER COMPRESSORS | * | MISTRAL Series | X | X | | | | | | | | | | | | OI | | | | | 150 to580 | 10 to 40 | | | 1, 2 | | | 980 | 1780 |
| | | PASSAT Series | | X | | | | | | | | | | | | OI | | | | | 580 to 1160 | 40 to 80 | | | 3 | | | 980 | 1780 |
| | | BREEZE Series | | X | | | | | | | | | | | | OI | | | | | 580 | 40 | | | 3 | | | 980 | 1780 |
| | | HURRICANE Series | | X | | | | | | | | | | | | OI | | | | | 5800 | 350, 400 | | | 4 | | | 980 | 1780 |
| | | TORNADO Series | | X | | | | | | | | | | | | OI | | | | | 5080, 5800 | 350, 400 | | | 3, 4 | | | 980 | 1780 |
| | | HARMATTAN Series | | X | | | | | | | | | | | | OF | | | | | 150, 220 | 10, 15 | | | 2 | | | 980 | 1780 |
| | | TYPHOON Series | | X | | | | | | | | | | | | OI | | | | | 440 | 30 | | | 2 | | | 980 | 1780 |
| | | TYPHOON WP3100 | | X | | | | | | | | | | | | OI | | | | | 1450 | 100 | | | 3 | | | 980 | 1780 |
| | | WP Series | | X | | | | | | | | | | | | OI | | | | | 730 to 7250 | 50 to 500 | | | 3, 4, 5 | | | 980 | 1780 |
| sera Hydrogen GmbH | * | MV1 - MV6 | X | X | X | | | X | | | | | | | | OF | 0.058 | 294 | 0.100 | 500* | 13053 | 900 | on request | on request | up to 15** | 100 | 75 | 200 | 340 |

# RECIPROCATING AND ROTARY COMPRESSORS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Reciprocating: Single Stage | Multiple Stages | Integral Engine Driven | Separable | Balanced/Opposed | Diaphragm | Rotary: Straight Lobe | Helical Lobe (Screw) | Single Screw | Sliding Vane | Liquid-Ring | Trochoidal | Scroll | OF = Oil Free / OI = Oil Injected | Inlet Flow Range acfm min | acfm max | m³/min min | m³/min max | MAWP Maximum Allowable Working Pressure psig | bar | MARL Maximum Allowable Rod Load lb | Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | kW | Speed Range (rpm) min | max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| SERTCO | * | 98 HP | X | | | X | X | | | | | | | | | OI | 51 | 118 | 1.4 | 3.3 | 300 | 21 | | | 12 | 50 | 36 | 900 | 2100 |
| | | 350 LP | X | | | X | X | | | | | | | | | OI | 91 | 425 | 2.6 | 12 | 125 | 9 | | | 6 | 125 | 92 | 900 | 2100 |
| | | 350 HP | X | | | X | X | | | | | | | | | OI | 91 | 425 | 2.6 | 12 | 350 | 24 | | | 6 | 150 | 110 | 900 | 2100 |
| | | 632 HP | X | | | X | X | | | | | | | | | OI | 91 | 425 | 2.6 | 12 | 350 | 24 | | | 6 | 180 | 132 | 900 | 2100 |
| SIAD MACCHINE IMPIANTI S.P.A. | 107 | HT Series | X | X | | X | X | | | | | | | | | OI/OF | | | | | 8,700 | 600 | 140,000 | 625,000 | 4 | 11,900 | 8,700 | 300 | 500 |
| | | HSF Series | X | X | | X | X | | | | | | | | | OI/OF | | | | | 8,700 | 600 | 78,000 | 350,000 | 4 | 8,700 | 6400 | 490 | 1200 |
| | | HSD Series | X | X | | X | X | | | | | | | | | OI/OF | | | | | 8,700 | 600 | 55,000 | 240,000 | 4 | 5,600 | 4100 | 490 | 1500 |
| | | HD Series | X | X | | X | X | | | | | | | | | OI/OF | | | | | 8,700 | 600 | 55,000 | 240,000 | 4 | 3,275 | 2400 | 300 | 750 |
| | | HP Series | X | X | | X | X | | | | | | | | | OI/OF | | | | | 8,700 | 600 | 37,500 | 165,000 | 4 | 1,950 | 1400 | 300 | 750 |
| | | HM Series | X | X | | X | X | | | | | | | | | OI/OF | | | | | 5,075 | 350 | 22,000 | 95,000 | 4 | 950 | 700 | 300 | 750 |
| | | P Series | X | X | | X | | | | | | | | | | OI/OF | | | | | 5,075 | 350 | 30,000 | 133,000 | 4 | 1,500 | 1100 | 300 | 750 |
| | | M Series | X | X | | X | | | | | | | | | | OI/OF | | | | | 5,075 | 350 | 22,000 | 95,000 | 4 | 700 | 525 | 300 | 750 |
| | | W Series | X | X | | X | | | | | | | | | | OI/OF | | | | | 5,075 | 350 | 10,500 | 45,000 | 4 | 280 | 200 | 300 | 1200 |
| | | T Series | X | X | | X | | | | | | | | | | OI/OF | | | | | 5,075 | 350 | 3,500 | 15000 | 4 | 105 | 75 | 300 | 1200 |
| | | I Series | X | X | | X | | | | | | | | | | OI/OF | | | | | 1,450 | 100 | 1,350 | 6000 | 4 | 14 | 10 | 300 | 1200 |
| SIEMENS ENERGY | * | AVIP | X | X | | X | X | | | | | | | | | OF/OI | | | | | 11,000 | 758 | 15,400 | 68,500 | | 1300 | 969 | 750 | 1800 |
| | | BVIP | X | X | | X | X | | | | | | | | | OF/OI | | | | | 11,000 | 758 | 24,200 | 107,640 | | 2125 | 1585 | 600 | 1500 |
| | | CVIP | X | X | | X | X | | | | | | | | | OF/OI | | | | | 11,000 | 758 | 33,000 | 146,780 | | 2880 | 2148 | 600 | 1800 |
| | | MOS | X | X | | X | X | | | | | | | | | OF/OI | | | | | 11,000 | 758 | 45,000 | 200,100 | | 4440 | 3281 | 500 | 1500 |
| | | HOS | X | X | | X | X | | | | | | | | | OF/OI | | | | | 11,000 | 758 | 60,000 | 266,880 | | 7200 | 5369 | 500 | 1500 |
| | | HOSS (Super HOS) | X | X | | X | X | | | | | | | | | OF/OI | | | | | 11,000 | 758 | 75,000 | 333,600 | | 8700 | 6488 | 500 | 1200 |
| | | 7"ESH/ESV | X | X | | | | | | | | | | | | OF/OI | | | | | 12,000 | 827 | 10,200 | 45,400 | | 75 | 56 | 300 | 600 |
| | | 9"/11"ESH | X | X | | | | | | | | | | | | OF/OI | | | | | 12,000 | 827 | 21,600 | 96,100 | | 180 | 134 | 300 | 450 |
| | | BDC-12H | X | X | | | X | | | | | | | | | OF/OI | | | | | 12,000 | 827 | 80,000 | 356,000 | | 8200 | 6115 | 277 | 600 |
| | | BDC-18H3 | X | X | | | X | | | | | | | | | OF/OI | | | | | 12,000 | 827 | 350,000 | 1,557,000 | | 45,000 | 33,556 | 277 | 450 |
| | | HHE Series | X | X | | | X | | | | | | | | | OF/OI | | | | | 12,000 | 827 | 30,000 to 200,00 | 133,500 to 890,000 | | 2250 to 22,500 | 1678 to 16,778 | 277, 300 | 450, 600 |
| | | HSE | X | X | | | X | | | | | | | | | OF/OI | | | | | 12,000 | 827 | 27,600 | 122,800 | | 786 | 586 | 300 | 600 |
| | | PHE | X | X | | | X | | | | | | | | | OF/OI | | | | | 12,000 | 827 | 10,200 | 45,200 | | 240 | 179 | 300 | 720 |
| SULLAIR | * | PDX Series | X | | | | | | | X | | | | | | OI | 75 to 1168 | 251 to 3893 | | | 200 | 14.0 | | | | | | | |
| | | PDR Series | X | | | | | | | X | | | | | | OI | 863, 1168 | 2810 to 3893 | | | 100, 150 | 7.0, 10.0 | | | | | | | |
| *continued* | | PDH Series | X | | | | | | | X | | | | | | OI | 86, 136 | 288, 453 | | | 500, 400 | 35, 28 | | | | | | | |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# RECIPROCATING AND ROTARY COMPRESSORS

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Single Stage | Multiple Stages | Reciprocating: Integral Engine Driven | Reciprocating: Separable | Reciprocating: Balanced/Opposed | Reciprocating: Diaphragm | Rotary: Straight Lobe | Rotary: Helical Lobe (Screw) | Rotary: Single Screw | Rotary: Sliding Vane | Rotary: Liquid-Ring | Rotary: Trochoidal | Rotary: Scroll | OF = Oil Free, OI = Oil Injected | Inlet Flow Range acfm min | Inlet Flow Range acfm max | Inlet Flow Range m³/min min | Inlet Flow Range m³/min max | MAWP Maximum Allowable Working Pressure psig | MAWP bar | MARL Maximum Allowable Rod Load lb | MARL Newtons | Compression Ratio (Per Stage) | Maximum Input Power hp | Maximum Input Power kW | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| SULLAIR | * | PC16L | X | | | | | | | X | | | | | | OI | 220 1392 | 733 to 4640 | | | 350, 400 | 24.0, 28.0 | | | | | | | |
| | | DC Series | X | | | | | | | X | | | | | | OI | 330 to 1200 | 200 to 1050 | | | 200 | 14.0 | | | | | | | |
| TM.I.C. - Termomeccanica Industrial Compressors | 86, 87 | NG8 | X | | | | | | | X | | | | | | OI | 21 | 88 | 0.6 | 2.5 | 247 | 17 | | | | 31 | 23 | 2600 | 10,300 |
| | | NG9 | X | | | | | | | X | | | | | | OI | 28 | 106 | 0.8 | 3.0 | 232 | 16 | | | | 43 | 32 | 1750 | 6150 |
| | | SCG10 | X | | | | | | | X | | | | | | OI | 35 | 222 | 1.0 | 6.3 | 290 | 20 | | | | 76 | 57 | 1850 | 6800 |
| | | NG13 | X | | | | | | | X | | | | | | OI | 71 | 290 | 2.0 | 8.2 | 290 | 20 | | | | 114 | 85 | 1500 | 6600 |
| | | NG14 | X | | | | | | | X | | | | | | OI | 78 | 424 | 2.2 | 12 | 290 | 20 | | | | 148 | 110 | 1300 | 6500 |
| | | NG21 | X | | | | | | | X | | | | | | OI | 134 | 706 | 4 | 20 | 290 | 20 | | | | 295 | 220 | 1000 | 5000 |
| | | NG22 | X | | | | | | | X | | | | | | OI | 177 | 883 | 5 | 25 | 290 | 20 | | | | 335 | 250 | 900 | 3900 |
| | | NG30 | X | | | | | | | X | | | | | | OI | 283 | 1589 | 8 | 45 | 290 | 20 | | | | 536 | 400 | 700 | 3400 |
| | | ITA-HP13 | X | | | | | | | X | | | | | | OI | 53 | 318 | 1.5 | 9 | 363 | 25 | | | | 148 | 110 | 1500 | 7400 |
| | | ITA-HP26 | X | | | | | | | X | | | | | | OI | 318 | 1271 | 9 | 36 | 363 | 25 | | | | 469 | 350 | 1100 | 3600 |
| | | ITA-TS23 | | X | | | | | | X | | | | | | OI | 494 | 953 | 14 | 27 | 363 | 25 | | | | 362 | 270 | 1800 | 3600 |
| TM.P. BARE SHAFT COMPRESSOR | * | NG Series | X | | | | | | | X | | | | | | OI | | | 0.5 to 15 | 2.7 to 50 | | 20 | | | | | 25 to 500 | 1000 to 3000 | 3500 to 10,500 |
| | | NG twin | X | | | | X | | | | | | | | | OI | | | 50 | 100 | | 20 | | | | | 950 | 1000 | 3500 |
| VILTER MANUFACTURING LLC | * | VSG Series | | | | | | | | | X | | | | | OI | | 310 to 2962 | | 8.2 to 83.9 | 485 to 950 | 33 to 65 | | | 20 | 400 to 2000 | 298 to 1491 | | 3800 to 4800 |
| VPT KOMPRESSOREN GMBH | * | RS Compact | X | X | | | | | | X | | | | | | OI | 0 | 15 | | | | 16 | | | 15 | | 110 | | 6000 |
| | | WV | X | X | | | | | | X | | | | | | OI | 1 | 130 | | | | 80 | | | 25 | | 3000 | | 3600 |
| | | WF | X | X | | | | | | X | | | | | | OI | 1 | 15 | | | | 26 | | | 25 | | 250 | | 9000 |
| | | WCV | X | X | | | | | | X | | | | | | OI | 1 | 130 | | | | 80 | | | 25 | | 3000 | | 3600 |
| | | WCF | X | X | | | | | | X | | | | | | OI | 1 | 15 | | | | 26 | | | 25 | | 250 | | 9000 |
| | | WST | X | X | | | | | | X | | | | | | OF | 10 | 150 | | | | 60 | | | 4 | | 3000 | | 24,000 |
| YORK/FRICK (JCI) | * | RXF Series | | | | | | | | X | | | | | | OI | | 89 to 596 | | 2.5 to 16.9 | 334 | 23 | | | | | | | 3550 to 6297 |
| | | RWF Series | | | | | | | | X | | | | | | OI | | 592 to 5012 | | 6.8 to 142 | 600 | 41, 48 | | | | | | | 3600, 4500 |
| | | RWHII Series | | | | | | | | X | | | | | | OI | | 5068 to 8212 | | 143 to 232 | 600 | 41 | | | | | | | 3600 |
| | | HPS Series | | | | | | | | X | | | | | | OI | | 346 to 1298 | | 9.8 to 36.8 | 1300 | 90 | | | | | | | 3600, 6000 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

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# Compressors and expanders

**FIG. 13-1**
**Nomenclature**

ACFM = actual cubic feet per minute (i.e. at process conditions)
$A_p$ = cross sectional area of piston,sq in
$A_r$ = cross sectional area of piston rod, sq in
BHP = brake or shaft horsepower
C = cylinder clearance as a percent of piston displacement
$C_p$ = specific heat at constant pressure, BTU/(lb · °F)
$C_v$ = specific heat at constant volume, BTU/(lb · °F)
D = cylinder inside diameter, in
d = piston rod diameter, in
E = overall efficiency
High speed reciprocating units — 0.82
Low speed reciprocating units — 0.85
EP = extracted horsepower of expander
F = an allowance for interstage pressure drop, Eq 13-4
GHP = gas horsepower, actual compression horsepower, excluding mechanical losses, BHP
H = head, ft · lb/lb
h = enthalpy, Btu/lb
ICFM = inlet cubic feet per minute, usually at suction conditions
k = Cp/Cv
$MC_p$ = molar specific heat at constant pressure, BTU/(lb mole · °F)
$MC_v$ = molar specific heat at constant volume, BTU/(lb mole · °F)
MW = molecular weight, lb/lb mole
$M_N$ = machine mach number
N = speed, rpm
$N_m$ = molar flow, moles/min
n = polytropic exponent or number of moles
P = pressure, psia
$P_c$ = critical pressure, psia
PD = piston displacement, $ft^3$/min
$P_L$ = pressure base used in the contract or regulation, psia
$pP_c$ = pseudo critical pressure, psia
$P_R$ = reduced pressure, $P/P_c$
$pT_c$ = pseudo critical temperature, °R
Q = inlet capacity (ICFM)
$Q_g$ = standard gas flow rate, MMSCFD

R = universal gas constant = $10.73 \dfrac{psia \cdot ft^3}{lb\ mole \cdot °R}$
$= 1545 \dfrac{lb/ft^3}{lb\ mole \cdot °R}$ or $\dfrac{ft \cdot lb}{lb\ mole \cdot °R}$
$= 1.986 \dfrac{Btu}{lb\ mole \cdot °R}$
r = compression ratio, $P_2/P_1$
s = entropy, BTU/(lb · °R)
sm = surge margin
SCFM = cubic feet per minute measured at 14.7 psia and 60°F
stroke = length of piston movement, in
T = absolute temperature, °R
$T_c$ = critical temperature, °R
$T_R$ = reduced temperature, $T/T_c$
t = temperature, °F
U = impeller tip speed
V = specific volume, $ft^3$/lb
v = velocity ft/s
VE = volumetric efficiency, percent
W = work, ft · lb
w = weight flow, lb/min
X = temperature rise factor
y = mole fraction
Z = compressibility factor
$Z_{avg}$ = average compressibility factor = $(Z_s + Z_d)/2$
η = efficiency, expressed as a decimal
ρ = density, $lb/ft^3$

**Subscripts**

avg = average
d = discharge
g = gas
is = isentropic process
L = standard conditions used for calculation or contract
m = mechanical
p = polytropic process
S = standard conditions, usually 14.7 psia, 60°F
s = suction
t = total or overall
1 = inlet conditions
2 = outlet conditions

### DEFINITIONS OF WORDS AND PHRASES USED IN COMPRESSORS AND EXPANDERS

**Absolute pressure:** the pressure measured from an absolute vacuum. It equals the algebraic sum of barometric pressure and gauge pressure.

**Static pressure:** the pressure in the gas measured in such a manner that no effect is produced by the velocity of the gas stream. It is the pressure that would be shown by a measuring instrument moving at the same velocity as the moving stream and is the pressure used as a property in defining the thermodynamic state of the fluid.

**Stagnation (total) pressure:** the pressure which would be measured at the stagnation point when a moving gas stream is brought to rest and its kinetic energy is converted to an enthalpy rise by an isentropic compression from the flow condition to the stagnation condition. It is the pressure usually measured by an impact tube. In a stationary body of gas, the static and stagnation pressures are numerically equal.

**Velocity pressure (dynamic pressure):** the stagnation pressure minus the static pressure in a gas stream. It is the pressure generally measured by the differential pressure reading of a Pitot tube

**Absolute temperature:** the temperature above absolute zero. It is equal to the degrees Fahrenheit plus 459.66, and is stated as degrees Rankine.

**Static temperature:** the temperature that would be shown by a measuring instrument moving at the same velocity as the fluid stream. It is the temperature used as a property in defining the thermodynamic state of the gas.

**Stagnation (total) temperature:** that temperature which would be measured at the stagnation point if a gas stream were brought to rest and its kinetic energy converted to an enthalpy rise by an isentropic compression process from the flow condition to the stagnation condition.

**Capacity:** (Actual Flow) of a compressor is the volume rate of flow of gas compressed and delivered referred to conditions of pressure, temperature and gas composition prevailing at the compressor inlet.

**Standard or normal flow:** the rate of flow under certain 'standard' conditions, for example 60°F and 14.7 psia (US Standard) or 15°C and 101.325 kPa (GPA-SI Standard).

**Mass flow:** the rate of flow in mass units.

**Isentropic compression:** refers to the reversible adiabatic compression process.

**Isentropic work (head):** the work required to compress a unit mass of gas in an isentropic compression process from the inlet pressure and temperature to the discharge pressure.

**Isentropic power:** defined as the power required to compress isentropically and deliver the capacity of the compressor from the compressor inlet conditions to the compressor discharge pressure.

**Isentropic efficiency:** the ratio of the isentropic work to the work required for the compression process.

**Polytropic compression:** a reversible compression process between the compressor inlet and discharge conditions, which follows a path such that, between any two points on the path, the ratio of the reversible work input to the enthalpy rise is constant. In other words, the compression process is described as an infinite number of isentropic compression steps, each followed by an isobaric heat addition. The result is an ideal, reversible process that has the same suction pressure, discharge pressure, suction temperature and discharge temperature as the actual process.

**Polytropic work (head):** the reversible work required to compress a unit mass of the gas in a polytropic compression process.

## Compressors

Depending on application, compressors are manufactured as positive-displacement, dynamic, or thermal type (Fig. 13-2).

Positive displacement types fall in two basic categories: reciprocating and rotary.

The reciprocating compressor consists of one or more cylinders each with a piston or plunger that moves back and forth, displacing a positive volume with each stroke.

The diaphragm compressor uses a hydraulically pulsed flexible diaphragm to displace the gas.

Rotary compressors cover lobe-type, screw-type, vane-type, and liquid ring type, each having a casing with one or more rotating elements that either mesh with each other such as lobes or screws, or that displace a fixed volume with each rotation.

The dynamic types include radial-flow (centrifugal), axial-flow, and mixed flow machines. They are rotary continuous-flow compressors in which the rotating element (impeller or bladed rotor) accelerates the gas as it passes through the element, converting the velocity head into static pressure, partially in the rotating element and partially in stationary diffusers or blades.

Ejectors are "thermal" compressors that use a high velocity gas or steam jet to entrain the inflowing gas, then convert the velocity of the mixture to pressure in a diffuser.

Fig. 13-3 covers the normal range of operation for compressors of the commercially available types.

The advantages of a centrifugal compressor over a reciprocating machine are:

1. Lower installed first cost where pressure and volume conditions are favorable,
2. Lower maintenance expense,

3. Greater continuity of service and dependability,
4. Less operating attention,
5. Greater volume capacity per unit of plot area,
6. Adaptability to high-speed low-maintenance-cost drivers.

The advantages of a reciprocating compressor over a centrifugal machine are:

1. Greater flexibility in capacity and pressure range,
2. Higher compressor efficiency and lower power cost,
3. Capability of delivering higher pressures,
4. Capability of handling smaller volumes,
5. Less sensitive to changes in gas composition and density.

## RECIPROCATING COMPRESSORS

Reciprocating compressor ratings vary from fractional to more than 40,000 hp per unit. In gas processing it would be unusual for units larger than 10,000 hp to be used. Pressures range from low vacuum at suction to 30,000 psi and higher at discharge for special process compressors.

Reciprocating compressors are furnished either single-stage or multi-stage. The number of stages is determined by the overall compression ratio. The compression ratio per stage (and valve life) is generally limited by the discharge temperature and usually does not exceed 4, although small-sized units (intermittent duty) are furnished with a compression ratio as high as 8.

Gas cylinders are generally lubricated, although a non-lubricated design is available when warranted; example: nitrogen, oxygen, and instrument air.

On multistage machines, intercoolers may be provided between stages. These are heat exchangers which remove the heat of compression from the gas and reduce its temperature to approximately the temperature existing at the compressor intake. Such cooling reduces the actual volume of gas going to the high-pressure cylinders, reduces the horsepower required for compression, and keeps the temperature within safe operating limits.

Reciprocating compressors should be supplied with clean gas as they cannot satisfactorily handle liquids and solid particles that may be entrained in the gas. Liquids and solid particles tend to destroy cylinder lubrication and cause excessive wear. Liquids are non-compressible and their presence could cause major damage to the compressor cylinder or frame components.

Reciprocating compressors are typically designed to one of the following industry standard specifications:

API Standard 618 "Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services."

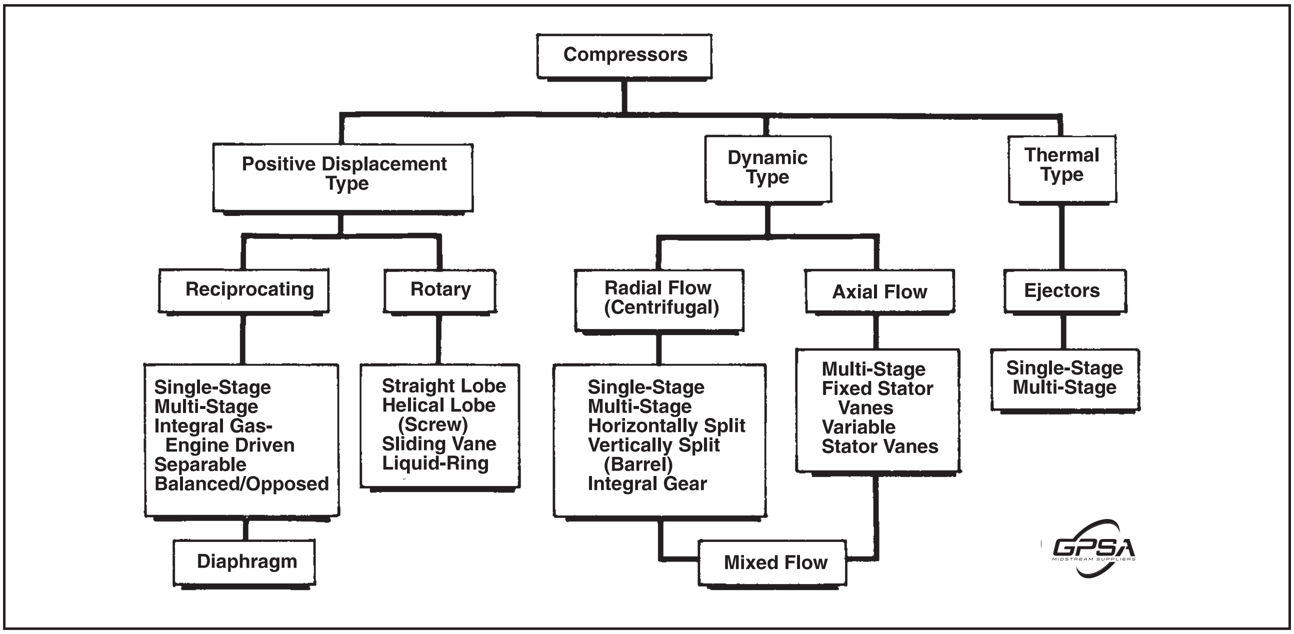
ISO Standard 13631: 2002, "Petroleum and Natural Gas Industries — Packaged Reciprocating Compressors."

Low to moderate speed compressors, typically 300–700 rpm, have historically been used in refineries, chemical plants and also can be used in gas plant service. They are normally driven by electric motors. These compressors are typically applied in accordance with API Standard 618 "Reciprocating Compressors for Petroleum, Chemical and Gas Industry Services."

Moderate to high speed compressors, typically 600–1800 rpm packaged separable compressors are used for field gas compression, mid-stream compression, gas plant and mainline compression. These units are normally driven by gas engines or electric motors. These compressors are typically applied in accordance with ISO Standard 13631.

**FIG. 13-2**

**Types of Compressors**



A low speed "integral" compressor refers to a compressor driven by a gas engine where the power cylinders of the engine that turn the crankshaft are in the same housing as the gas compression cylinders. (See Fig. 13-4). These compressors are no longer manufactured but there are a number of them still in operation in pipeline boosting service as well as inlet compression service at field gas plants. Integral compressors were designed to API 11 which is no longer supported by API.

## Performance Calculations

The engineer in the field is frequently required to:

1. determine the approximate horsepower required to compress a certain volume of gas from some intake conditions to a given discharge pressure, and
2. estimate the capacity of an existing compressor under specified suction and discharge conditions.

The following text outlines procedures for making these calculations from the standpoint of quick estimates and also presents more detailed calculations. For specific information on a given compressor, consult the manufacturer of that unit.

For a compression process, the enthalpy change is the best way of evaluating the work of compression. If a P-H diagram is available (as for propane refrigeration systems), the work of compression would always be evaluated by the enthalpy change of the gas in going from suction to discharge conditions. Years ago the capability of easily generating P-H diagrams for natural gases did not exist. The result was that many ways of estimating the enthalpy change were developed. They were used as a crutch and not because they were the best way to evaluate compression horsepower requirements.

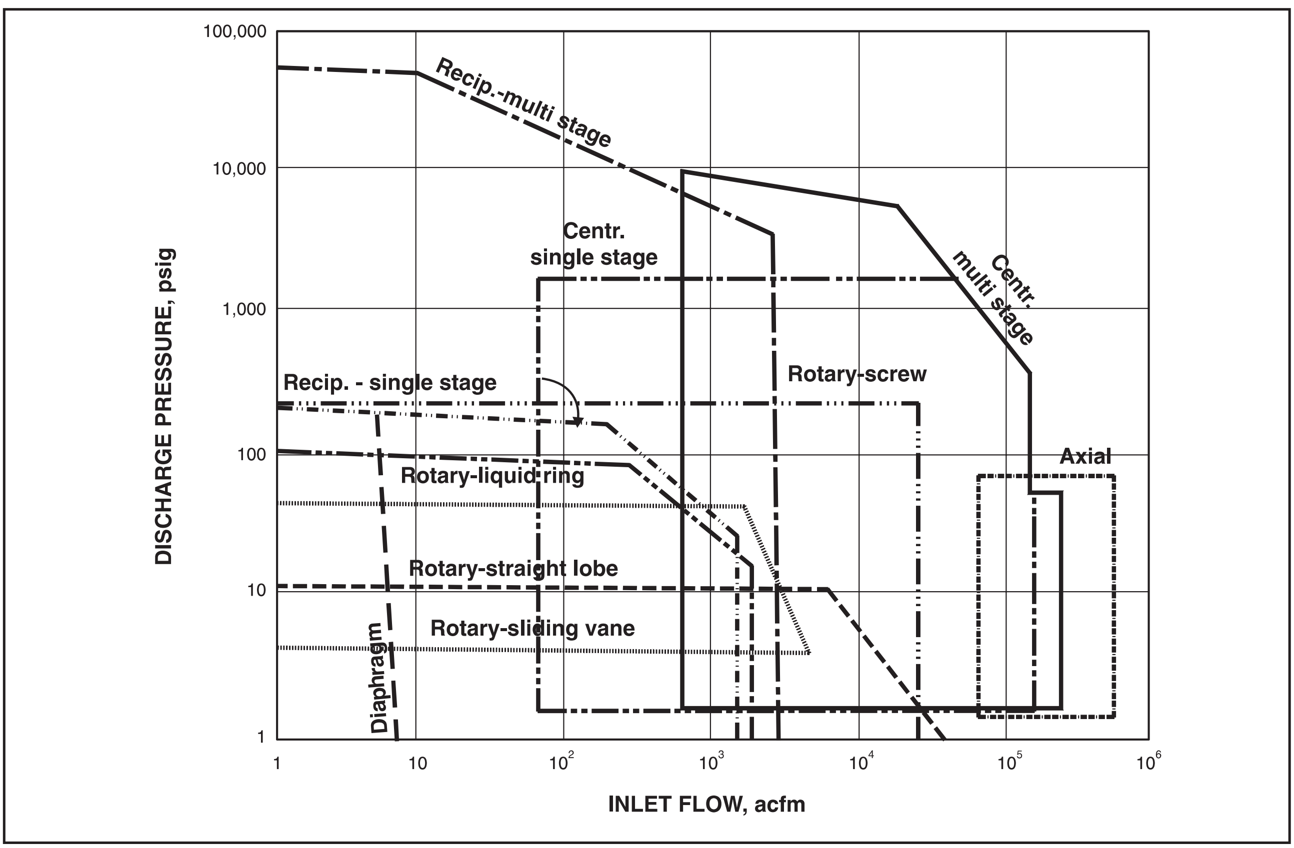
Today the engineer does have available, in many cases, the capability to generate that part of the P-H diagram required for compression purposes. This is done using equations of state on a computer. This still would be the best way to evaluate the compression horsepower. The other methods are used only if access to a good equation of state is not available.

Section 13 continues to treat reciprocating and centrifugal machines as being different so far as estimation of horsepower requirements is concerned. This treatment reflects industry practice. The only difference in the horsepower evaluation is the efficiency of the machine. Otherwise the basic thermodynamic equations are the same for all compression.

The reciprocating compressor horsepower calculations presented are based on charts. However, they may equally well be calculated using the equations in the centrifugal compressor section, particularly Equations 13-25 through 13-43. This also includes the mechanical losses in Equations 13-37 and 13-38.

There are two ways in which the thermodynamic calculations for compression can be carried out — by assuming:

**FIG. 13-3**
**Compressor Coverage Chart**



1. isentropic reversible path — a process during which there is no heat added to or removed from the system and the entropy remains constant, $pv^k$ = constant

2. polytropic reversible path — a process in which changes in gas characteristics during compression are considered, $pv^n$ = constant

Fig. 13-5 shows a plot of pressure vs. volume for each value of the above exponents. The work, W, performed in proceeding from $p_1$ to $p_2$ along any polytropic curve (Fig. 13-5) is

$$W = \int_1^2 V \cdot dp = \int_{p1}^{p2} V \cdot dp \qquad \textbf{Eq 13-1}$$

The amount of work required is dependent upon the polytropic curve involved and increases with increasing values of n. The path requiring the least amount of input work is n = 1, which is equivalent to isothermal compression, a process during which there is no change in temperature. For isentropic compression, the exponent used is k = ratio of specific heat at constant pressure to that at constant volume.

It is usually impractical to build sufficient heat-transfer equipment into the design of most compressors to carry away the bulk of the heat of compression. Most machines tend to operate along a polytropic path which approaches the isentropic. Most compressor calculations are therefore based on an efficiency applied to account for true behavior.

A compression process following the outer curve in Fig. 13-5 has been widely referred to in industry as "adiabatic". However, all compression processes of practical importance are adiabatic. The term adiabatic does not adequately describe this process, since it only implies no heat transfer. The ideal process also follows a path of constant entropy and should be called "isentropic," as will be done subsequently in this chapter.

Equation 13-3 which applies to all ideal gases can be used to calculate k.

$$MC_p - MC_v = R = 1.986 \text{ Btu/(lbmol} \cdot \text{°F)} \qquad \textbf{Eq 13-2}$$

By rearrangement and substitution we obtain:

$$k = \frac{C_p}{C_v} = \frac{MC_p}{MC_v} = \frac{MC_p}{MC_p - 1.986} \qquad \textbf{Eq 13-3}$$

To calculate k for a gas we need only know the constant pressure molar heat capacity ($MC_p$) for the gas. Fig. 13-6 gives values of molecular weight and ideal-gas state heat capacity (i.e. at 1 atm) for various gases. The heat capacity varies considerably with temperature. Since the temperature of the gas increases as it passes from suction to discharge in the compressor, k is normally determined at the average of suction and discharge temperatures.

For a multi-component gas, the mole weighted average value of molar heat capacity must be determined at average cylinder temperature. A sample calculation is shown in Fig. 13-7.

The calculation of $pP_c$ and $pT_c$ in Fig. 13-7 permits calculation of the reduced pressure $P_R = P/pP_c$ mix and reduced temperature $T_R = T/pT_c$ mix. The compressibility Z at T and P can then be determined using the charts in Section 23.

If only the molecular weight of the gas is known and not its composition, an approximate value for k can be determined from the curves in Fig. 13-8.

## Estimating Compressor Horsepower

Equation 13-4 is useful for obtaining a quick and reasonable estimate for compressor horsepower. It was developed for large slow-speed (300 to 450 rpm) compressors handling gases with a specific gravity of 0.65 and having stage compression ratios above 2.5.

CAUTION: Compressor manufacturers generally rate their machines based on a standard condition of 14.4 psia rather than the more common gas industry value of 14.7 psia.

Due to higher valve losses, the horsepower requirement for high-speed compressors (1000 rpm range, and some up to 1800 rpm) can be as much as 20% higher, although this is a very arbitrary value. Some compressor designs do not merit a higher horsepower allowance and the manufacturers should be consulted for specific applications.

$$\text{Brake horsepower} = (22)\left(\frac{\text{ratio}}{\text{stage}}\right)(\text{\# of stages})(\text{MMcfd})(F) \qquad \textbf{Eq 13-4}$$

Where:

MMcfd = Compressor capacity referred to 14.4 psia and intake temperature

F = 1.0 for single-stage compression
1.08 for two-stage compression
1.10 for three-stage compression

Equation 13-4 will also provide a rough estimate of horsepower for lower compression ratios and/or gases with a higher specific gravity, but it will tend to be on the high side. To allow for this the tendency is to use a multiplication factor of 20 instead of 22 for gases with a specific gravity in the 0.8 to 1.0

**FIG. 13-4**
**Integral Engine Compressor**

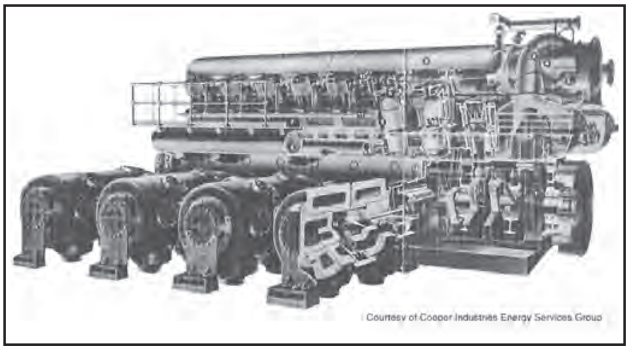
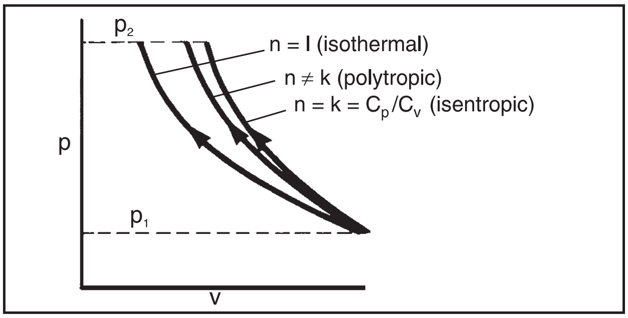



**FIG. 13-5**
**Compression Curves**



### FIG. 13-6
### Molar Heat Capacity $MC_p$ (Ideal-Gas State), Btu/(lb mol • °R)

*Data source: Selected Values of Properties of Hydrocarbons, API Research Project 44; MW updated to agree with Fig. 23-2

| Gas | Chemical formula | Mol wt | Temperature | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0°F | 50°F | 60°F | 100°F | 150°F | 200°F | 250°F | 300°F |
| Methane | $CH_4$ | 16.043 | 8.23 | 8.42 | 8.46 | 8.65 | 8.95 | 9.28 | 9.64 | 10.01 |
| Ethyne (Acetylene) | $C_2H_2$ | 26.038 | 9.68 | 10.22 | 10.33 | 10.71 | 11.15 | 11.55 | 11.90 | 12.22 |
| Ethene (Ethylene) | $C_2H_4$ | 28.054 | 9.33 | 10.02 | 10.16 | 10.72 | 11.41 | 12.09 | 12.76 | 13.41 |
| Ethane | $C_2H_6$ | 30.070 | 11.44 | 12.17 | 12.32 | 12.95 | 13.78 | 14.63 | 15.49 | 16.34 |
| Propene (Propylene) | $C_3H_6$ | 42.081 | 13.63 | 14.69 | 14.90 | 15.75 | 16.80 | 17.85 | 18.88 | 19.89 |
| Propane | $C_3H_8$ | 44.097 | 15.65 | 16.88 | 17.13 | 18.17 | 19.52 | 20.89 | 22.25 | 23.56 |
| 1-Butene (Butlyene) | $C_4H_8$ | 56.108 | 17.96 | 19.59 | 19.91 | 21.18 | 22.74 | 24.26 | 25.73 | 27.16 |
| cis-2-Butene | $C_4H_8$ | 56.108 | 16.54 | 18.04 | 18.34 | 19.54 | 21.04 | 22.53 | 24.01 | 25.47 |
| trans-2-Butene | $C_4H_8$ | 56.108 | 18.84 | 20.23 | 20.50 | 21.61 | 23.00 | 24.37 | 25.73 | 27.07 |
| iso-Butane | $C_4H_{10}$ | 58.123 | 20.40 | 22.15 | 22.51 | 23.95 | 25.77 | 27.59 | 29.39 | 31.11 |
| n-Butane | $C_4H_{10}$ | 58.123 | 20.80 | 22.38 | 22.72 | 24.08 | 25.81 | 27.55 | 29.23 | 30.90 |
| iso-Pentane | $C_5H_{12}$ | 72.150 | 24.94 | 27.17 | 27.61 | 29.42 | 31.66 | 33.87 | 36.03 | 38.14 |
| n-Pentane | $C_5H_{12}$ | 72.150 | 25.64 | 27.61 | 28.02 | 29.71 | 31.86 | 33.99 | 36.08 | 38.13 |
| Benzene | $C_6H_6$ | 78.114 | 16.41 | 18.41 | 18.78 | 20.46 | 22.45 | 24.46 | 26.34 | 28.15 |
| n-Hexane | $C_6H_{14}$ | 86.177 | 30.17 | 32.78 | 33.30 | 35.37 | 37.93 | 40.45 | 42.94 | 45.36 |
| n-Heptane | $C_7H_{16}$ | 100.204 | 34.96 | 38.00 | 38.61 | 41.01 | 44.00 | 46.94 | 49.81 | 52.61 |
| Ammonia | $NH_3$ | 17.0305 | 8.52 | 8.52 | 8.52 | 8.52 | 8.52 | 8.53 | 8.53 | 8.53 |
| Air | | 28.9625 | 6.94 | 6.95 | 6.95 | 6.96 | 6.97 | 6.99 | 7.01 | 7.03 |
| Water | $H_2O$ | 18.0153 | 7.98 | 8.00 | 8.01 | 8.03 | 8.07 | 8.12 | 8.17 | 8.23 |
| Oxygen | $O_2$ | 31.9988 | 6.97 | 6.99 | 7.00 | 7.03 | 7.07 | 7.12 | 7.17 | 7.23 |
| Nitrogen | $N_2$ | 28.0134 | 6.95 | 6.95 | 6.95 | 6.96 | 6.96 | 6.97 | 6.98 | 7.00 |
| Hydrogen | $H_2$ | 2.0159 | 6.78 | 6.86 | 6.87 | 6.91 | 6.94 | 6.95 | 6.97 | 6.98 |
| Hydrogen sulfide | $H_2S$ | 34.08 | 8.00 | 8.09 | 8.11 | 8.18 | 8.27 | 8.36 | 8.46 | 8.55 |
| Carbon monoxide | CO | 28.010 | 6.95 | 6.96 | 6.96 | 6.96 | 6.97 | 6.99 | 7.01 | 7.03 |
| Carbon dioxide | $CO_2$ | 44.010 | 8.38 | 8.70 | 8.76 | 9.00 | 9.29 | 9.56 | 9.81 | 10.05 |

* Exceptions: Air — Keenan and Keyes, Thermodynamic Properties of Air, Wiley, 3rd Printing 1947. Ammonia — Edw. R. Grabl, Thermodynamic Properties of Ammonia at High Temperatures and Pressures, Petr. Processing, April 1953. Hydrogen Sulfide — J. R. West, Chem. Eng. Progress, 44, 287, 1948.

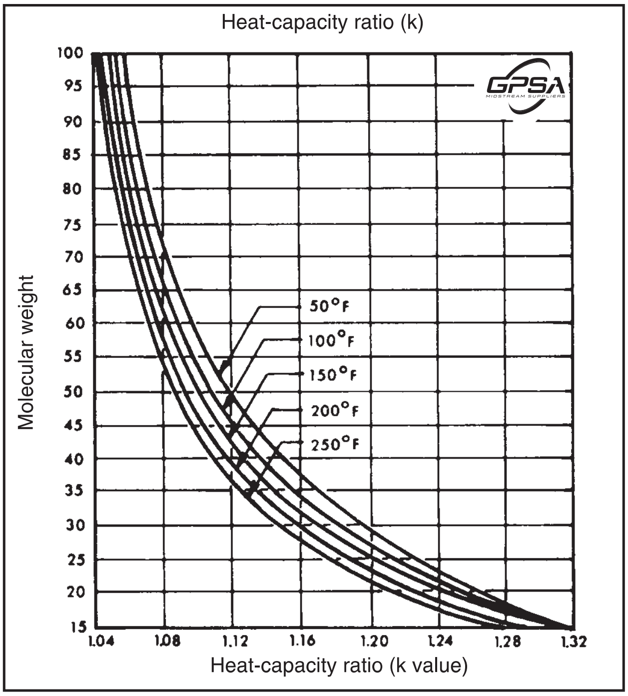
### FIG. 13-7
### Calculation of k

| Example gas mixture | | Determination of mixture mol weight | | Determination of MCp, Molar heat capacity | | Determination of pseudo critical pressure, $pP_c$, and temperature, $pT_c$ | | | |
|---|---|---|---|---|---|---|---|---|---|
| Component name | Mol fraction y | Individual Component Mol weight MW | y • MW | Individual Component $MC_p$ @ 150°F* | y • $MC_p$ @ 150°F | Component critical pressure $P_c$ psia | y • Pc | Component critical temperature $T_c$°R | y • $T_c$ |
| methane | 0.9216 | 16.04 | 14.782 | 8.95 | 8.248 | 666 | 615.6 | 343 | 316.1 |
| ethane | 0.0488 | 30.07 | 1.467 | 13.78 | 0.672 | 707 | 34.6 | 550 | 26.8 |
| propane | 0.0185 | 44.10 | 0.816 | 19.52 | 0.361 | 616 | 11.4 | 666 | 12.3 |
| i-butane | 0.0039 | 58.12 | 0.227 | 25.77 | 0.101 | 528 | 2.1 | 734 | 2.9 |
| n-butane | 0.0055 | 58.12 | 0.320 | 25.81 | 0.142 | 551 | 3.0 | 765 | 4.2 |
| i-pentane | 0.0017 | 72.15 | 0.123 | 31.66 | 0.054 | 490 | 0.8 | 829 | 1.4 |
| Total | 1.0000 | MW = | 17.735 | $MC_p$ = | 9.578 | $pP_c$ = | 667.5 | $pT_c$ = | 363.7 |
| $MC_v = MC_p - 1.986 = 7.592$ | | | | | $k = MC_p/MC_v = 9.578/7.592 = 1.26$ | | | | |

*For values of $MC_p$ other than @ 150°F, refer to Fig. 13-6

**FIG. 13-8**

**Approximate Heat-Capacity Ratios of Hydrocarbon Gases**




range; likewise, use a factor in the range of 16 to 18 for compression ratios between 1.5 and 2.0.

Curves are available which permit easy estimation of approximate compression-horsepower requirements. Fig. 13-9 is typical of these curves.

**Example 13-1** — Compress 2 MMcfd of gas at 14.4 psia and intake temperature through a compression ratio of 9 in a 2-stage compressor. What will be the horsepower?

**Solution Steps**

Ratio per stage = $\sqrt{9} = 3$

From Equation 13-4 we find the brake horsepower to be:

(22) (3) (2) (2) (1.08) = 285 BHP

From Fig. 13-9, using a k of 1.15, we find the horsepower requirement to be 136 BHP/MMcfd or 272 BHP. For a k of 1.4, the power requirement would be 147 BHP/MMcfd or 294 total horsepower.

The two procedures give reasonable agreement, particularly considering the simplifying assumptions necessary in reducing compressor horsepower calculations to such a simple procedure.

## Detailed Calculations

There are many variables which enter into the precise calculation of compressor performance. Generalized data as given in this section are based upon the averaging of many criteria. The results obtained from these calculations, therefore, must be considered as close approximations to true compressor performance.

## Capacity

Most gases encountered in industrial compression do not exactly follow the ideal gas equation of state but differ to varying degrees. The degree in which any gas varies from the ideal is expressed by a compressibility factor, Z, which modifies the ideal gas equation:

$$PV = nRT \qquad \textbf{Eq 13-5}$$

to

$$PV = nZRT \qquad \textbf{Eq 13-6}$$

Compressibility factors can be determined from charts in Section 23 using the $pP_R$ and $pT_R$ of the gas mixture. For pure components such as propane, compressibility factors can be determined from the P-H diagrams, although the user would be better advised to determine the compression horsepower using the P-H diagram (see Section 24).

For the purpose of performance calculations, compressor capacity is expressed as the actual volumetric quantity of gas at the inlet to each stage of compression on a per minute basis (ICFM).

From SCFM

$$Q = \text{SCFM}\left(\frac{14.7}{520}\right)\left(\frac{T_1\,Z_1}{P_1\,Z_L}\right) \qquad \textbf{Eq 13-7}$$

From weight flow (w, lb/min)

$$Q = \frac{10.73}{MW}\left(\frac{wT_1\,Z_1}{P_1\,Z_L}\right) \qquad \textbf{Eq 13-8}$$

From molar flow ($N_m$, mols/min)

$$Q = \left(\frac{379.5 \cdot 14.7}{520}\right)\left(\frac{N_m T_1\,Z_1}{P_1\,Z_L}\right) \qquad \textbf{Eq 13-9}$$

From these equations, inlet volume to any stage may be calculated by using the inlet pressure $P_1$ and temperature $T_1$. Moisture should be handled just as any other component in the gas.

In a reciprocating compressor, effective capacity may be calculated as the piston displacement (generally in cu ft/min) multiplied by the volumetric efficiency.

The piston displacement is equal to the net piston area multiplied by the length of piston sweep in a given period of time. This displacement may be expressed:

For a single-acting piston compressing on the outer end only,

$$PD = \frac{(\text{stroke})\,(N)\,(D^2)\,\pi}{(4)\cdot(1728)} \qquad \textbf{Eq 13-10}$$

$$= 4.55\,(10^{-4})\,(\text{stroke})\,(N)\,(D^2)$$

For a single-acting piston compressing on the crank end only,

$$PD = \frac{(\text{stroke})\,(N)\,(D^2 - d^2)\,\pi}{(4)\cdot(1728)} \qquad \textbf{Eq 13-11}$$

$$= 4.55\,(10^{-4})\,(\text{stroke})\,(N)\,(D^2 - d^2)$$

For a double-acting piston (other than tail rod type),

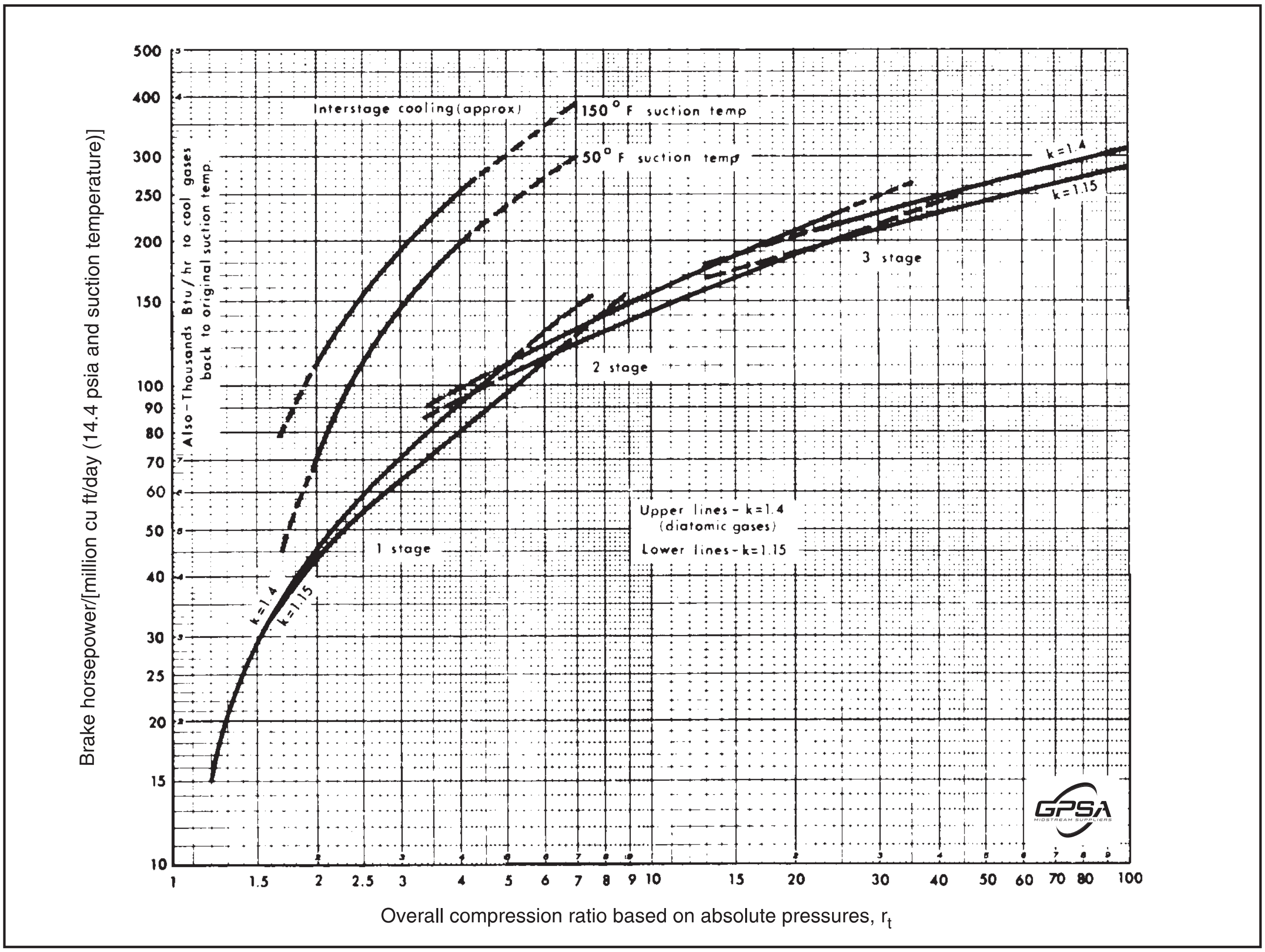
$$PD = \frac{(\text{stroke})\,(N)\,(2\,D^2 - d^2)\,\pi}{(4)\cdot(1728)} \qquad \textbf{Eq 13-12}$$

$$= 4.55\,(10^{-4})\,(\text{stroke})\,(N)\,(2\,D^2 - d^2)$$

## Volumetric Efficiency

In a reciprocating compressor, the piston does not travel completely to the end of the cylinder at the end of the discharge stroke. Some clearance volume is necessary and it includes the space between the end of the piston and the cylinder head when

**FIG. 13-9**
**Approximate Horsepower Required to Compress Gases**




the piston is at the end of its stroke. It also includes the volume in the valve ports, the volume in the suction valve guards, and the volume around the discharge valve seats.

Clearance volume is usually expressed as a percent of piston displacement and referred to as percent clearance, or cylinder clearance, C.

$$C = \frac{\text{clearance volume, cu in.}}{\text{piston displacement, cu in.}} (100) \qquad \textbf{Eq 13-13}$$

For double acting cylinders, the percent clearance is based on the total clearance volume for both the head end and the crank end of a cylinder. These two clearance volumes are not the same due to the presence of the piston rod in the crank end of the cylinder. Sometimes additional clearance volume (external) is intentionally added to reduce cylinder capacity.

The term "volumetric efficiency" refers to the actual pumping capacity of a cylinder compared to the piston displacement. Without a clearance volume for the gas to expand and delay the opening of the suction valve(s), the cylinder could deliver its entire piston displacement as gas capacity. The effect of the gas contained in the clearance volume on the pumping capacity of a cylinder can be represented by:

$$VE = 100 - r - C\left[\frac{Z_s}{Z_d}\left(r^{1/k}\right) - 1\right] \qquad \textbf{Eq 13-14}$$

Volumetric efficiencies as determined by Equation 13-14 are theoretical in that they do not account for suction and discharge valve losses. The suction and discharge valves are actually spring-loaded check valves that permit flow in one direction only. The valve springs require a small differential pressure to open. For this reason, the pressure within the cylinder at the end of the suction stroke is lower than the line suction pressure and, likewise, the pressure at the end of the discharge stroke is higher than line discharge pressure.

One method for accounting for suction and discharge valve losses is to reduce the volumetric efficiency by an arbitrary amount, typically 4%, thus modifying Equation 13-14 as follows:

$$VE = 96 - r - C\left[\frac{Z_s}{Z_d}\left(r^{1/k}\right) - 1\right] \qquad \textbf{Eq 13-15}$$

When a non-lubricated compressor is used, the volumetric efficiency should be corrected by subtracting an additional 5% for slippage of gas. This is a capacity correction only and, as a first approximation, would not be considered when calculating compressor horsepower. The energy of compression is used by the gas even though the gas slips by the rings and is not discharged from the cylinder.

If the compressor is in propane, or similar heavy gas service, an additional 4% should be subtracted from the volumetric efficiency. These deductions for non-lubricated and propane performance are both approximate and, if both apply, cumulative.

Volumetric efficiencies for "high speed" separable compressors in the past have tended to be slightly lower than estimated from Equation 13-14. Recent information suggests that this modification is not necessary for all models of high speed compressors.

In evaluating efficiency, horsepower, volumetric efficiency, etc., the user should consider past experience with different speeds and models. Larger valve area for a given swept volume will generally lead to higher compression efficiencies.

## Equivalent Capacity

The net capacity for a compressor, in cubic feet per day @ 14.4 psia and suction temperature, may be calculated by Equation 13-16a which is shown in dimensioned form:

$$\text{MMcfd} = \frac{\left[\text{PD}\,\frac{\text{ft}^3}{\text{min}}\right] \cdot 1440\,\frac{\text{min}}{\text{d}} \cdot \left[\frac{\text{VE\%}}{100}\right] \cdot \text{P}_s\,\frac{\text{lb}}{\text{in}^2} \cdot 10^{-6}\,\frac{\text{MMft}^3}{\text{ft}^3} \cdot \text{Z}_{14.4}}{14.4\,\frac{\text{lb}}{\text{in}^2} \cdot \text{Z}_s} \qquad \textbf{Eq 13-16a}$$

which can be simplified to Equation 13-16b when $Z_{14.4}$ is assumed to equal 1.0.

$$\text{MMcfd} = \frac{\text{PD} \cdot \text{VE} \cdot \text{P}_s \cdot 10^{-6}}{\text{Z}_s} \qquad \textbf{Eq 13-16b}$$

For example, a compressor with 200 cu ft/min piston displacement, a volumetric efficiency of 80%, a suction pressure of 75 psia, and suction compressibility of 0.9 would have a capacity of 1.33 MMcfd at 14.4 psia. If compressibility is not used as a divisor in calculating cu ft/min, then the statement "not corrected for compressibility" should be added.

In many instances the gas sales contract or regulation will specify some other measurement standard for gas volume. To convert volumes calculated using Equation 13-16 (i.e. at 14.4 psia and suction temperature) to a $P_L$ and $T_L$ basis, Equation 13-17 would be used:

$$\text{MMscfd at P}_L\text{, T}_L = (\text{MMcfd from Eq 13-16}) \left(\frac{14.4}{\text{P}_L}\right)\left(\frac{\text{T}_L}{\text{T}_s}\right)\left(\frac{\text{Z}_L}{\text{Z}_s}\right) \qquad \textbf{Eq 13-17}$$

## Discharge Temperature

The temperature of the gas discharged from the cylinder can be estimated from Equation 13-18, which is commonly used but not recommended. (Note: the temperatures are in absolute units, °R or K.) Equation 13-32 gives better results.

$$T_d = T_s\,(r^{(k-1)/k}) \qquad \textbf{Eq 13-18}$$

The discharge temperature determined from Equation 13-18 is the theoretical value. It neglects heat from friction, irreversibility effects, etc., and is therefore too low,

## Rod Loading

Each compressor frame has definite limitations as to maximum load-carrying capacity. The load-carrying of a compressor involves two primary considerations: rod loading and horsepower.

The horsepower rating of a compressor frame is an indicator of the supporting structure and crankshaft to withstand the torque (turning force) and the loads. Rod loads are established to limit the static and dynamic loads on the frame, crankshaft, connecting rod, frame, crosshead, piston rod, bolting, and projected bearing surfaces.

Rod loads are calculated differently based upon the compressor manufacturer. Some manufacturers use flange-to-flange pressures while others use internal pressures and others may use combined rod loads (gas load plus inertia load).

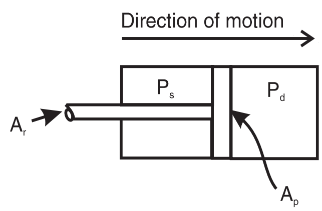
Many manufacturers also require a load reversal of the load at the crosshead pin. This load reversal is required so that lube oil can lubricate and cool the crosshead pin and bushings.

Gas rod loadings may be calculated by the use of Equations 13-19 and 13-20.

$$\begin{aligned}\text{Load in compression} &= P_d A_p - P_s (A_p - A_r)\\ &= (P_d - P_s) A_p + P_s A_r \qquad \textbf{Eq 13-19}\end{aligned}$$

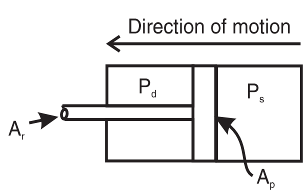
$$\begin{aligned}\text{Load in tension} &= P_d (A_p - A_r) - P_s A_p\\ &= (P_d - P_s) A_p - P_d A_r \qquad \textbf{Eq 13-20}\end{aligned}$$

**Rod in Compression**




Using Equations 13-19 and 13-20, a plus value for the load in both compression and tension indicates a reversal of loads based on gas pressure only. Inertial effects will tend to increase the degree of reversal.

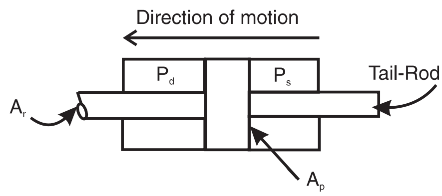
**Rod in Tension**




The true rod loads would be those calculated using internal cylinder pressures after allowance for valve losses. Normally, the operator will know only line pressures, and because of this, manufacturers generally rate their compressors based on line-pressure calculations.

A further refinement in the rod-loading calculation would be to include inertial forces. While the manufacturer may consider inertial forces when rating compressors, useful data on this point is seldom available in the field. Except in special cases, inertial forces are ignored.

A tail-rod cylinder would require consideration of rod cross-section area on both sides of the piston instead of on only one side of the piston, as in Equations 13-19 and 13-20.




## Detailed Horsepower Calculation

A more detailed calculation of reciprocating compressor power requirements can be performed using the following equation:

$$\text{BHP/stage} = 3.03 \cdot Z_{avg} \cdot [Q_g T_s/E] \cdot (k/(k\text{-}1)) \cdot \left(\frac{P_L}{T_L}\right) \cdot [(P_d/P_s)^{((k-1)/k)} - 1] \qquad \textbf{Eq 13-21}$$

The total horsepower for the compressor is the sum of the horsepower required for each of the stages that are utilized. For multistage machines an allowance should be made for the interstage pressure drop associated with piping, cooler, scrubber, etc., typically 5–10 psi.

**Procedure**

1. Calculate overall compression ratio ($r_t = P_{dfinal}/P_s$).
2. Calculate the compression ratio per stage, r, by taking the s root of $r_t$, where s is the number of compression stages. The number of stages, s, should be increased until the ratio per stage, r, is < ~ 4. This should generally result in stage discharge temperatures of < 300°F depending on the interstage cooler outlet temperature assumed.
3. Multiplying r by the absolute suction pressure of the stage being considered will give you discharge pressure of the stage.
4. Calculate the horsepower required for the stage using Equation 13-21.
5. Subtract the assumed interstage pressure loss from the discharge pressure of the preceding stage to obtain the suction pressure for the next stage.
6. Repeat steps 4 and 5 until all stages have been calculated.
7. Sum the stage horsepowers to obtain the total compressor power required.

**Example 13-2** — Compress 2 MMscfd of gas measured at 14.65 psia and 60°F. Intake pressure is 100 psia, and intake temperature is 100°F. Discharge pressure is 900 psia. The gas has a specific gravity of 0.80 (23 MW). What is the required brake horsepower, assuming a high speed compressor?

Assume E = 0.82

1. Compression ratio is

$$\frac{900 \text{ psia}}{100 \text{ psia}} = 9$$

This would be a two-stage compressor; therefore, the ratio per stage is $\sqrt{9}$ or 3.

2. 100 psia x 3 = 300 psia (1st stage discharge pressure). Suction pressure to second stage is given by

300 psia – 5 = 295 psia

Where the 5 psi represents the pressure drop between first stage discharge and second stage suction.

$$\frac{900 \text{ psia}}{295 \text{ psia}} = 3.05 \text{ (compression ratio for 2nd stage)}$$

It may be desirable to recalculate the interstage pressure to balance the ratios. For this sample problem, however, the first ratios determined will be used.

3. From Fig. 13-8 a gas with specific gravity of 0.8 at 150°F would have an approximate k of 1.21. For most compression applications, the 150°F curve will be adequate. This should be checked after determining the average cylinder temperature.
4. Discharge temperature for the 1st stage may be obtained by using Fig. 13-32 or solving Equation 13-18. For a compression ratio of 3, discharge temperature = approximately 220°F. Average cylinder temperature = 160°F.
5. In the same manner, discharge temperature for the second stage (with r = 3.05 and assuming interstage cooling to 120°F) equals approximately 244°F. Average cylinder temperature = 182°F.
6. From the physical properties section (Section 23), estimate the compressibility factors at suction and discharge pressure and temperature of each stage.

1st stage: $Z_s$ = 0.98
$Z_d$ = 0.97
$Z_{avg}$ = 0.975

2nd stage: $Z_s$ = 0.94
$Z_d$ = 0.92
$Z_{avg}$ = 0.93

7. Calculate the horsepower required for the first and second stages from Equation 13-21:

$$\text{BHP for 1st stage} = 3.03 \cdot (0.975) \cdot [2 \cdot 560/0.82] \cdot [1.21/(1.21 - 1)] \cdot \left(\frac{14.65}{520}\right) \cdot [(300/100)^{((1.21-1)/1.21)} - 1] = 137.6$$

$$\text{BHP for 2nd stage} = 3.03 \cdot (0.93) \cdot [2 \cdot 580/0.82] \cdot [1.21/(1.21 - 1)] \cdot \left(\frac{14.65}{520}\right) \cdot [(900/295)^{((1.21-1)/1.21)} - 1] = 138.2$$

Total BHP required = 137.6 + 138.2 = 275.8

Note that in Example 13-1 the same conditions result in a compression power of 285 BHP which is close agreement.

**Limits to compression ratio per stage** — The maximum ratio of compression permissible in one stage is usually limited by the discharge temperature or by rod loading.

When handling gases containing oxygen, which could support combustion, there is a possibility of fire and explosion because of the oil vapors present.

To reduce carbonization of the oil and the danger of fires, a safe operating limit may be considered to be approximately 300°F. Where no oxygen is present in the gas stream, temperatures of 350°F may be considered as the maximum, even though mechanical or process requirements usually dictate a lower figure.

Packing life may be significantly shortened by the dual requirement to seal both high pressure and high temperature gases. For this reason, at higher discharge pressures, a temperature closer to 250°F or 275°F may be the practical limit.

In summary, and for most field applications, the use of 300°F maximum would be a good average. Recognition of the above variables is, however, still useful.

Economic considerations are also involved because a high ratio of compression will mean a low volumetric efficiency and require a larger cylinder to produce the same capacity. For this reason a high rod loading may result and require a heavier and more expensive frame.

Where multi-stage operation is involved, equal ratios of compression per stage are used (plus an allowance for piping and cooler losses if necessary) unless otherwise required by process design. For two stages of compression the ratio per stage would approximately equal the square root of the total compression ratio; for three stages, the cube root, etc. In practice, especially in high-pressure work, decreasing the compression ratio in the higher stages to reduce excessive rod loading may prove to be advantageous.

## Cylinder Design

Depending on the size of the machine and the number of stages, reciprocating compressors are furnished with cylinders fitted with either single- or double-acting pistons, see examples in Figs. 13-10 through 13-12.

In the same units, double-acting pistons are commonly used in the first stages and occasionally single-acting in the higher stages of compression.

Cylinder materials are normally selected for strength; however, thermal shock, mechanical shock, or corrosion resistance may also be a determining factor. The table below shows discharge pressure limits generally used in the gas industry for cylinder material selection.

| Cylinder Material | Discharge Pressure (psig) |
|---|---|
| Cast Iron | up to 1,200 |
| Nodular Iron | about 2,500 |
| Cast Steel | 1,200 to 3,000 |
| Forged Steel | above 2,500 |

API standard 618 recommends 1000 psig as the maximum pressure for both cast iron and nodular iron.

Cylinders are designed both as a solid body (no liner) and with liners. Cylinder liners are inserted into the cylinder body to either form or line the pressure wall. There are two types. The wet liner forms the pressure wall as well as the inside wall of the water jacket. The dry type lines the cylinder wall and is not required to add strength.

Standard cylinder liners are cast iron. If cylinders are required to have special corrosion or wear resistance, other materials or special alloys may be needed.

Most compressors use oils to lubricate the cylinder with a mechanical, force-feed lubricator having one or more feeds to each cylinder.

The non-lubricated compressor has found wide application where it is desirable or essential to compress air or gas without contaminating it with lubricating oil.

For such cases a number of manufacturers furnish a "non-lubricated" cylinder (Fig. 13-13). Non-metallic packing seal rings of a type that requires no lubricant is used on the stuffing box. Although oilwiper rings are used on the piston rod where it leaves the compressor frame, minute quantities of oil might conceivably enter the cylinder on the rod. Where even such small amounts of oil are objectionable, an extended cylinder connecting piece can be furnished. This simply lengthens the piston rod so that no lubricated portion of the rod enters the cylinder.

A small amount of gas leaking through the packing can be objectionable. Special distance pieces are furnished between the cylinder and frame, which may be either single-compartment or double-compartment. These may be furnished gas tight and vented back to the suction, or may be filled with a sealing gas or fluid and held under a slight pressure, or simply vented.

Compressor valves for non-lubricated service operate in an environment that has no lubricant in the gas or in the cylinder. Therefore, the selection of valve materials is important to prevent excessive wear.

Piston rod packing universally used in non-lubricated compressors is of the full-floating mechanical type, consisting of a case containing pairs of non-metallic rings of conventional design.

When handling oxygen and other gases such as nitrogen and helium, it is absolutely necessary that all traces of hydrocarbons in cylinders be removed. With oxygen, this is required for safety, with other gases to prevent system contamination.

High-pressure compressors with discharge pressures from 5,000 to 30,000 psi usually require special design and a complete knowledge of the characteristics of the gas.

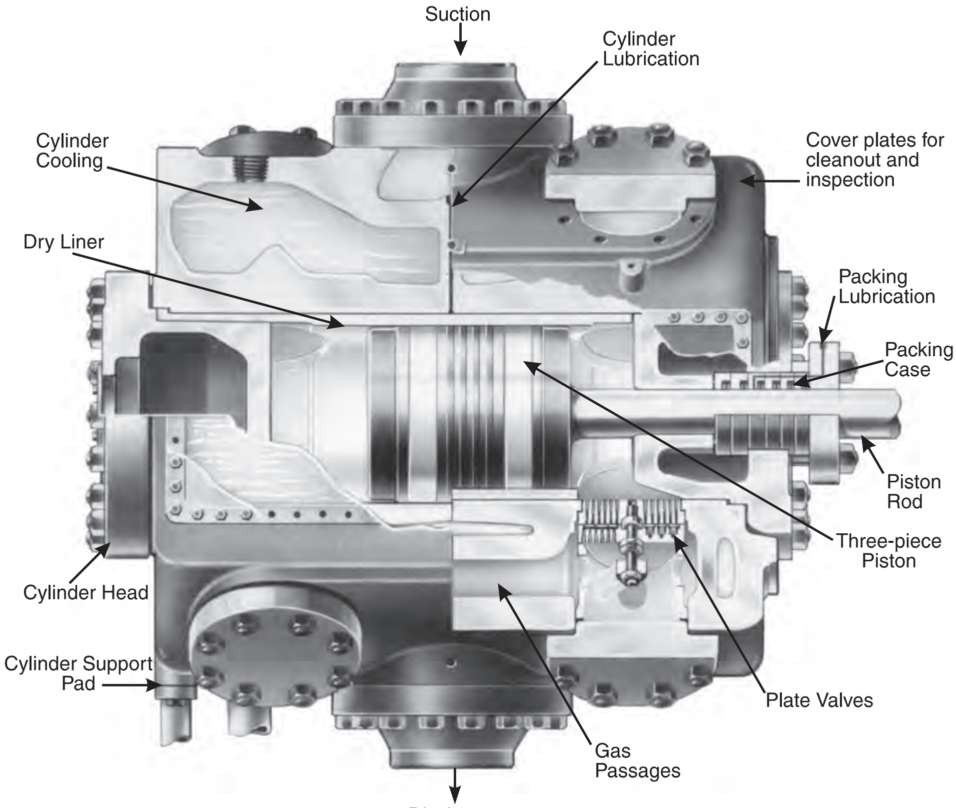
As a rule, inlet and discharge gas pipe connections on the cylinder are fitted with flanges of the same rating for the following reasons:

- Practicality and uniformity of casting and machinery,
- Hydrostatic test, usually at 150% design pressure,
- Suction pulsation bottles are usually designed for the same pressure as the discharge bottle (often federal, state, or local government regulation).

## Reciprocating Compressor Control Devices

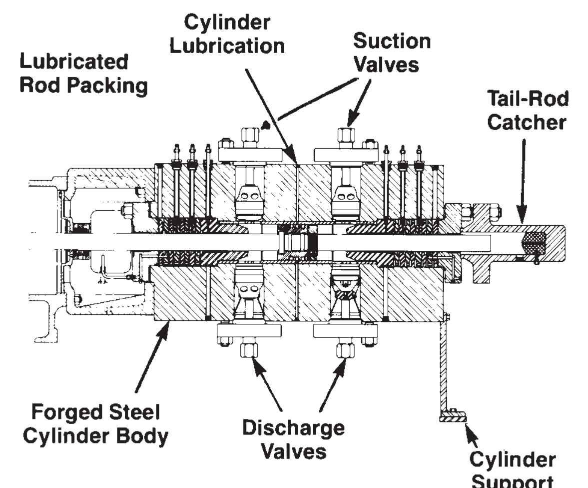
Output of compressors must be controlled (regulated) to match system demand.

**FIG. 13-10**
**Low Pressure Cylinder with Double-Acting Piston**



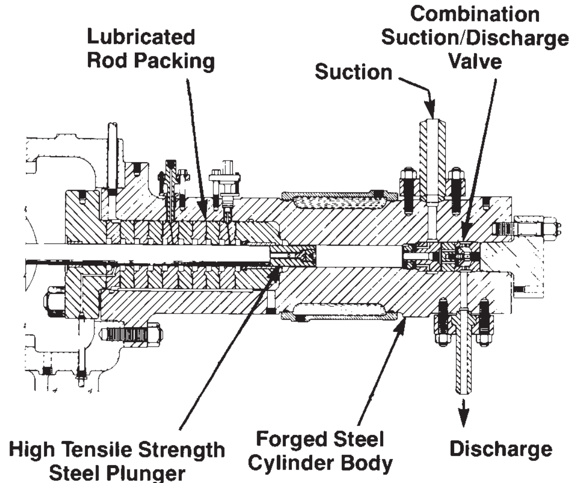

*Courtesy Cooper Industries*

**FIG. 13-11**
**High Pressure Cylinder with Double-Acting Piston and Tail-Rod**




Courtesy of Ingersoll-Rand Co.

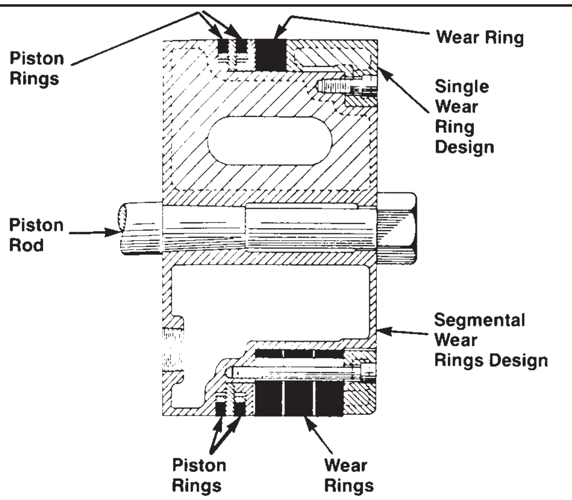
**FIG. 13-12**
**Single-Acting Plunger Cylinder Designed for 15,000 psig Discharge**




Courtesy of Ingersoll-Rand Co.

FIG. 13-13

**Piston Equipped with Teflon® Piston and Wear Rings for a Single-acting Non-Lubricated Cylinder**




Courtesy of McGraw-Hill Book Co.

In many installations some means of controlling the output of the compressor is necessary. Often constant flow or a specific power is required despite variations in operating conditions. Compressor capacity, speed, or pressure may be varied in accordance with the requirements. The nature of the control device will depend on the regulating variable — whether pressure, flow, temperature, or some other variable — and on type of compressor driver.

**Unloading for Starting** — Practically all reciprocating compressors must be unloaded to some degree before starting so that the driver torque available during acceleration is not exceeded. Both manual and automatic compressor startup unloading is used. Common methods of unloading include: discharge venting, discharge to suction bypass, and holding open the inlet valves using valve lifters.

**Capacity Control** — Capacity control is required to either regulate capacity or maintain the compressor load within the driver rating. Capacity control devices/unloading devices can be manually actuated or actuated by air or gas pressure depending on their design. A falling pressure indicates that gas is being used faster than it is being compressed and that more gas is required. A rising pressure indicates that more gas is being compressed than is being used downstream and that less gas is required.

A common method of controlling the capacity of a compressor is to vary the speed. This method is applicable to variable frequency drive (VFD) electric motor driven compressors and to units driven by internal combustion engines. In these cases the regulator actuates the VFD controller or fuel-admission valve on the compressor driver to control the speed.

Electric motor-driven compressors usually operate at constant speed, although variable speed drives are becoming increasingly more common. For constant speed motors other methods of controlling the capacity are necessary. On reciprocating compressors up to about 100 hp, two types of control are usually available. These are automatic-start-and-stop control and constant-speed control.

Automatic-start-and-stop control, as its name implies, stops or starts the compressor by means of a pressure-actuated switch as the gas demand varies. It should be used only when the demand for gas will be intermittent.

Constant-speed control permits the compressor to operate at full speed continuously, but loaded part of the time and fully or partially unloaded at other times. Two methods of unloading the compressor with this type of control are in common use: inlet-valve unloaders, and clearance unloaders. Inlet-valve unloaders (Fig. 13-14) operate to hold the compressor inlet valves open and thereby prevent compression. Clearance unloaders (Fig. 13-15) consist of pockets or small reservoirs which are opened when unloading is desired. The gas is compressed into them on the compression stroke and expands back into the cylinder on the return stroke, reducing the intake of additional gas.

Motor-driven reciprocating compressors above 100 hp in size are usually equipped with a step control. This is in reality a variation of constant-speed control in which unloading is accomplished in a series of steps, varying from full load down to no load.

Five-step control (full load, three-quarter load, one-half load, one-quarter load, and no load) is accomplished by means of clearance pockets. On some makes of machines inlet-valve and clearance control unloading are used in combination.

A common practice in the natural gas industry is to prepare a single set of curves for a given machine unless there are side loads or it is a multi-service machine.

Fig. 13-16 shows indicator cards which demonstrate the unloading operation for a double acting cylinder at three capacity points. The letters adjacent to the low-pressure diagrams represent the unloading influence of the respective and cumulative effect of the various pockets as identified in Fig. 13-15. Full load, one-half, and no load capacity (used for start-up only) is obtained by holding corresponding suction valves open or adding sufficient clearance to produce a zero volumetric efficiency. Zero-capacity operation includes holding all suction valves open.

Fig. 13-17 shows an alternative representation of compressor unloading operation with a step-control using fixed volume clearance pockets. The curve illustrates the relationship between compressor capacity and driver capacity for a varying compressor suction pressure at a constant discharge pressure and constant speed. The driver can be a gas engine or electric motor.

The purpose of this curve is to determine what steps of unloading are required to prevent the driver and piston rods from serious overloading. All lines are plotted for a single stage compressor.

The driver capacity line indicates the maximum allowable capacity for a given horsepower. The cylinder capacity lines represent the range of pressures calculated with all possible combinations of pockets open, as necessary, to cover the capacity of the driver.

Starting at the end (line 0-0) with full cylinder capacity, the line is traced until it crosses the driver capacity line at which point it is dropped to the next largest cylinder capacity and follow until it crosses the driver line, etc. This will produce a "saw tooth" effect, hence the name "saw tooth" curve. The number of "teeth" depends upon the number of combinations of pockets (opened or closed) required for unloading. If suction valves were also unloaded then there would be more "teeth" on the curve.

The same method is followed for multi-stage units. For each additional stage another “saw tooth” curve must be constructed, i.e., for a two stage application, two curves are required to attain the final results.

Although control devices are often automatically operated, manual operation is satisfactory for many services. Where manual operation is provided, it often consists of a valve, or valves, to open and close clearance pockets. In some cases, a movable cylinder head is provided for variable clearance in the cylinder (Fig. 13-18).

## Gas Pulsation Control

Pulsation is inherent in reciprocating compressors because suction and discharge valves are open during only part of the stroke.

Pulsation must be damped (minimized) in order to:

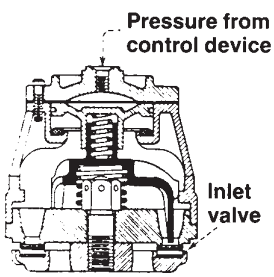
a. provide smooth flow of gas to and from the compressor,

b. prevent overloading or underloading of the compressors, and

c. reduce overall vibration.

**FIG. 13-14**

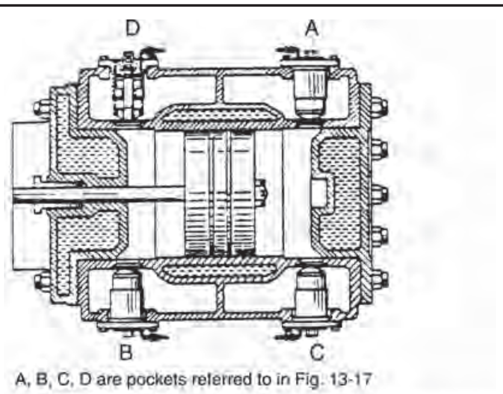
**Inlet Valve Unloader**




Courtesy of McGraw-Hill Book Co.

**FIG. 13-15**

**Pneumatic Actuated Valves Controlling Four Fixed Pockets in Compressor for Five-Step Control**

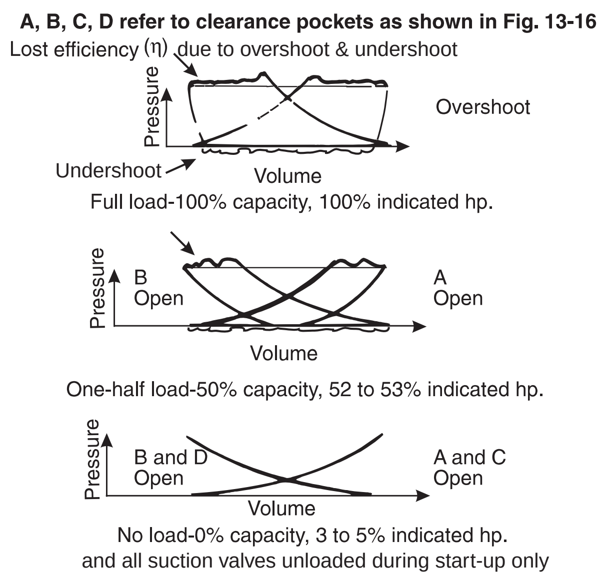



*Courtesy McGraw-Hill Book Co.*

There are several types of pulsation chambers. The simplest one is a volume bottle, or a surge drum, which is a pressure vessel, unbaffled internally and mounted on or very near a cylinder inlet or outlet.

**FIG. 13-16**

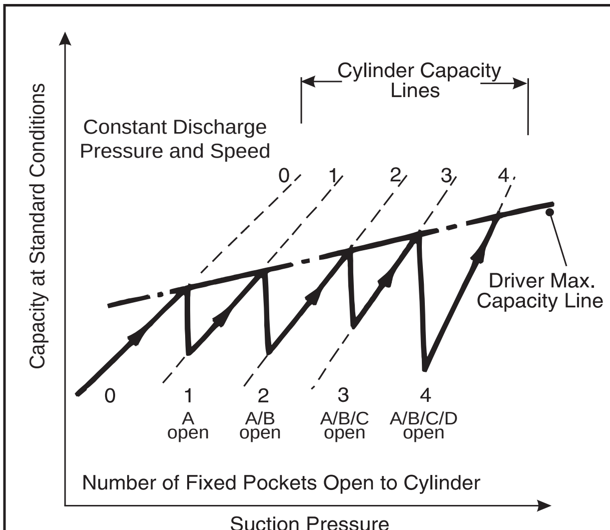
**Indicator Diagram for Three Load Points of Operation**




*Courtesy McGraw-Hill Book Co.*

**FIG. 13-17**

**“Saw Tooth” Curve for Unloading Operation**

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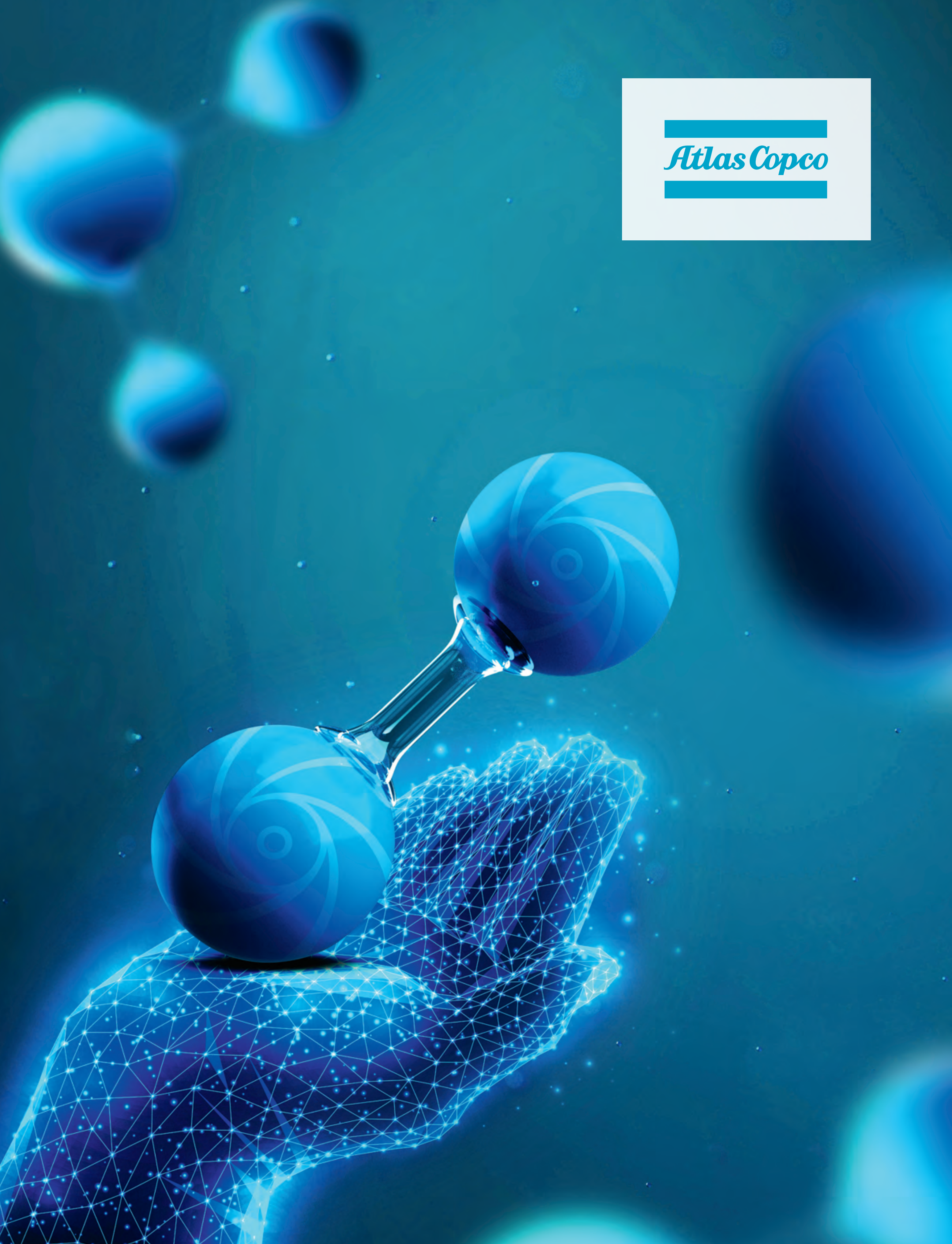

Atlas Copco

A manifold joining the inlet and discharge connections of cylinders operating in parallel can also serve as a volume bottle.

Performance of volume bottles is not normally guaranteed without an analysis of the piping system from the compressor to the first process vessel.

Volume bottles are sized empirically to provide an adequate volume to absorb most of the pulsation. Several industry methods were tried in an effort to produce a reasonable rule-of-thumb for their sizing. Fig. 13-19 may be used for approximate bottle sizing.

**Example 13-3**

Indicated suction pressure = 600 psia

Indicated discharge pressure = 1400 psia

Cylinder bore = 6 in

Cylinder stroke = 15 in

Swept volume = $\pi$ ($6^2$/4) (15) = 424 cu in

From Fig. 13-19:

At 600 psi inlet pressure, the suction bottle multiplier is approximately 7.5. Suction-bottle volume = (7.5) (424) = 3,180 cu in.

NOTE: When more than one cylinder is connected to a bottle, the sum of the individual swept volumes is the size required for the common bottle.

For more accurate sizing, compressor manufacturers can be consulted. Organizations which provide designs and/or equipment for gas-pulsation control are also available.

Having determined the necessary volume of the bottle, the proportioning of diameter and length to provide this volume requires some ingenuity and judgment. It is desirable that manifolds be as short and of as large diameter as is consistent with pressure conditions, space limitations, and appearance.

A good general rule is to make the manifold diameter 1-1/2 times the inside diameter of the largest cylinder connected to it, but this is not always practicable, particularly where large cylinders are involved.

Inside diameter of pipe must be used in figuring manifolds. This is particularly important in high-pressure work and in small sizes where wall thickness may be a considerable percentage of the cross sectional area. Minimum manifold length is

**FIG. 13-18**

**Sectional View of a Cylinder Equipped with a Hand-Operated Valve Lifter and Variable-Volume Clearance**

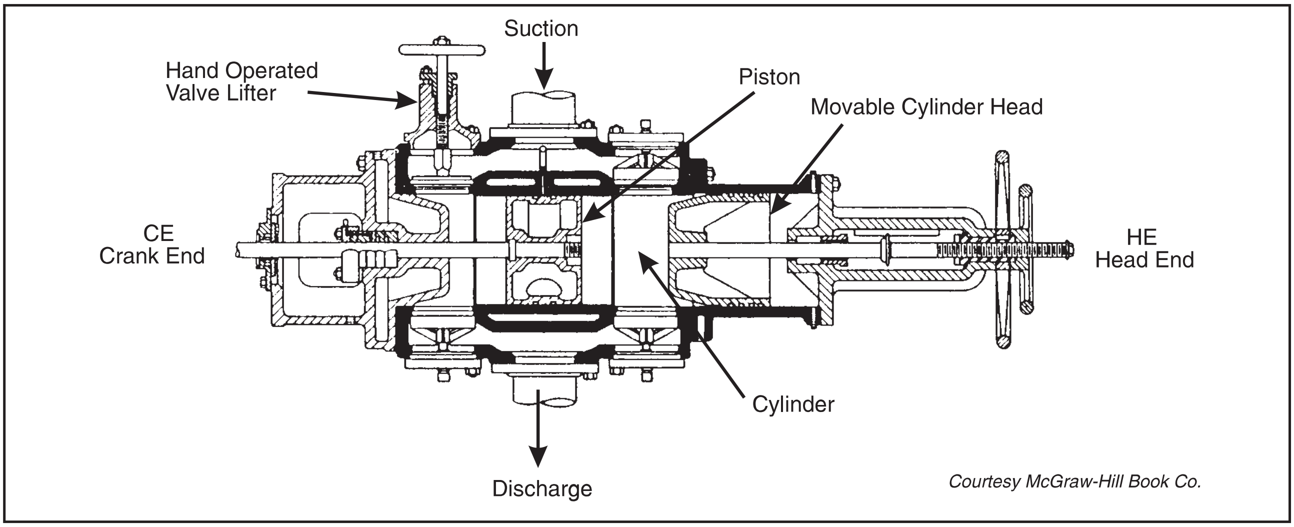
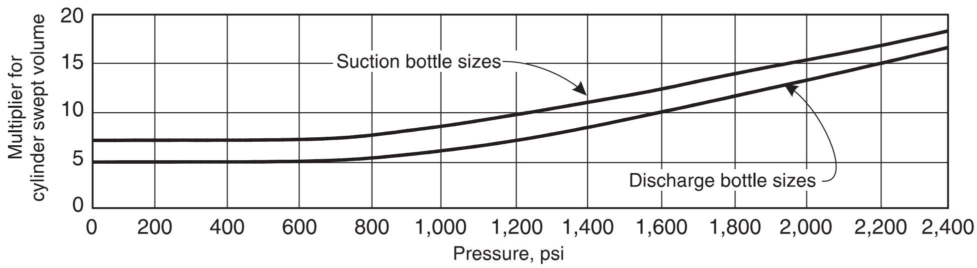



**FIG. 13-19**

**Approximate Bottle Sizing Chart**



determined from cylinder center distances and connecting pipe diameters. Some additions must be made to the minimum thus determined to allow for saddle reinforcements and for welding of caps.

It is customary to close the ends of manifolds with welding caps which add both volume and length. Fig. 13-20 gives approximate volume and length of standard caps.

## Pulsation Dampeners (Snubbers)

A pulsation dampener is an internally-baffled device. The design of the pulsation dampening equipment is based on an acoustical study which takes into account the specified operating speed range, conditions of unloading, and variations in gas composition.

Analog evaluation is accomplished with an active analog that simulates the entire compressor, pulsation dampeners, piping and equipment system and considers dynamic interactions among these elements.

Pulsation dampeners also should be mounted as close as possible to the cylinder, and in large volume units, nozzles should be located near the center of the chamber to reduce unbalanced forces.

Pulsation dampeners are typically guaranteed for a maximum residual peak-to-peak pulsation pressure of 2% of average absolute pressure at the point of connection to the piping system, and pressure drop through the equipment of not more than 1% of the absolute pressure. This applies at design condition and not necessarily for other operating pressures and flows. A detailed discussion of recommended design approaches for pulsation suppression devices is presented in API Standard 618, Reciprocating Compressors for General Refinery Services.

As pressure vessels, all pulsation chambers (volume bottles and dampeners) are generally built to Section VIII of ASME Code and suitable for applicable cylinder relief valve set pressure.

Suction pulsation chambers are often designed for the same pressure as the discharge units, or for a minimum of 2/3 of the design discharge pressure.

## Torsional Analysis

All rotating equipment experiences a torsional load. Examples of torsional loads are:

- inertia and gas loads from the pistons in a reciprocating compressor or engine; or
- the torque fluctuations from a synchronous motor during startup.

A complete drive train (for example reciprocating compressor, coupling, and an electric motor) will have torsional natural frequencies. Those torsional natural frequencies are analogous to mechanical natural frequencies of piping or the compressor shaft. For the mechanical natural frequency, the deflection occurs in a horizontal or vertical direction. For a torsional natural frequency, the deflection is a twisting about the axis of the shaft. Consider fixing one end of a shaft and twisting the free end; when released the shaft will rotate back and forth. The frequency of the oscillation is the torsional natural frequency.

When the system is started and the compressor is loaded what is the impact? Typically the torsional loads will happen at run speed and harmonics. If a torsional natural frequency occurs near a frequency where there is significant torsional energy, the results can be catastrophic! Destroyed coupling, broken compressor shaft or broken motor shaft are potential consequences of torsional resonance. The cost of repair can be large, but the downtime and cost of having the unit unavailable for 1-2 months is often much larger.

A torsional failure will typically occur without warning. The vibration sensors installed at bearings or on component frames are designed to detect lateral (horizontal or vertical) vibrations, and will not detect torsional problems. The best insurance against a torsional failure is a design study before a unit is built.

A design study will consider each component and the role it plays in the torsional system. Manufacturing tolerances, installation differences, and loading all play a critical part in the system's ability to operate without failure. As well as the normal operating loads, the torsional analysis should consider the loads of other operating scenarios, such as compressor valve failures (upset) or the unit startup (transient).

**FIG. 13-20**

**Welding Caps**

| Pipe size | Standard weight | | Extra strong | | Double Extra strong | |
|---|---|---|---|---|---|---|
| | Volume, cu in. | Length, in. | Volume, cu in. | Length, in. | Volume, cu in. | Length, in. |
| 4" | 24.2 | $2\frac{1}{2}$ | 20.0 | $2\frac{1}{2}$ | 15 | 3 |
| 6" | 77.3 | $3\frac{1}{2}$ | 65.7 | $3\frac{1}{2}$ | 48 | 4 |
| 8" | 148.5 | $4\frac{11}{16}$ | 122.3 | $4\frac{11}{16}$ | 120 | 5 |
| 10" | 295.6 | $5\frac{3}{4}$ | 264.4 | $5\frac{3}{4}$ | | |
| 12" | 517.0 | $6\frac{7}{8}$ | 475.0 | $6\frac{7}{8}$ | | |
| 14" | 684.6 | $7\frac{13}{16}$ | 640.0 | $7\frac{13}{16}$ | | |
| 16" | 967.6 | 9 | 911.0 | 9 | | |
| 18" | 1432.6 | $10\frac{1}{16}$ | 1363.0 | $10\frac{1}{16}$ | | |
| 20" | 2026.4 | $11\frac{1}{4}$ | 1938.0 | $11\frac{1}{4}$ | | |
| 24" | 3451.0 | $13\frac{7}{16}$ | 3313.0 | $13\frac{7}{16}$ | | |

With early involvement by designers, the system can be modified. Changing the coupling size or style, adding a flywheel, changing shaft material or size are all easily done if the components have not been built. Modifications to a system that is already built can be expensive and may require re-design. If this occurs, delivery of the unit will be delayed.

Torsional design analyses should be done on all new units unless there is successful operating experience with a similarly configured compressor (the same compressor frame, cylinders, staging, coupling and driver) and similar operating conditions (the same pressures, temperatures and load steps). Consideration should be given to doing a torsional analysis on existing units where the operating conditions will be changed significantly from the existing conditions, and especially if the unit is being restaged.

## Troubleshooting

Minor troubles can normally be expected at various times during routine operation of the compressor. These troubles are most often traced to dirt, liquid, and maladjustment, or to operating personnel being unfamiliar with functions of the various machine parts and systems. Difficulties of this type can usually be corrected by cleaning, proper adjustment, elimination of an adverse condition, or quick replacement of a relatively minor part.

Major trouble can usually be traced to long periods of operation with unsuitable coolant or lubrication, careless operation and inadequate maintenance, or the use of the machine on a service for which it was not intended.

A defective inlet valve can generally be found by feeling the valve cover. It will be much warmer than normal. Discharge valve leakage is not as easy to detect since the discharge is always hot. Experienced operators of water-cooled units can usually tell by feel if a particular valve is leaking. The best indication of discharge valve trouble is the discharge temperature. This will rise, sometimes rapidly, when a valve is in poor condition or breaks. This is one very good reason for keeping a record of the discharge temperature from each cylinder.

Recording of the interstage pressure on multistage units is valuable because any variation, when operating at a given load point, indicates trouble in one or the other of the two stages. If the pressure drops, the trouble is in the low pressure cylinder. If it rises, the problem is in the high pressure cylinder.

Troubleshooting is largely a matter of elimination based on a thorough knowledge of the interrelated functions of the various parts and the effects of adverse conditions. A complete list of possible troubles with their causes and corrections is impractical, but the following list of the more frequently encountered troubles and their causes is offered as a guide (Fig. 13-21).

# CENTRIFUGAL COMPRESSORS

This section is intended to supply information sufficiently accurate to determine whether a centrifugal compressor should be considered for a specific job. The secondary objective is to present information for evaluating compressor performance.

Fig. 13-22 gives an approximate idea of the flow range that a centrifugal compressor will handle. A multi-wheel (multistage) centrifugal compressor is normally considered for inlet volumes between 500 and 200,000 inlet acfm. A single-wheel (single stage) compressor would normally have application between 100 and 150,000 inlet volume. A multiwheel compressor can be thought of as a series of single wheel compressors contained in a single casing.

Fig. 13-22 efficiency values should be used as a reference only. The efficiencies of centrifugal compressors rely on the ability to select optimized impeller flow coefficients for the specified process conditions, and will deteriorate for non-optimal impeller flow coefficients and high compression ratios or compressors with more than 4–5 impellers.

These efficiencies reflect compressor designs after say 1998; in general earlier designs could be 4% lower in efficiency.

Most centrifugal compressors operate at speeds of 3,000 rpm or higher, a limiting factor being impeller stress considerations as well as velocity limitation of 0.8 to 0.85 Mach number at the impeller tip and eye. Recent advances in machine design have resulted in production of some units running at speeds in excess of 40,000 rpm.

Centrifugal compressors are usually driven by electric motors, steam or gas turbines (with or without speed-increasing gears), or turboexpanders.

There is an overlap of centrifugal and reciprocating compressors on the low end of the flow range, see Fig. 13-3. On the higher end of the flow range an overlap with the axial compressor exists. The extent of this overlap depends on a number of things. Before a technical decision could be reached as to the type of compressor that would be installed, the service, operational requirements, and economics would have to be considered.

Design requirements for centrifugal compressors are covered by API Standard 617.

## Components of Centrifugal Compressors

Figs. 13-23 through 13-25 provide cross sectional drawings and identification of major components for typical centrifugal compressors. The essential components of a centrifugal compressor that accomplish the compression task are described in the following text referring to Figure 13-24. The gas entering the inlet nozzle of the compressor is guided (often with the help of guide vanes) to the inlet of the impeller. An impeller consists of a number of rotating vanes that impart mechanical energy to the gas. As we will see later, the gas will leave the impeller with an increased velocity and increased static pressure. In the diffuser, part of the velocity is converted into static pressure. Diffusers can be vaneless or contain a number of vanes. If the compressor has more than one impeller, the gas will be again brought in front of the next impeller through the return channel and the return vanes. If the compressor has only one impeller, or after the diffuser of the last impeller in a multi stage compressor, the gas enters the discharge system. The discharge system can either make use of a volute, which can further convert velocity into static pressure, or a simple cavity that collects the gas before it exits the compressor through the discharge nozzle.

The rotating part of the compressor consists of all the impellers. This rotor runs on two radial bearings (on all modern compressors, these are hydrodynamic tilting pad bearings), while the axial thrust generated by the impellers is balanced by a balance piston, and the resulting force is balanced by a hydrodynamic tilting pad thrust bearing.

To keep the gas from escaping at the shaft ends, dry gas seals are typically used on both shaft ends. Other seal types have been used in the past, but virtually all modern centrifugal compressors used in the oil and gas industry use dry gas seals. Refer to the Dry Gas Seals discussion for additional information.

The entire assembly is contained in a casing. For discharge pressures below about 3400 kPa (500 psi), the casing is horizontally split to allow the installation of the rotating components. For higher pressures, the compressors are usually of the barrel type. The pressure containing casing, consists of a center body with end caps on either end. Bearings, seals, shaft and aerodynamic components (both rotating and stationary) can slide in and out of the center body once one of the endcaps is removed (Fig 13-25).

## Performance Calculations

The operating characteristics must be determined before an evaluation of compressor suitability for the application can be made. Fig. 13-26 gives a rough comparison of the characteristics of the axial, centrifugal, and reciprocating compressor.

The centrifugal compressor approximates the constant head-variable volume machine, while the reciprocating is a constant volume-variable head machine. The axial compressor, which is a low head, high flow machine, falls somewhere in between. A compressor is a part of the system, and its performance is dictated by the system resistance. The desired system capability or objective must be determined before a compressor can be selected.

Fig. 13-27 is a typical performance map which shows the basic shape of performance curves for a variable-speed centrifugal compressor. The curves are affected by many variables, such as desired compression ratio, type of gas, number of wheels, sizing of compressor, etc.

**FIG. 13-21**

**Probable Causes of Reciprocating Compressor Trouble**

| Trouble | Probable Cause(s) |
|---|---|
| Compressor Will not Start | 1. Power supply failure.<br>2. Switchgear or starting panel.<br>3. Low oil pressure shutdown switch.<br>4. Control panel. |
| Motor Will Not Synchronize | 1. Low voltage.<br>2. Excessive starting torque.<br>3. Incorrect power factor.<br>4. Excitation voltage failure. |
| Low Oil Pressure | 1. Oil pump failure.<br>2. Oil foaming from counterweights striking oil surface.<br>3. Cold oil.<br>4. Dirty oil filter.<br>5. Interior frame oil leaks.<br>6. Excessive leakage at bearing shim tabs and/or bearings.<br>7. Improper low oil-pressure switch setting.<br>8. Low gear oil pump by-pass/relief valve setting.<br>9. Defective pressure gauge.<br>10. Plugged oil sump strainer.<br>11. Defective oil relief valve. |
| Noise In Cylinder | 1. Loose piston.<br>2. Piston hitting outer head or frame end of cylinder.<br>3. Loose crosshead lock nut.<br>4. Broken or leaking valve(s).<br>5. Worn or broken piston rings or expanders.<br>6. Valve improperly seated/damaged seat gasket.<br>7. Free air unloader plunger chattering. |
| Excessive Packing Leakage | 1. Worn packing rings.<br>2. Improper lube oil and/or insufficient lube rate (blue rings).<br>3. Dirt in packing.<br>4. Excessive rate of pressure increase.<br>5. Packing rings assembled incorrectly.<br>6. Improper ring side- or end-gap clearance.<br>7. Plugged packing vent system.<br>8. Scored piston rod.<br>9. Excessive piston rod run-out. |

| Trouble | Probable Cause(s) |
|---|---|
| Packing Over-Heating | 1. Lubrication failure.<br>2. Improper lube oil and/or insufficient lube rate.<br>3. Insufficient cooling. |
| Excessive Carbon On Valves | 1. Excessive lube oil.<br>3. Oil carryover from inlet system or previous stage.<br>4. Broken or leaking valves causing high temperature.<br>5. Excessive temperature due to high pressure ratio across cylinders. |
| Relief Valve Popping | 1. Faulty relief valve.<br>2. Leaking suction valves or rings on next higher stage.<br>3. Obstruction (foreign material, rags), blind or valve closed in discharge line. |
| High Discharge Temperature | 1. Excessive compression ratio on cylinder due to leaking inlet valves or rings on next higher stage.<br>2. Fouled intercooler/piping.<br>3. Leaking discharge valves or piston rings.<br>4. High inlet temperature.<br>5. Fouled water jackets on cylinder.<br>6. Improper lube oil and/or lube rate. |
| Frame Knocks | 1. Loose crosshead pin, pin caps or crosshead shoes.<br>2. Loose/worn main, crankpin or crosshead bearings.<br>3. Low oil pressure.<br>4. Cold oil.<br>5. Incorrect oil.<br>6. Knock is actually from cylinder end. |
| Crankshaft Oil Seal Leaks | 1. Faulty seal installation.<br>2. Clogged drain hole. |
| Piston Rod Oil Scraper Leaks | 1. Worn scraper rings.<br>2. Scrapers incorrectly assembled.<br>3. Worn/scored rod.<br>4. Improper fit of rings to rod/side clearance. |

Courtesy of Ingersoll-Rand Co

**FIG. 13-22**
**Approximate Centrifugal Compressor Flow Range**

| Nominal flow range (inlet acfm) | Average polytropic efficiency | Average isentropic efficiency | Speed to develop 10,000 ft head/wheel |
|---|---|---|---|
| 100- 500 | 0.68 | 0.65 | 20,500 |
| 500- 7,500 | 0.78 | 0.76 | 10,500 |
| 7,500- 20,000 | 0.84 | 0.81 | 8,200 |
| 20,000- 33,000 | 0.84 | 0.81 | 6,500 |
| 33,000- 55,000 | 0.84 | 0.81 | 4,900 |
| 55,000- 80,000 | 0.84 | 0.81 | 4,300 |
| 80,000-115,000 | 0.84 | 0.81 | 3,600 |
| 115,000-145,000 | 0.84 | 0.81 | 2,800 |
| 145,000-200,000 | 0.84 | 0.81 | 2,500 |

With variable speed, the centrifugal compressor can deliver constant capacity at variable pressure, variable capacity at constant pressure, or a combination variable capacity and variable pressure.

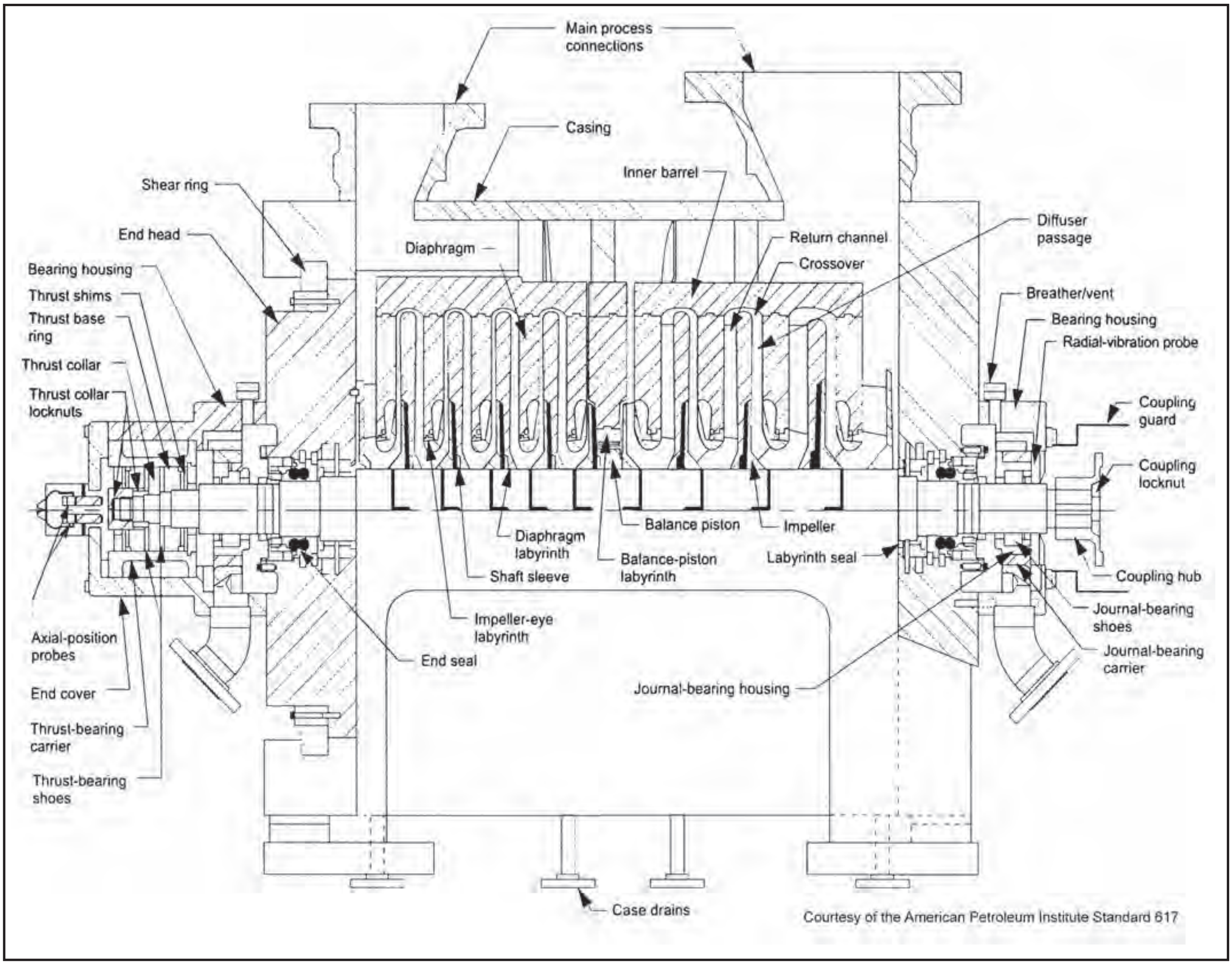
## Similarity Law (Fan Law)

Under certain simplifying conditions, operating points of a compressor at different speeds can be compared (Kurz and Ohanian, 2003). This fact is captured in the fan law, which is strictly only true for identical Mach numbers in all stages, but which is still a good approximation for cases where the machine Mach number:

$$M_N = \frac{u}{\sqrt{k_1 Z_1 R T_1}} \qquad \textbf{Eq 13-22}$$

changes by less than 10% (for single and two stage compressors). The more stages the compressor has, the less deviation is acceptable (Kurz and Fozi, 2002). The fan law is based on the fact that if for two operating points A and B all velocities change by the same factor (which in particular means that none of the flow angles change), then the compressor will show the follow-

**FIG. 13-23**
**Example Centrifugal Compressor Showing Nomenclature of Key Parts**



ing relations between two different operating points :

$$\frac{Q_A}{N_A} = \frac{Q_B}{N_B} \quad \textbf{Eq 13-23a}$$

$$\frac{H_{isA}}{N_A^2} = \frac{H_{isB}}{N_B^2} \quad \textbf{Eq 13-23b}$$

$$\eta_A = \eta_B$$

and therefore

$$\frac{GHP_A}{N_A^3} = \frac{GHP_B}{N_B^3} \quad \textbf{Eq 13-24}$$

This does not imply that the system within which the compressor operates will force the compressor to operate along the fan line. In general, the system will enforce a head and flow relationship that is not (at least not exactly) following the fan law. The intersection of the new resulting system pressure (not described by the fan law), and the new operating condition of the compressor (as described by the fan law) sets the new operating condition of the system.

Fig. 13-27 depicts typical performance curves with a small compression ratio. The system resistance has been superimposed on the chart: Line A represents typical system resistance of a closed system, such as a refrigeration unit where there is a relatively constant discharge pressure. Line B is an open-end system, such as pipeline application where pressure increases with capacity.

Fig. 13-28 shows a higher compression ratio. The range of stable operation is reduced because of the larger compression ratio. This is indicated by the surge line in Fig. 13-28 being further to the right than in Fig. 13-27.

## Estimating Performance

Figs. 13-29 through 13-36 may be used for estimating compressor performance. These curves are only suitable for estimating only and are not intended to take the place of a "wheel-by-wheel" selection by the compressor manufacturer, nor should the curves be used to calculate performance using field data in an attempt to determine a variance from predicted performance based on manufacturer's data. Fig. 13-29 is used to convert scfm to icfm. All centrifugal compressors are based on flows that are converted to inlet or actual cubic feet per minute. This is done because the centrifugal wheel is sensitive to inlet volume, compression ratio (i.e., head), and specific speed.

Fig. 13-30 is a useful curve to find inlet (actual) cfm when the weight flow in lb/min is known. Actual cfm and inlet cfm both denote the gas at suction conditions. These terms are often used interchangeably. This curve can be used in reverse to determine mass flow.

**FIG. 13-24**

**Typical Centrifugal Compressor Cutaway**

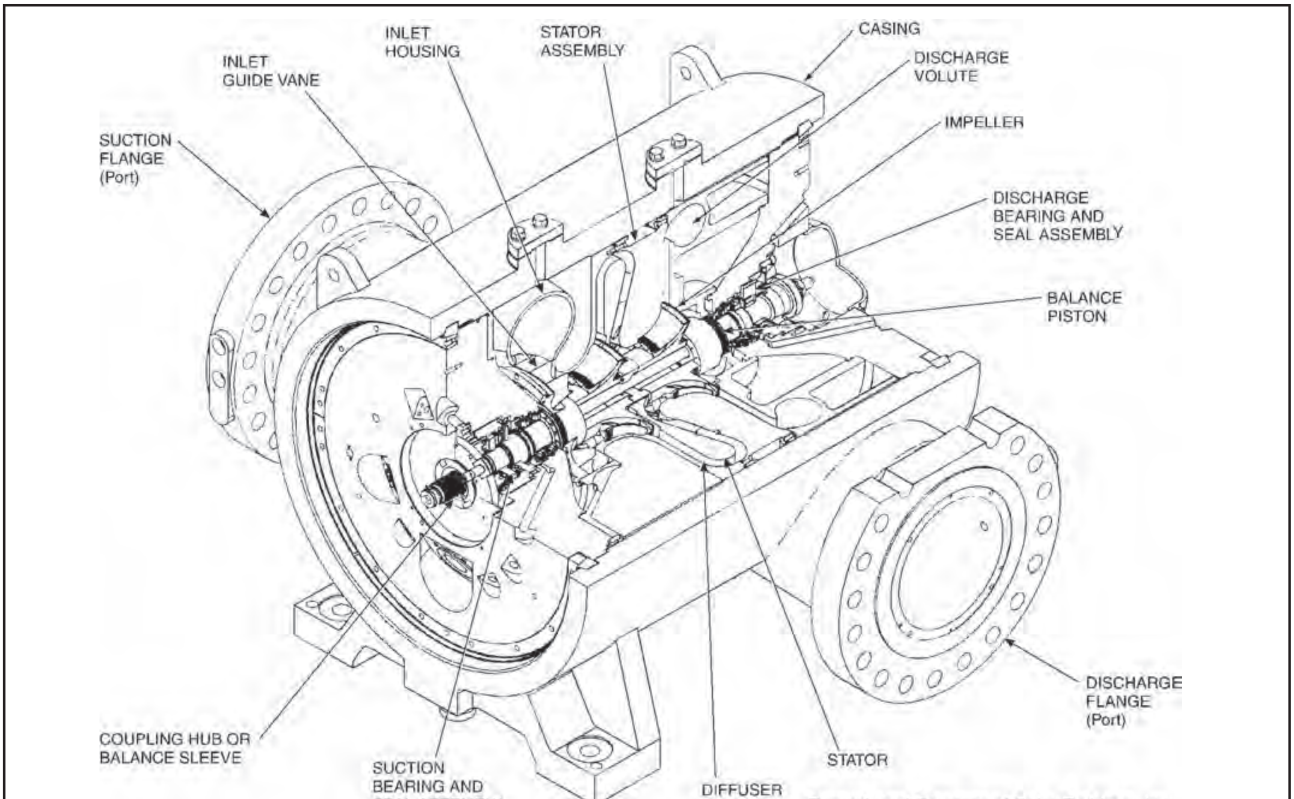


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Fig. 13-31 is used to determine the approximate discharge temperature that is produced by the compression ratio. Discharge temperatures above the 400°F range should be checked since mechanical problems as well as safety problems may exist. This curve includes compressor efficiencies in the range of 60 to 75%.

**Example 13-4** — Given: r = 10.0; $Q_1$ = 10,000 icfm

k = 1.15; $t_1$ = 0°F

Find: Discharge temperature

Answer: $t_2$ = 230°F (approximately) from Fig. 13-31.

Note: for a natural gas with k = 1.30 $t_2$ = 480°F (excessively high).

Fig. 13-33 gives the approximate horsepower required for the compression. It includes overall compressor efficiencies in the range of 60 to 70%.

**FIG. 13-25**

**Centrifugal Compressor Cross Section**

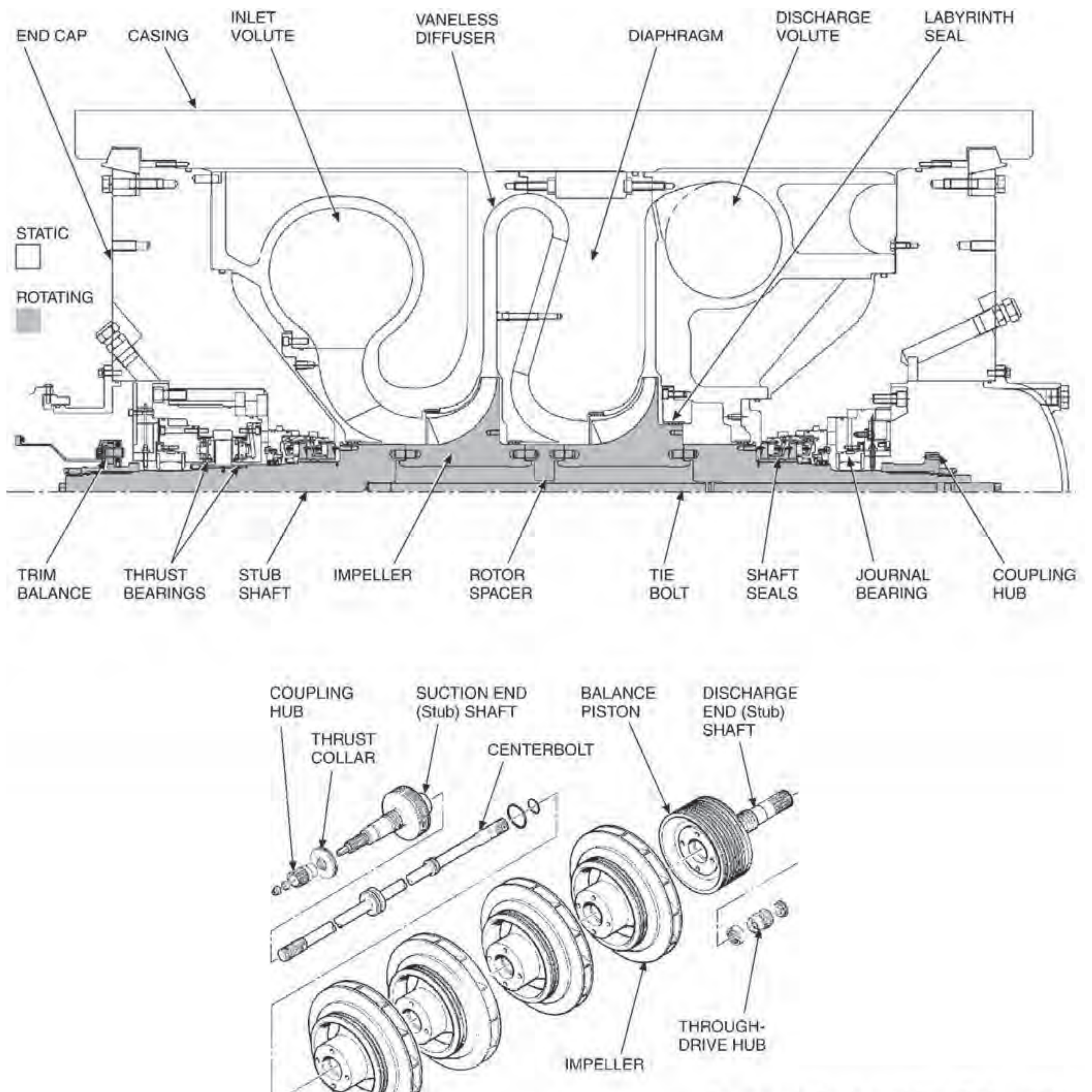


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**FIG. 13-26**
**Compressor Head**

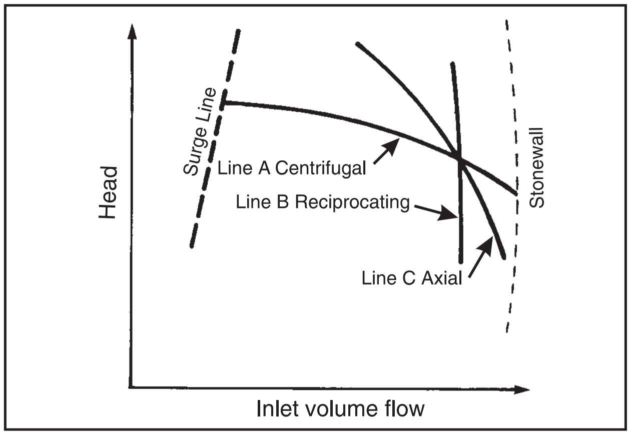
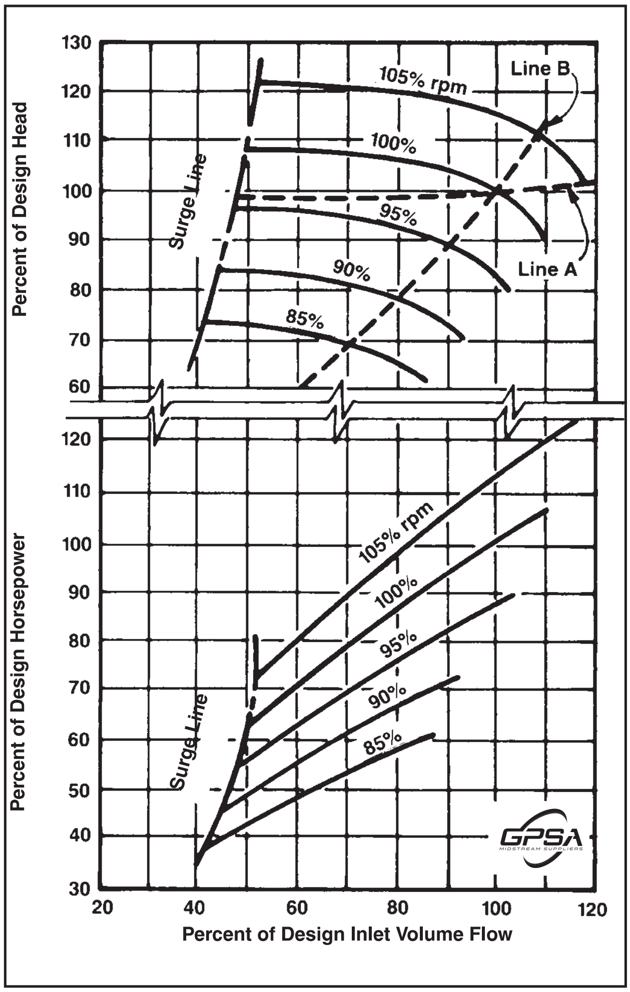



**FIG. 13-27**
**Compressor Performance, Low Compression Ratio**



**Example 13-5** — Given: Weight flow, w, = 1,000 lb/min

head = 70 000 ft-lb/lb

Find: Horsepower

Answer: GHP = 3,000 from Fig. 13-33.

Fig. 13-36 predicts the approximate number of compressor wheels required to produce the head. If the number of wheels is not a whole number, use the next highest number.

## Calculating Performance

When more accurate information is required for compressor head, gas horsepower, and discharge temperature, the equations in this section should be used. This method applies to a gas mixture for which a P-H diagram chart is not available. To calculate the properties of the gas, see Figs. 13-6 and 13-7. All values for pressure and temperature in these calculation procedures are the absolute values. Unless otherwise specified, volumes of flow in this section are actual volumes.

**FIG. 13-28**

**Compressor Performance, Higher Compression Ratio**

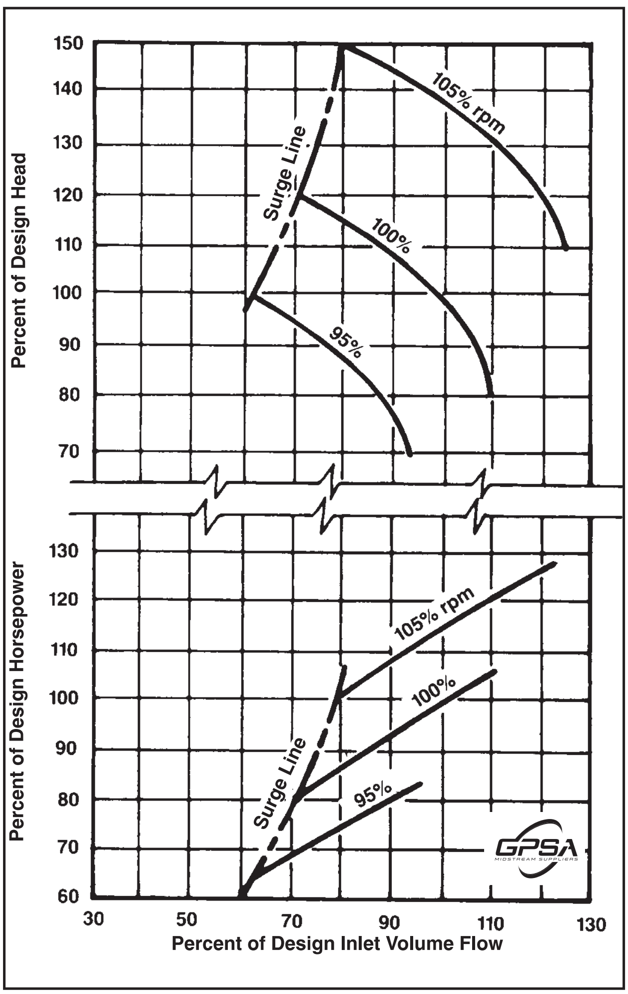


**FIG. 13-29**
**ICFM to SCFM**
**Z = 1**

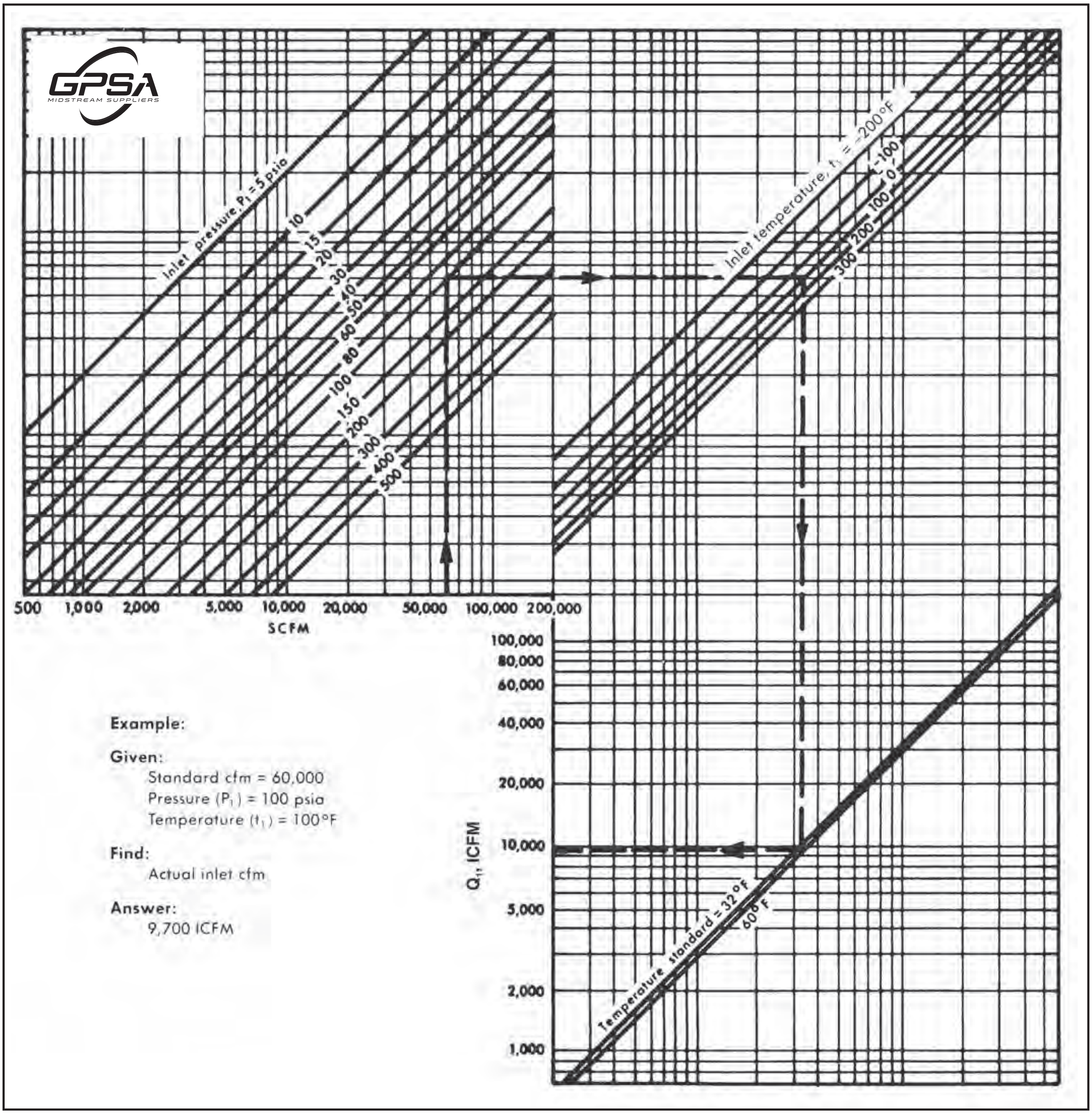


**FIG. 13-30**

**Mass Flow to Inlet Volume Flow**

**Z = 1**

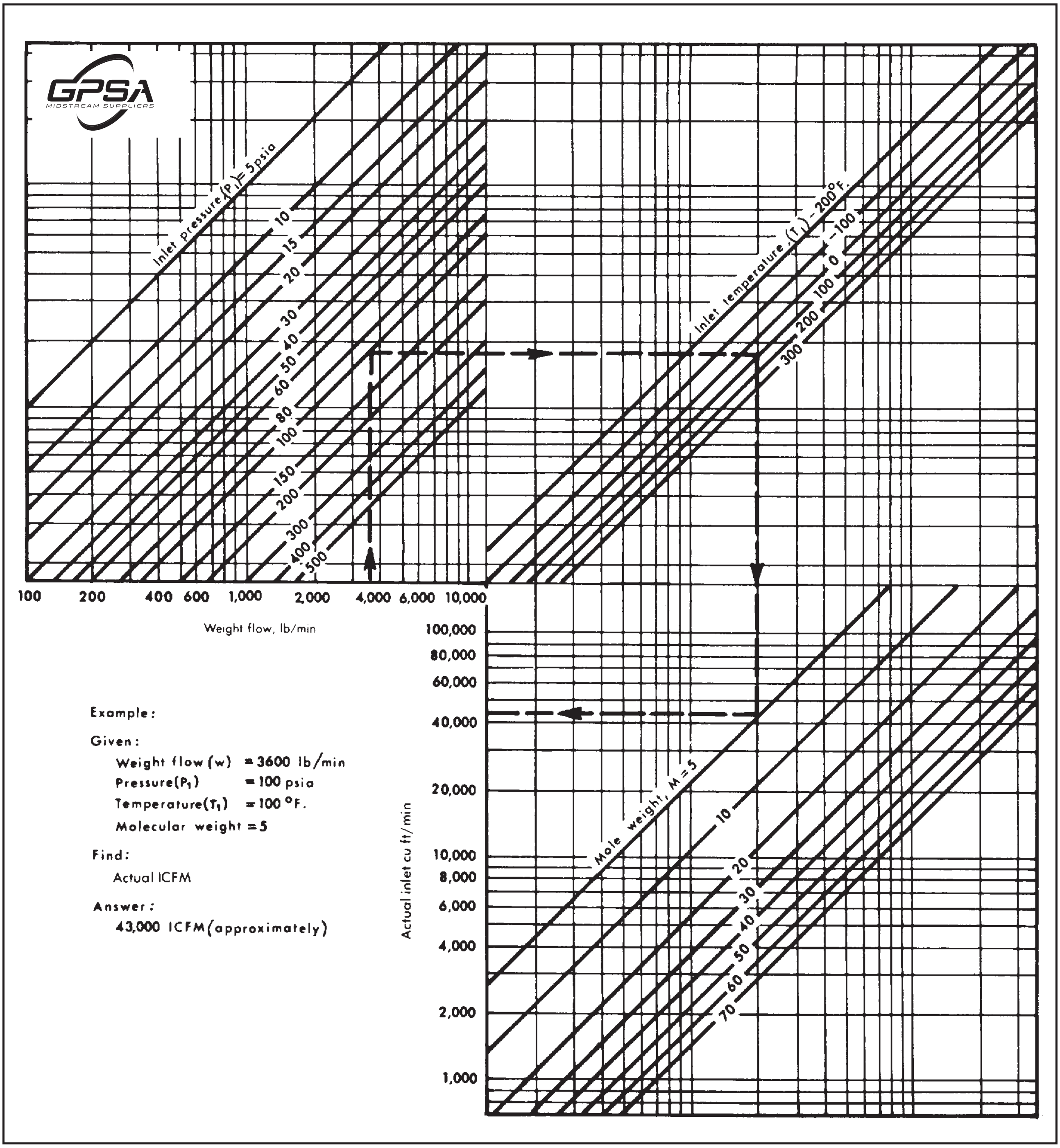


**FIG. 13-31**
**Approximate Discharge Temperature**
**Z = 1**

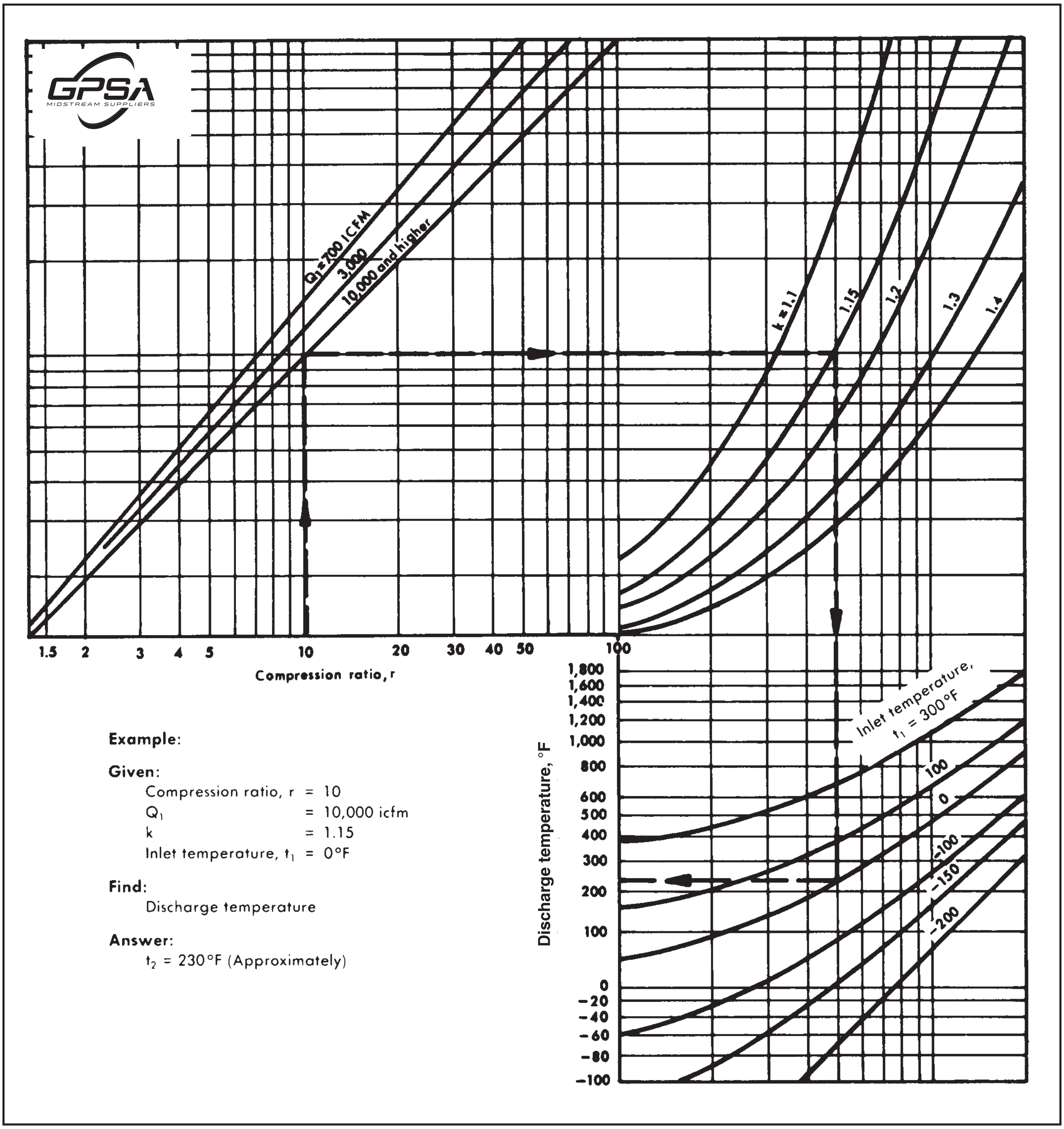
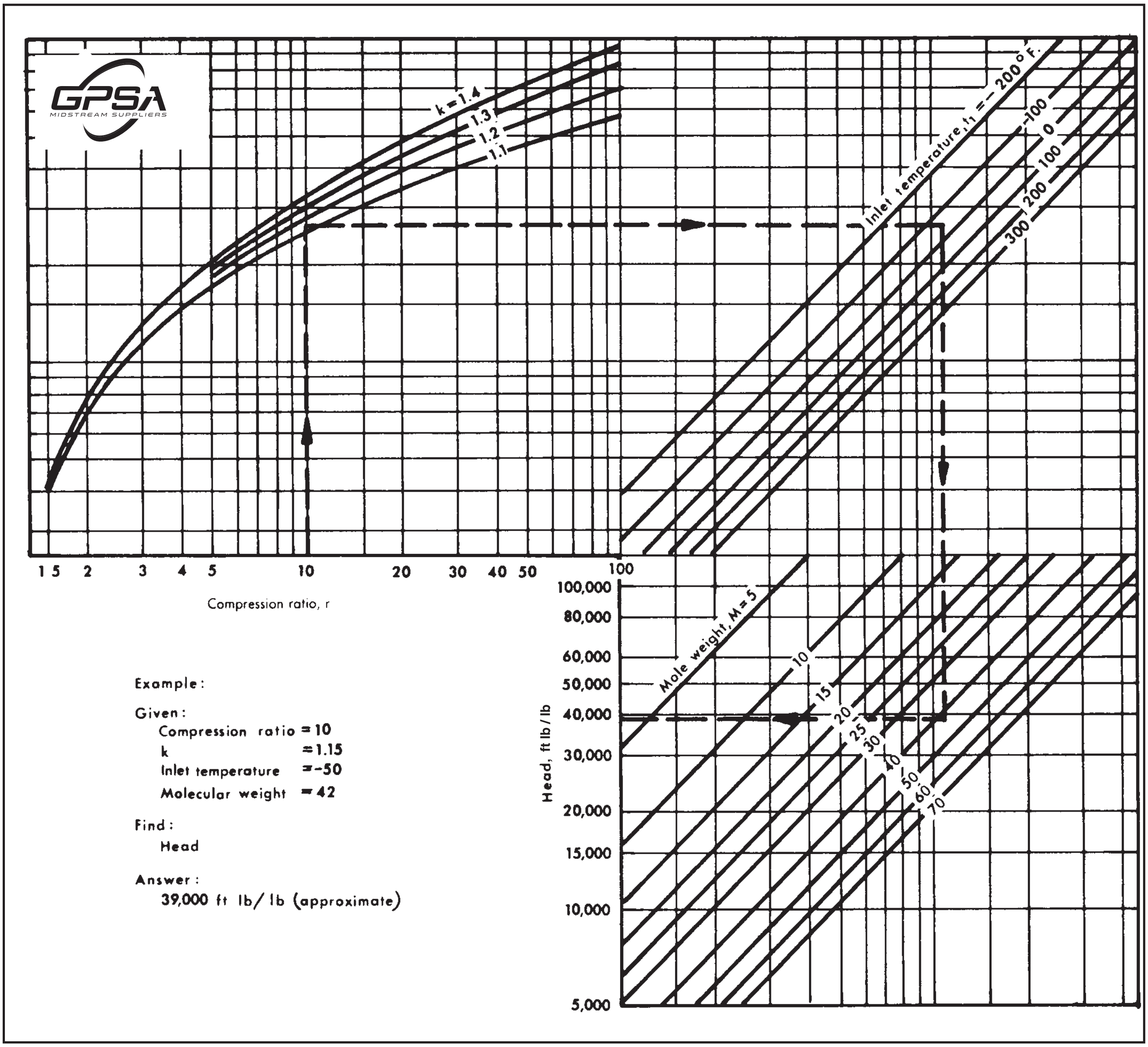


**FIG. 13-32**
**Head**
**Z = 1**



## SCG SERIES

Flow range: 1-13,5 m3/min
Power Range: 11-135 kW
Max Discharge Pressure: 20 barG

## NG SERIES

Flow range: 0,5-50 m3/min
Power Range: 2,2 – 400 kW
Max Discharge Pressure: 20 barG

## ITA-HP BOOSTER SERIES

Suction Pressure: up to 8 barG
Discharge Pressure: up to 25 barG
Flow range: 2-36 m3/min
Power Range: 30-350 kW

## ITA-TS TWO-STAGE

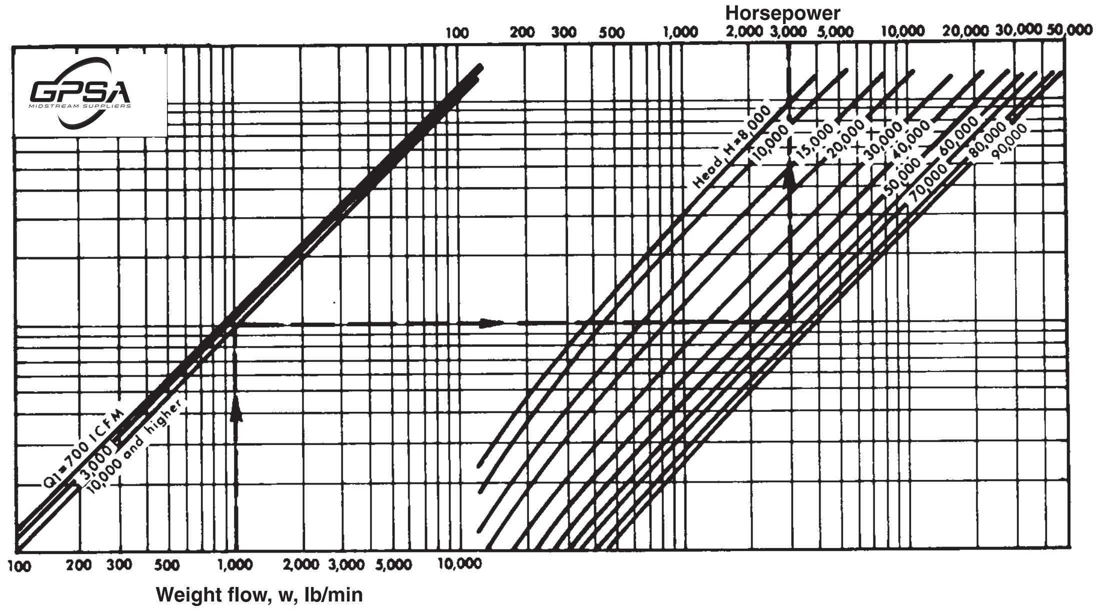
Discharge Pressure: up to 25 barG
Flow range:14-27 m3/min
Power Range: 90-250 kW

## OVER 100 YEARS OF INNOVATION

Termomeccanica Industrial Compressors (a.k.a TMIC) not only designs and manufactures oil injected bare shaft screw compressors for air & gas applications which are internationally acknowledged for their outstanding performance, long-life and innovation but also offers high-speed delivery and service to its worldwide customer base.

**FIG. 13-33**

**Approximate Horsepower Determination**




**Example:**

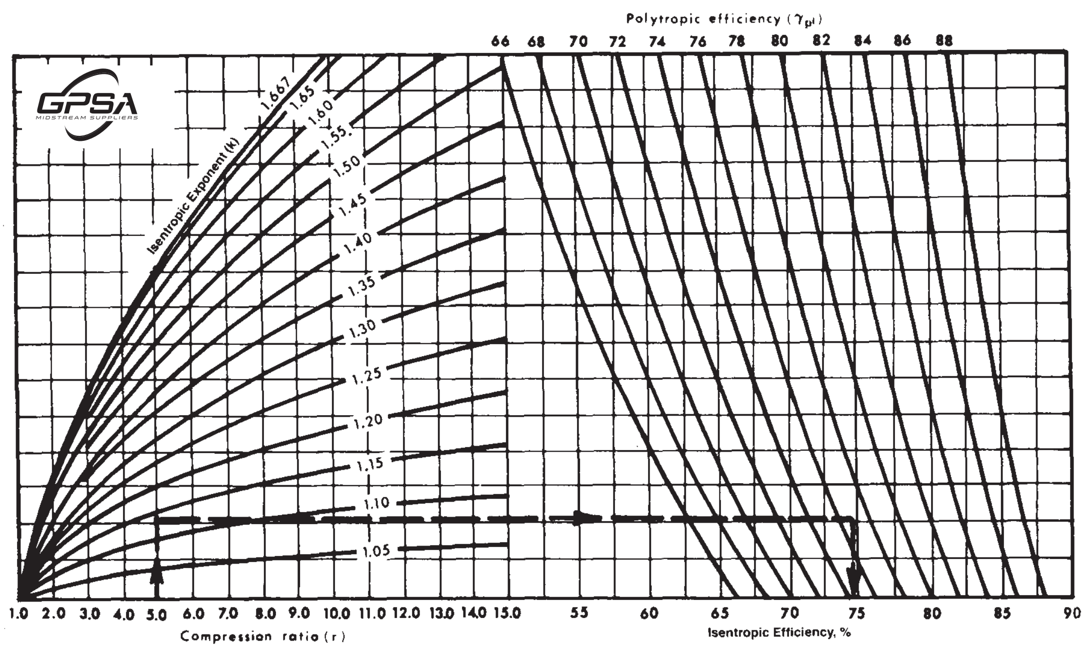
**Given:**
Weight flow, w = 1,000 lb/min
Head = 70,000 ft-lb/lb

**Find:**
Horsepower

**Answer:**
Hp = 3,000

**FIG. 13-34**

**Efficiency Conversion**



To calculate the inlet volume:

$$Q = \frac{(w)\,(1{,}545)\,(T_1)\,(Z_1)}{(MW)\,(P_1)\,(144)} \qquad \textbf{Eq 13-25}$$

If we assume the compression to be isentropic (reversible adiabatic, constant entropy), then:

$$H_{is} = \frac{ZRT}{MW\,(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-26}$$

Since these calculations will not be wheel-by-wheel, the head will be calculated across the entire machine. For this, use the average compressibility factor:

$$Z_{avg} = \frac{Z_1 + Z_2}{2}$$

The heat capacity ratio, k, is normally determined at the average suction and discharge temperature (see Figs. 13-7 and 13-8).

## Isentropic Calculation

To calculate the head:

$$H_{is} = \frac{Z_{avg}RT_1}{MW\,(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-27a}$$

which can also be written in the form:

$$H_{is} = \frac{1545}{MW}\;\frac{Z_{avg}T_1}{(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-27b}$$

The gas horsepower can now be calculated from:

$$GHP = \frac{(w)\,(H_{is})}{(\eta_{is})\,(33{,}000)} \qquad \textbf{Eq 13-28}$$

The approximate theoretical discharge temperature can be calculated from:

$$\Delta T_{ideal} = T_1\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-29}$$

$$T_2 = T_1 + \Delta T_{ideal} \qquad \textbf{Eq 13-30}$$

The actual discharge temperature can be approximated:

$$\Delta T_{actual} = T_1\,\frac{\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]}{\eta_{is}} \qquad \textbf{Eq 13-31}$$

$$T_2 = T_1 + \Delta T_{actual} \qquad \textbf{Eq 13-32}$$

## Polytropic Calculation

Sometimes compressor manufacturers use a polytropic path instead of isentropic. Polytropic efficiency is defined by:

$$\frac{n}{(n-1)} = \left[\frac{k}{(k-1)}\right]\eta_p \qquad \textbf{Eq 13-33}$$

(See Fig. 13-34 for conversion of isentropic efficiency to polytropic efficiency.)

The equations for head and gas horsepower based upon polytropic compression are:

$$H_p = \frac{Z_{avg}RT_1}{MW\,(n-1)/n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \qquad \textbf{Eq 13-34a}$$

which also can be written in the form:

$$H_p = \frac{1545}{MW}\;\frac{Z_{avg}T_1}{(n-1)/n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \qquad \textbf{Eq 13-34b}$$

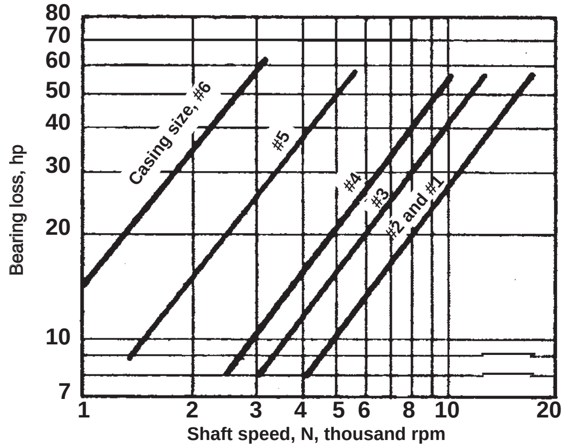
$$GHP = \frac{(w)\,(H_p)}{(\eta_p)\,(33{,}000)} \qquad \textbf{Eq 13-35}$$

Polytropic and isentropic head are related by

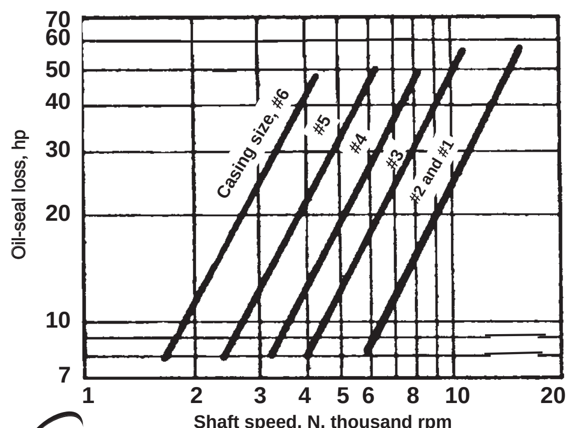
**FIG. 13-35**
**Mechanical Losses**

| Casing Size | Max Flow (inlet acfm) | Nominal Speed (rpm) |
|---|---|---|
| 1 | 7,500 | 10,500 |
| 2 | 20,000 | 8,200 |
| 3 | 33,000 | 6,400 |
| 4 | 55,000 | 4,900 |
| 5 | 115,000 | 3,600 |
| 6 | 150,000 | 2,800 |

**a. Bearing horsepower losses**




**b. Oil-seal horsepower losses**




*Courtesy Chemical Engineering Magazine*

$$H_p = \frac{H_{is}\,\eta_p}{\eta_{is}} \quad \textbf{Eq 13-36}$$

The approximate actual discharge temperature can be calculated in an analagous manner to Equations 13-29 through 13-32 but replacing (k–1)/k and ηis with (n–1)/n and ηpoly respectively.

### Mechanical Losses

After the gas horsepower has been determined by either method, horsepower losses due to friction in bearings, seals, and speed increasing gears must be added.

Fig. 13-35 shows losses related to the shaft speed and casing size for conventional multistage units.

Bearings and seal losses can also be roughly computed from Scheel's equation:

$$\text{Mechanical losses} = (\text{GHP})^{0.4} \quad \textbf{Eq 13-37}$$

To calculate the total compressor horsepower:

$$\text{BHP} = \text{GHP} + \text{mechanical losses} \quad \textbf{Eq 13-38}$$

The mechanical losses of centrifugal compressors (including windage, bearings) are typically between 1 and 2% of the total power, with the lower number for larger machines. Gearbox losses are usually 2 to 3%, for parallel shaft gearboxes, with the higher number for higher gearbox ratios, especially for gearboxes with an idler gear.

### Compressor Speed

The basic equation for estimating the speed of a centrifugal compressor is:

$$N = (N_{nominal}) \sqrt{\frac{\text{H total}}{(\text{No. of wheels})\ (\text{H max/wheel})}} \quad \textbf{Eq 13-39}$$

where the number of wheels is determined from Fig. 13-36.

Nominal speeds to develop 10,000 feet of head/wheel can be determined from Fig. 13-22. However, to calculate the maximum head per wheel, the following equation based on molecular weight (or more accurately, density) can be used.

$$\text{H max/wheel} = 15{,}000 - 1{,}500\ (\text{MW})^{0.35} \quad \textbf{Eq 13-40}$$

This equation will give a head of 10,000 ft for a gas when MW = 30 and 11,000 ft when MW = 16.

### P-H Diagram

When a P-H diagram is available for the gas to be compressed, the following procedure should be used. Fig. 13-37 represents a section of a typical P-H diagram.

For the given inlet conditions, the enthalpy can be shown as point 1 on the P-H diagram. For a single compression stage, starting from Point 1 follow the line of constant entropy to the required discharge pressure ($P_2$), locating the isentropic discharge state point ($2_{is}$). With these two points located the differential isentropic enthalpy can be calculated from the following equation:

$$\Delta h_{is} = h_{2is} - h_1 \quad \textbf{Eq 13-41}$$

To convert to isentropic head, the equation is:

$$H_{is} = \Delta h_{is}\ (778\ \text{ft} \cdot \text{lb/Btu}) \quad \textbf{Eq 13-42}$$

To find the discharge enthalpy:

$$h_2 = \frac{\Delta h_{is}}{\eta_{is}} + h_1 \quad \textbf{Eq 13-43}$$

The actual discharge temperature can now be obtained from the P-H diagram. The gas horsepower can be calculated using Equation 13-28 and Equation 13-35.

From Fig. 13-36 and Equations 13-39 and 13-40, the speed and number of wheels can be estimated.

To convert to polytropic head it will be necessary to assume a polytropic efficiency. See Fig. 13-22 for an efficiency corresponding to the inlet flow. Fig. 13-34 will give a corresponding adiabatic efficiency. The polytropic head may now be determined from Equation 13-36.

When a P-H diagram is available, it is the fastest and most accurate method of determining compressor horsepower and discharge temperature.

### Centrifugal Refrigeration Compressors

Compression ratio per wheel will vary on the order of 1.5 to 2.75 per wheel depending on the refrigerant and speed.

Due to the ease of applying external side loads to centrifugal machines, it is quite common to flash refrigerant from the condenser en route to the evaporators and/or to accept side loads from product being cooled by refrigerant at higher pressures than the lowest evaporator level.

Since side-loading is the practice rather than the exception, it is common to let the centrifugal compressor manufacturer obtain the desired performance characteristics from the following data: evaporator temperature levels; refrigeration loads required in MMBtu/hr; heat rejection medium (air or water); and type of driver.

## GENERAL

### Flow Limits

Two conditions associated with centrifugal compressors are surge (pumping) and stone-wall (choked flow).

At some point on the compressor's operating curve there exists a condition of minimum flow/maximum head where the developed head is insufficient to overcome the system resistance. This is the surge point. When the compressor reaches this point, the gas in the discharge piping back-flows into the compressor. Without discharge flow, discharge pressure drops until it is within the compressor's capability, only to repeat the cycle.

The repeated pressure oscillations at the surge point should be avoided since it can be detrimental to the compressor. Surging can cause the compressor to overheat to the point the maximum allowable temperature of the unit is exceeded. Also, surging can cause damage to the thrust bearing due to the rotor shifting back and forth from the active to the inactive side.

"Stonewall" or choked flow occurs when sonic velocity is reached at any point in the compressor. When this point is reached for a given gas, the flow through the compressor cannot be increased further.

FIG. 13-36
Wheels Required

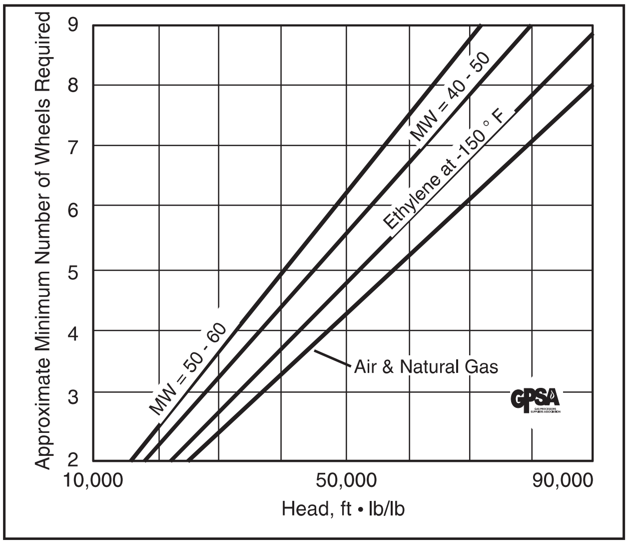
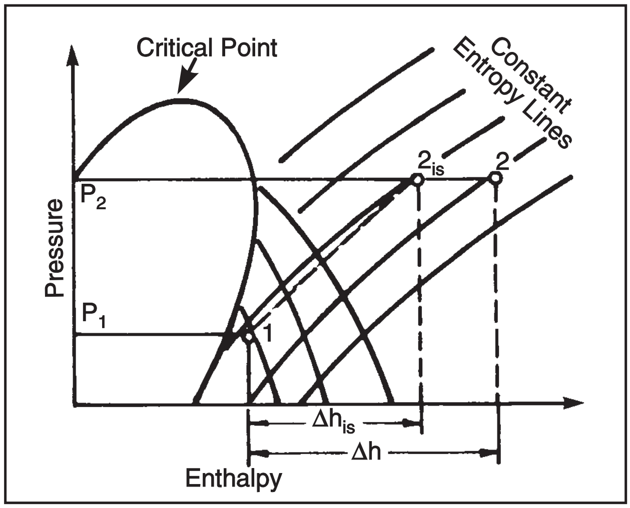



FIG. 13-37
P-H Diagram Construction




## Interstage Cooling

Multistage compressors rely on intercooling whenever the inlet temperature of the gas and the required compression ratio are such that the discharge temperature of the gas exceeds about 300°F.

There are certain processes that require a controlled discharge temperature. For example, the compression of gases such as oxygen, chlorine, and acetylene requires that the temperature be maintained below 200°F.

The thermal stress within the horizontal bolted joint is the governing design limitation in a horizontally split compressor case. The vertically split barrel-type case, however, is free from the thermal stress complication.

Substantial power economy can be gained by precooling the gas before it enters the interstage impellers. Performance calculations indicate that the head and the horsepower are directly proportional to the absolute gas temperature at each impeller.

The gas may be cooled within the casing or, more commonly, in external heat exchangers.

Two methods of cooling within the casing are used — water cooled diaphragms between successive stages and direct liquid injection into the gas.

Diaphragm cooling systems include high-velocity water circulation through cast jackets in the diffuser diaphragms. The diaphragm coolers are usually connected in series.

Liquid injection cooling is the least costly means of controlling discharge temperatures. It involves injecting and atomizing a jet of water or a compatible liquid into the return channels. In refrigeration units, liquid refrigerant is frequently used for this purpose. Injected liquid also functions as a solvent in washing the impellers free of deposits. Nevertheless, the hazards of corrosion, erosion, and flooding present certain problems resulting in possible replacement of the compressor rotor.

External intercoolers are commonly used as the most effective means of controlling discharge temperatures. The gas is discharged from the compressor casing after one or more stages of compression and, after being cooled, is returned to the next stage or series of stages for further compression.

Intercoolers usually are mounted separately. When there are two or more compressor casings installed in series, individual machines may or may not be cooled or have intercoolers. In some cases, it may be advantageous to use an external cooler to precool gas ahead of the first wheel.

## Journal and Thrust Bearings

Radial journal bearings are designed to handle high speeds and heavy loads and incorporate force-feed lubrication. They are self-aligning, straight sleeve, multi-lobe sleeve, or tilting pad type, each sized for good damping characteristics and high stability.

Tilting pad bearings have an advantage over the sleeve type as they eliminate oil whip or half-speed oil whirl which can cause severe vibrations.

Bearing sleeves or pads are fitted with replaceable steel-backed babbitted shells or liners.

Axial thrust bearings are bidirectional, double faced, pivoted-shoe type designed for equal thrust capacity in both directions and arranged for force-feed lubrication on each side.

Thrust bearings are sized for continuous operation at maximum differential pressure including surge thrust loads, axial forces transmitted from the flexible coupling and electric motor thrust.

On units where the thrust forces are low, a tapered land thrust bearing may be used but must be selected for proper rotation direction. At times a combination of pivoted-shoe and tapered land is recommended.

Compressor designs with impellers arranged in one direction usually have a balance drum (piston) mounted on the discharge end of the shaft to minimize axial loads on the thrust bearing.

# INTEGRATED SOLUTIONS ALONG THE $H_2$ VALUE CHAIN

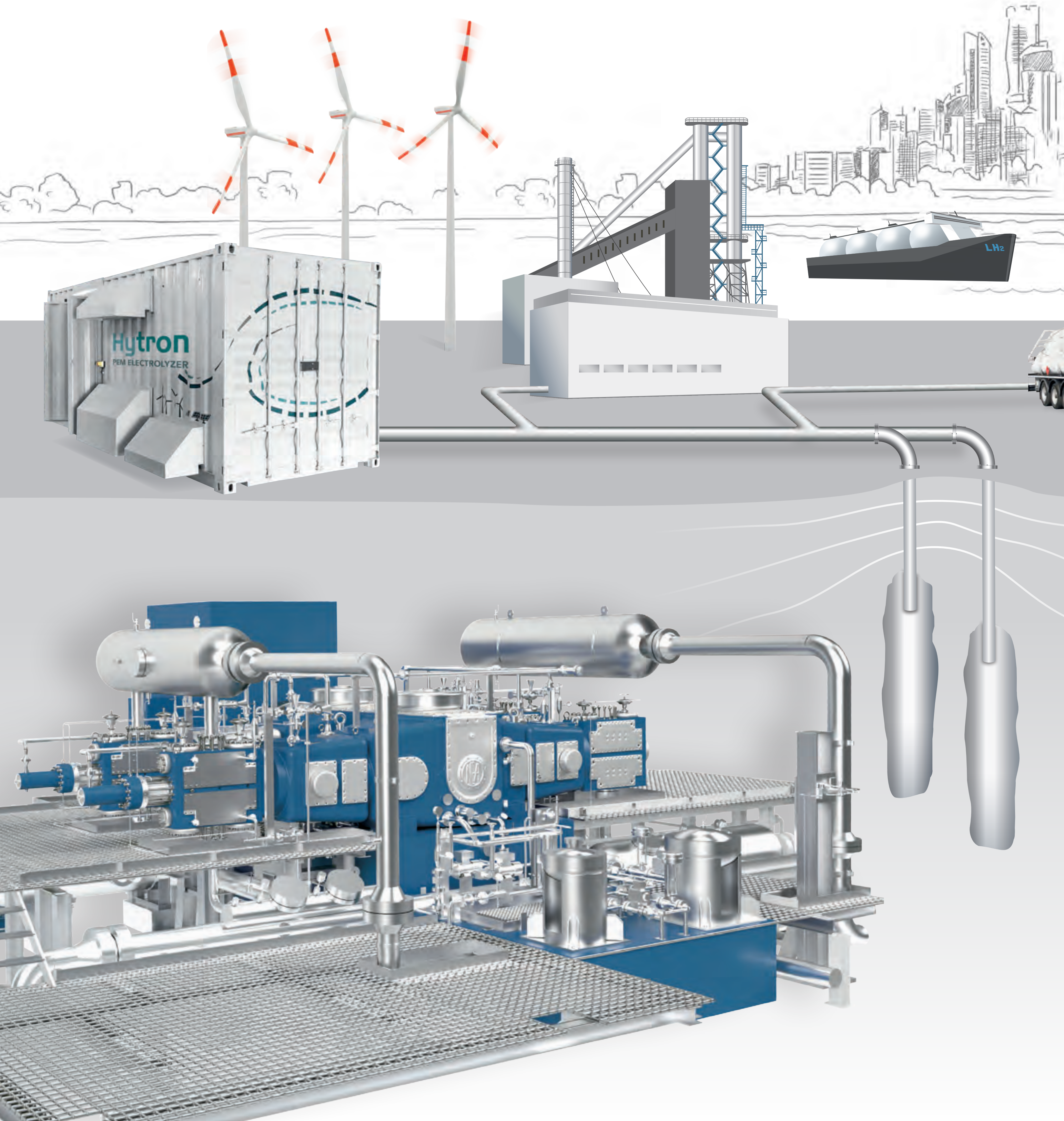


Fig. 13-38 illustrates typical journal and thrust bearings generally used in horizontally and vertically split casings.

Bearing supports are cast integral with, or bolted to, the case with isolated bearing chambers to prevent lubricating oil leakage into the gas system or contamination of the oil or the gas. Bearing housings are horizontally split and readily accessible for inspection and maintenance. Provisions are made to accommodate pick-ups and sensors for vibration and temperature monitoring.

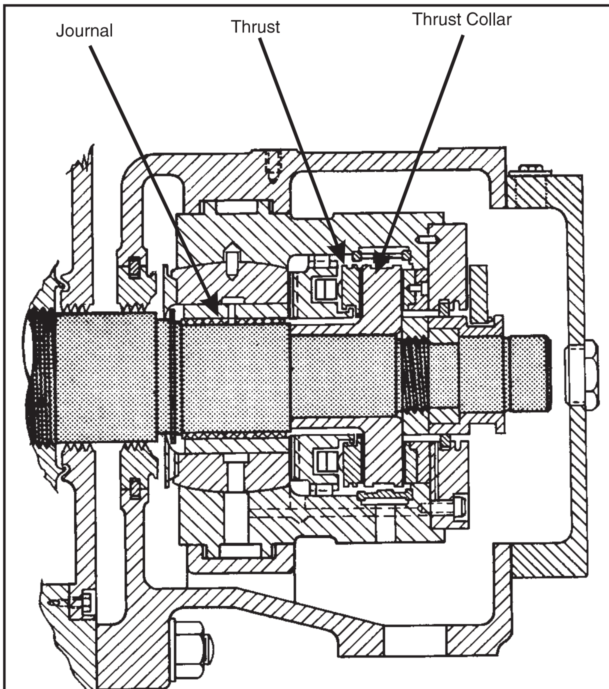
## Magnetic Bearings

Magnetic bearings are a relatively new development that are gaining in popularity. An active magnetic bearing comprises two main components — a mechanical part and an electronic part. The mechanical parts of the bearing are similar to an electric motor with a rotor and stator. An iron core in the stator is wound with coils through which is fed an electric current, thereby inducing a magnetic field. This magnetic field produces the forces that support the compressor shaft.

The electronic part of the active magnetic bearing is the digital control system. It includes sensors that measure the exact position of the shaft. Deviations from the desired position of the shaft will trigger the software in the control system to adjust the current flowing through the electromagnets that determine the strength of the magnetic field. The currents are adjusted according to a set algorithm that corrects the deviation. Magnetic bearings are available in radial and axial/thrust designs.

**FIG. 13-38**

**Journal and Thrust Bearing Assembly**




*Courtesy of Ingersoll-Rand Co.*

**Advantages**

- Reduced space/weight requirements due to elimination of the need for a bearing lube oil system
- Reduced long term costs for maintenance and repairs
- Reduced bearing-related losses (near zero friction)
- Increased reliability and availability
- Improved machine monitoring/diagnostic capabilities
- Higher speeds possible.

**Limitations**

- Generally physically larger than "conventional" bearings
- Higher complexity
- Requires electrical power

See Fig. 13-39 for a schematic of a typical magnetic bearing arrangement.

## Shaft Seals

Shaft seals are provided on all centrifugal compressors to limit, or completely eliminate, gas leakage along the shaft where it passes through the casing.

With the wide range of temperature, pressure, speed, and operating conditions encountered by compressors, there can be no one universal seal, or seal system, to handle all applications.

Basically, the designs of seals available are: labyrinth (gas), restrictive ring (oil or gas), liquid film (oil), and mechanical (contact) (oil or gas).

A mechanical (contact) seal, Fig. 13-40, has the basic elements similar to the liquid film seal. The significant difference is that clearances in this seal are reduced to zero. The seal operates with oil pressure 35 to 50 psi above internal gas pressure as opposed to 5 psi in the liquid film seal.

The mechanical (contact) seal can be applied to most gases, but finds its widest use on clean, heavier hydrocarbon gases, refrigerant gases, etc.

A mechanical gas seal uses the process gas as working fluid to eliminate the seal oil system. See Figs. 13-41 through 13-46.

The liquid film seal, Figs. 13-44 and 13-45, was also developed for the severe conditions of service but requires higher oil circulation rate than the mechanical (contact) type.

The seal consists of two sleeves which run at close clearance to the shaft with a liquid injected between the sleeves to flow to the seal extremities. The sleeves are lined with babbitt or a similar non-galling material which is compatible with the properties of the compressed gas and the sealing liquid.

The sealing liquid, usually a lubricating oil, is introduced between the two rings at a controlled differential pressure of about 5 psi above the internal gas pressure, presenting a barrier to direct passage of gas along the shaft. This fluid also performs the very important functions of lubricating the sleeves and removing heat from the seal area.

**FIG. 13-39**
**Active Magnetic Bearing System**

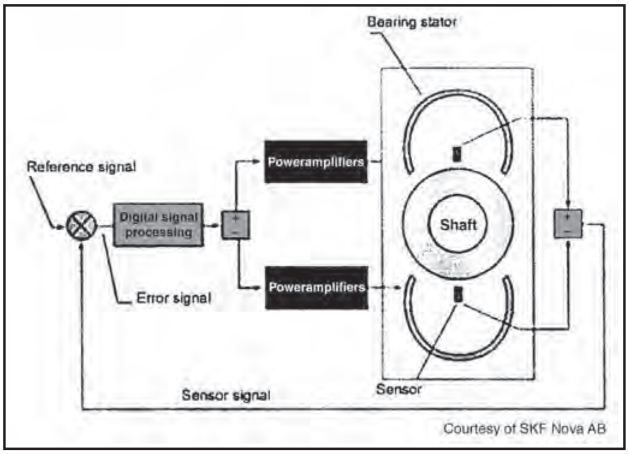



**FIG. 13-40**
**Mechanical (Contact) Shaft Seal**

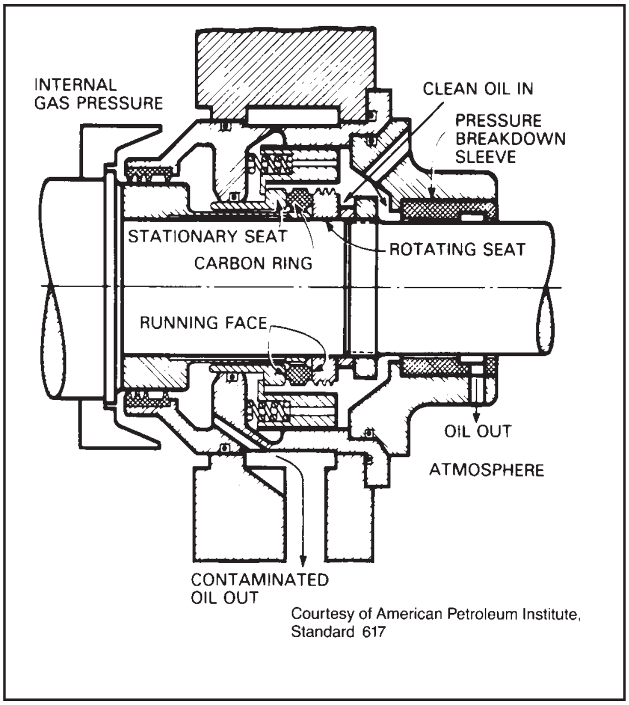



**FIG. 13-41**
**Single Gas Seal**

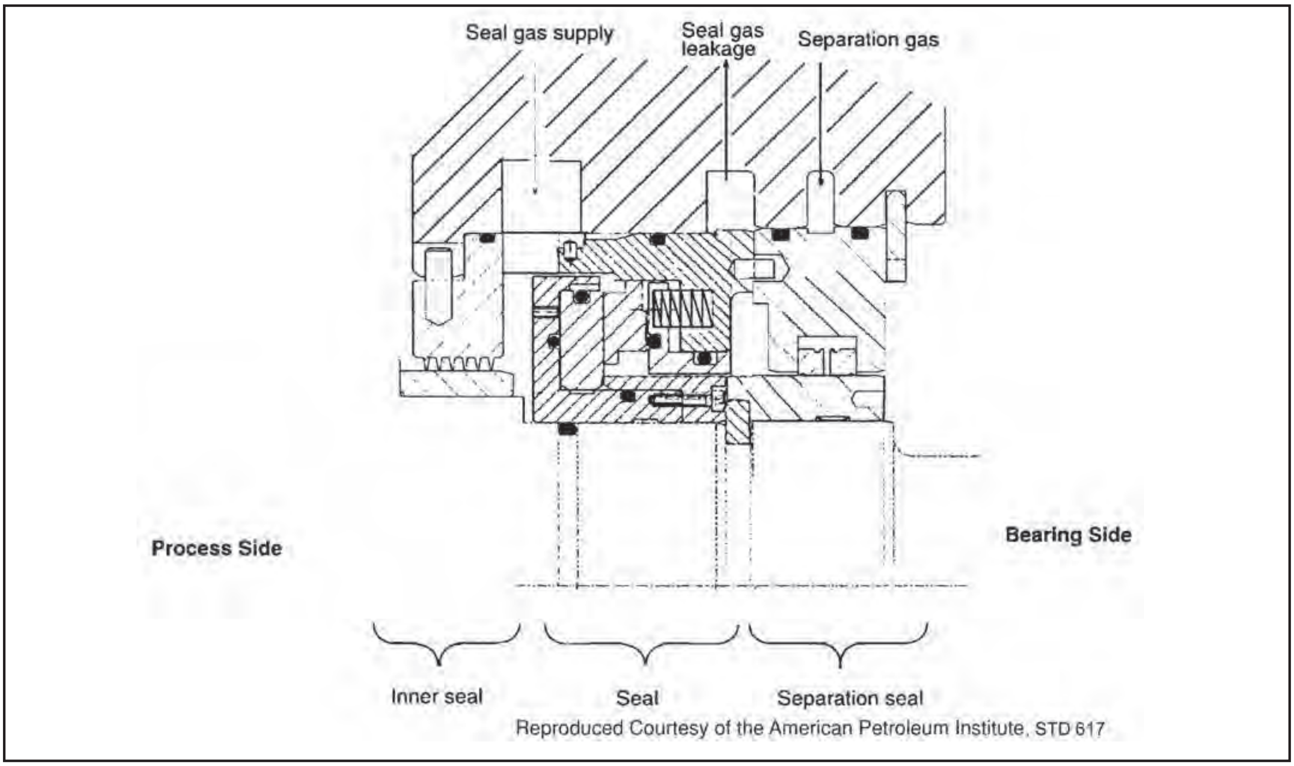


**FIG. 13-42**
**Double Gas Seal**

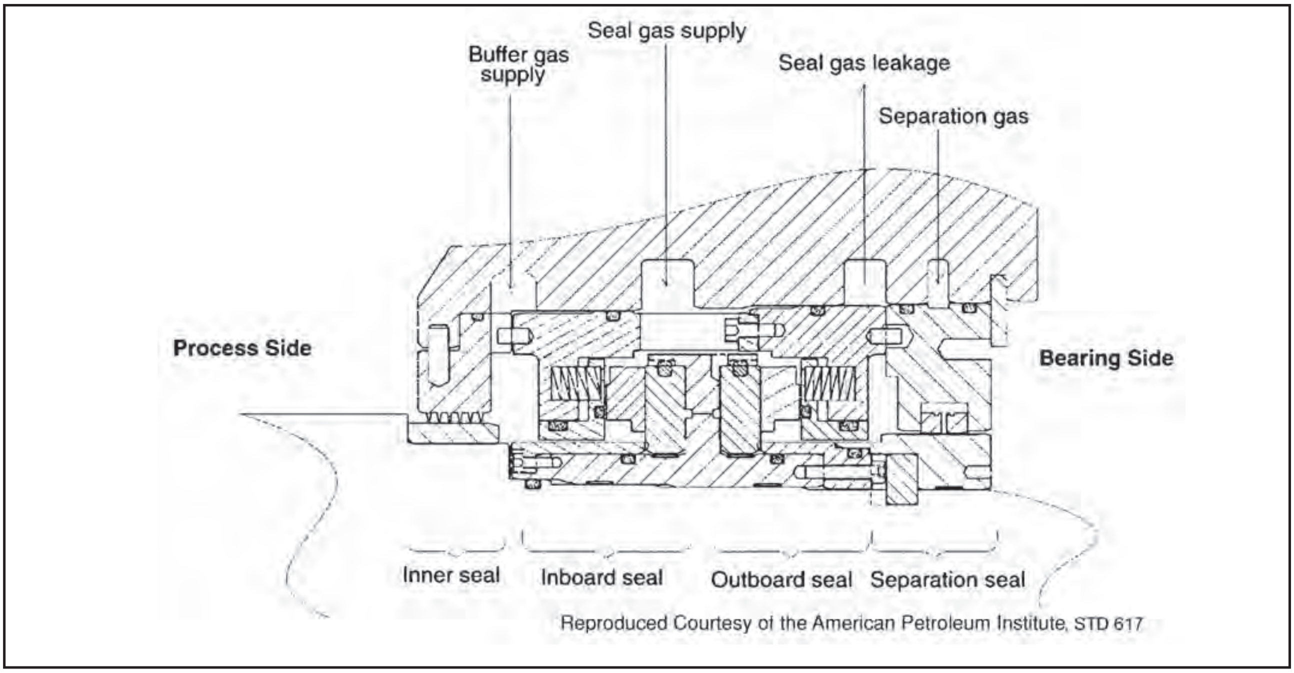
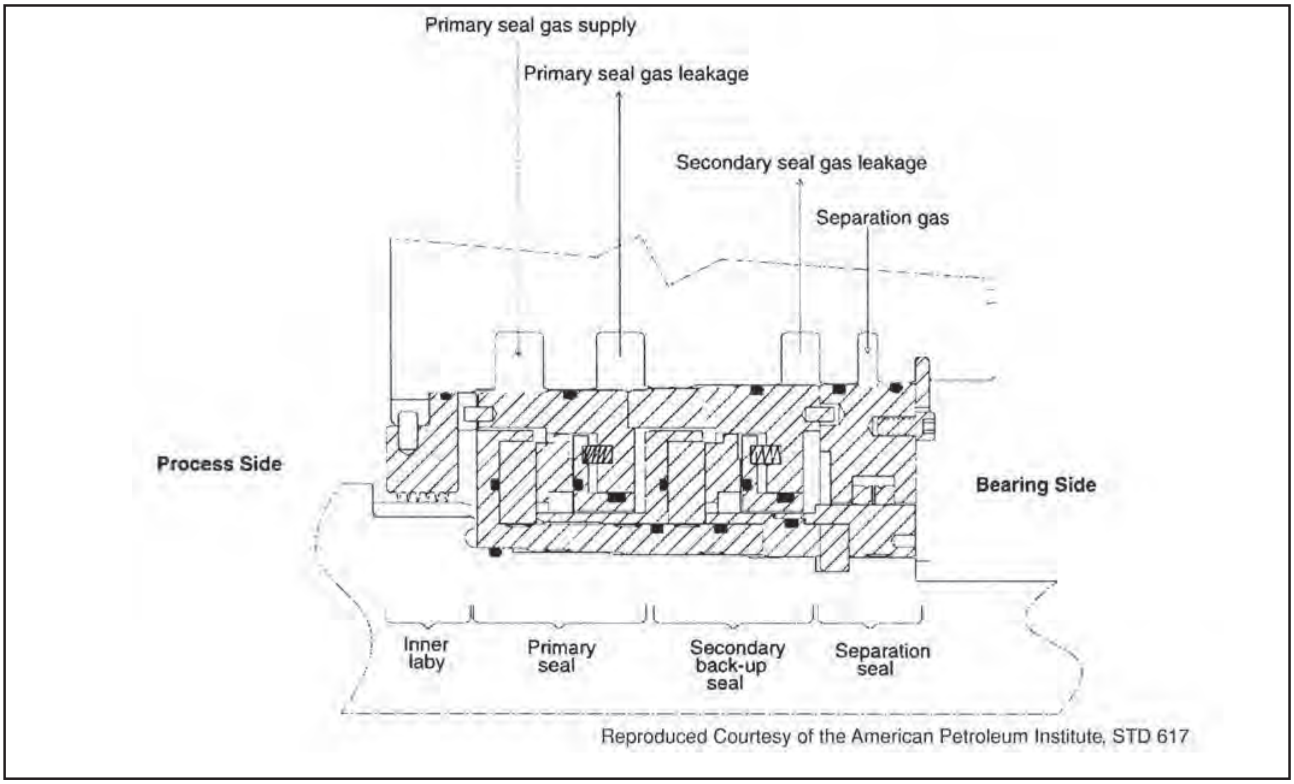



**FIG. 13-43**
**Tandem Gas Seal**



## Dry Gas Seals

Dry gas seals are used to prevent process gas from leaking along the rotating shaft of the compressor into the environment. Each seal consists of two rings, one of them a spring loaded seal face, the other a seat.

One ring is stationary with the compressor casing, the other rotates with the shaft. Silicon carbide, silicon nitride, tungsten carbide or carbon are typical materials for the seal rings.[2] The flat faces of the seal rings form the seal. Grooves, measuring a few microns in depth, are machined into one of the rings. When the machine is at stand-still the axially moveable seal ring is pressed on the other ring by the springs. Parting of the two faces is affected by the pressure differential across the seal faces, and the rotation of the shaft. During operation of the compressor, the forces from the springs, and the aerodynamic force created by the grooves due to gas flowing through the seal, are in equilibrium, and maintain a very narrow gap between the stationary and rotating face. Therefore, very low leakage can be maintained, while the fact that there is no mechanical contact between the rings avoids any seal deterioration, as long as the seal gas is free of solids and liquids.

The seals see, in general, 4 (four) modes of operation:

- **No Rotation, Case is Unpressurized:** The seal faces are held in contact by spring load.

- **No Rotation, Case Is Pressurized:** The seal faces remain in contact up to a certain pressure differential. At this pressure and above, the seal faces separate as the pressure overcomes the spring force between the faces. Normally, this lift-off pressure is 689 kPad (100 psid). Seal leakage is the same as, or less than, that during rotating conditions. This is the typical occurrence with pressurized equipment prior to start-up.

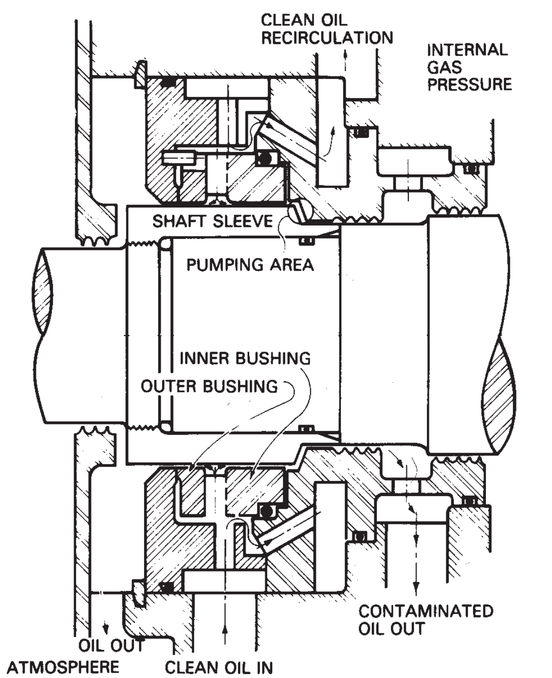
- **Rotation, Case is Unpressurized:** The seal faces remain in contact up to lift-off speed, which normally occurs at 150 rpm. Separation is caused by hydrodynamic effect due to the groove configuration in the face of the rotating seal member. This is the typical condition in an unpressurized seal at start-up.

- **Rotation, Case Is Pressurized:** The seal faces will maintain an equilibrium gap depending on the speed and pressure conditions.

For tandem dry gas seals, which are most commonly used in natural gas compression, we have two seals combined:

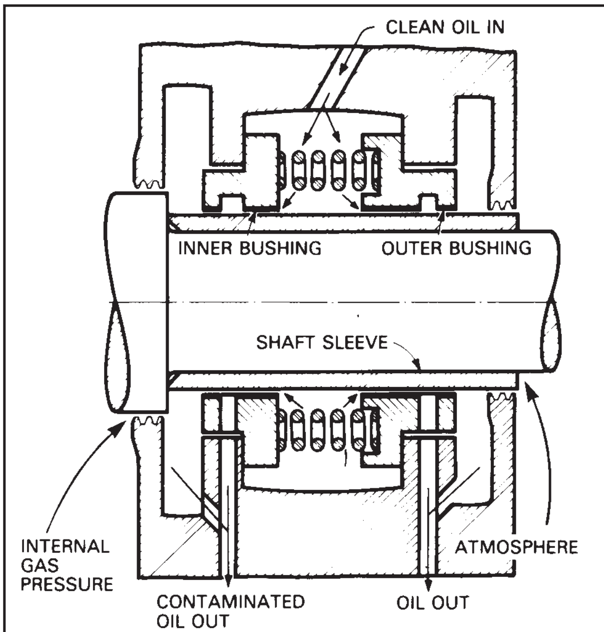
The primary face seal is exposed to the high-pressure seal gas on one side and approximately atmospheric pressure on the other, while the seal gas pressure is held slightly higher than the compressor suction pressure. By taking the full pressure drop, this seal provides the main sealing function. Filtered seal gas is injected between the process gas and the primary seal at a pressure nominally higher than the suction pressure. Most of the seal gas leaks into the compressor through the labyrinths at the shaft into the compressor suction flow. This portion of the primary seal gas is not lost, but is recycled. The quantity of this recycled gas is quite small (less than 0.1%) when compared to the compressor inlet flow; yet, it provides an important protective barrier for the dry seal. An even smaller portion of the primary seal gas leaks across the face seal to the primary seal vent. This leakage is lost to vent or flare. Both leakage rates, i.e., flow through the labyrinth and through the face seal, decrease as a fraction of compressor flow with increasing compressor frame size.

**FIG. 13-44**
**Liquid Film Shaft Seal with Pumping Bushing**




Courtesy of American Petroleum Institute, Standard 617

**FIG. 13-45**
**Liquid Film Shaft Seal with Cylindrical Bushing**




Courtesy of American Petroleum Institute, Standard 617

The secondary face seal acts as a backup to the primary face seal. It is similar to the primary seal and becomes active when the primary seal fails. It operates at near zero pressure-differential during normal running conditions. In order to protect the secondary face seal from failure, the secondary vent pressure should never be allowed to exceed the primary vent pressure. It is not necessary to inject seal gas ahead of the secondary seal as primary seal gas that leaks through the primary seal has already been filtered.

Some tandem dry seals also have an intermediate labyrinth seal located between the primary and secondary seals. The function of this intermediate labyrinth is to facilitate the use of a secondary seal gas. Secondary seal gas, usually an inert gas like nitrogen, may be injected between the secondary seal and the intermediate labyrinth. This gas also requires the same cleanliness as the primary seal gas.

The seal gas is usually process gas that has been filtered, and conditioned in the dry gas seal system. The dry gas seal system is set up to provide clean, filtered process gas to the seals. A typical dry gas seal system is designed to:

- Provide clean and dry seal gas to the face of the dry seal to prevent contamination and early failure of the seal.
- Monitor the leakage past the primary dry seal and alarm or shutdown if abnormal conditions exist.
- Provide clean air or nitrogen to the separation seals.
- Optionally, provide clean nitrogen to the intermediate labyrinth when needed.

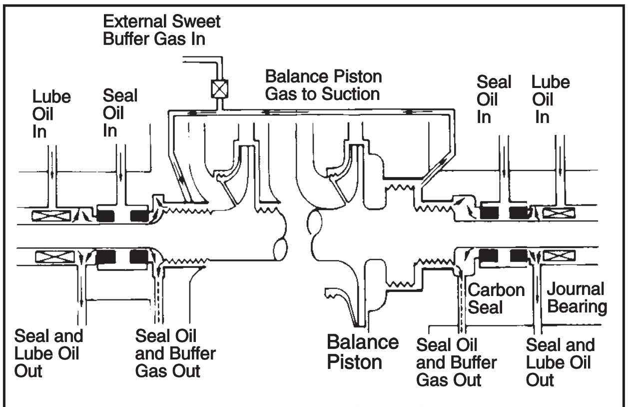
Dry gas seals need to be protected from lube oil migrating from the bearings of the compressor to the dry gas seals. This is accomplished by a separation seal (often referred to as buffer seal). This separation seal uses separation (or buffer) gas, usually air or nitrogen to avoid lube oil migration into the dry gas seal.

Compared with oil seal systems, dry gas seal systems have the following advantages:

- The dry gas seal system does not require external power source.

**FIG. 13-46**

**Combined Seal-Oil and Lube-Oil System with External Sweet Buffer Gas**




*Courtesy Solar Turbines International*

- Gas/Oil Interface. One of the main reasons for the interest in compressors with dry seals is that there is no process gas/lube oil interface. For transmission service, a dry seal system eliminates the addition of oil to the gas in the pipeline. For wellhead or field gas service, it eliminates sour gas carryover into lube oil tanks, oil degradation, and lube oil tank explosive mixture levels.
- Pressurized Hold. Pressurized holds of longer time are possible. As environmental limits become stricter, it will be increasingly advantageous to leave the compressor pressurized instead of blowing to vent at every shutdown.
- Degassing. Degassing flues/tank connections on wet seal units have a 127 mm (5 in.) of water column limit, while dry seal vent connections have a 34.5 kPag (5 psig) limit. This makes it much easier to capture and run leakage gas into a flare system.
- Seal Gas Quantity. The seal gas flow to the dry seal cavities is easier to limit and is less than buffer gas flows on wet seal compressors. The parasitic power requirement to compress seal gas is less with a dry gas seal system.

And disadvantages:

- The cost of dry seals is higher in comparison to oil seals.
- The dry seal cavities must have clean, dry gas to avoid contaminating the seals.

## Lubrication and Seal-oil Systems

On all centrifugal compressors that have force-feed lubricated bearings, a lubrication oil system is required. When oil-film or mechanical (contact) seals are used, a pressurized seal-oil system must be provided.

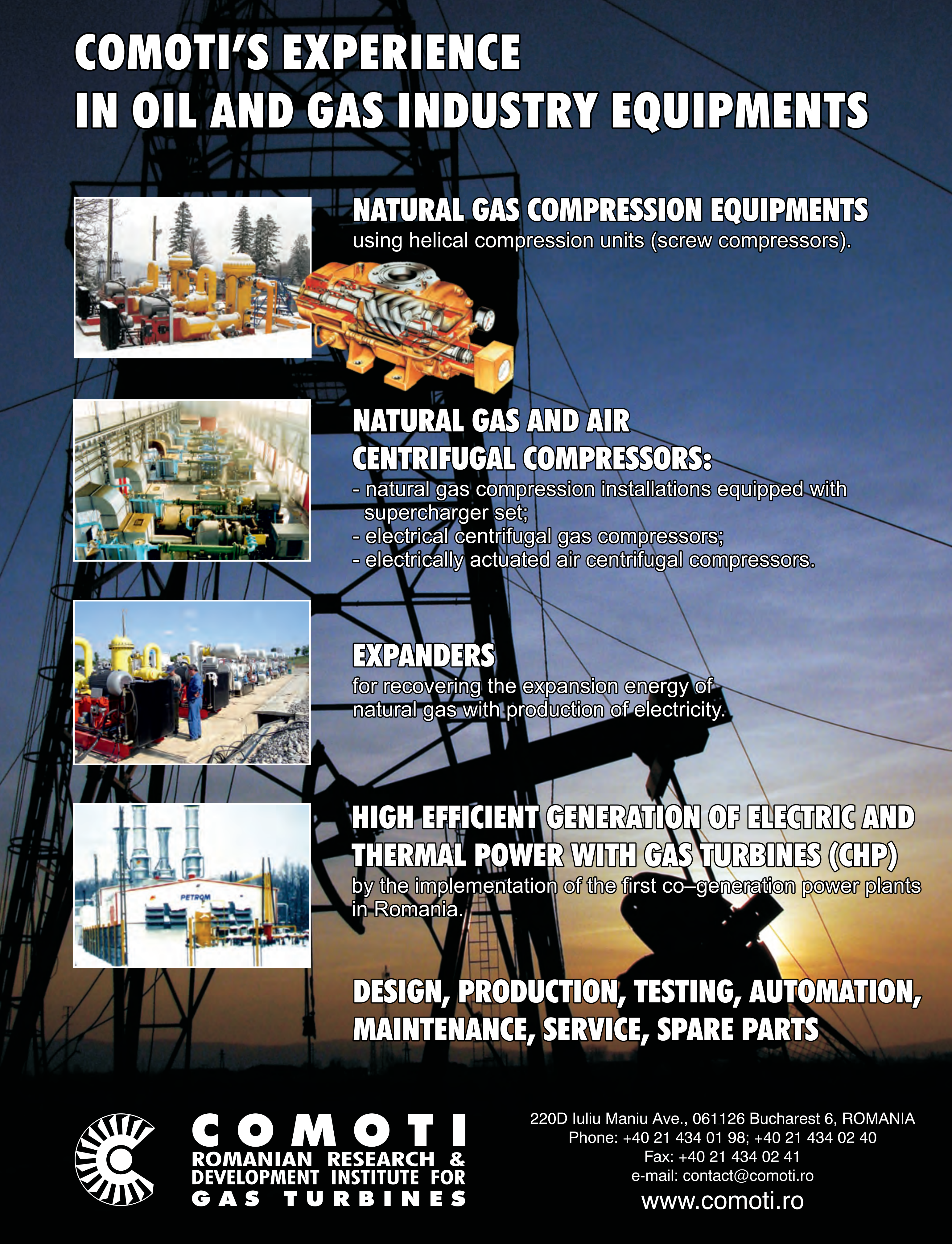
Each system is designed for continuous operation with all the elements (oil reservoir, pumps with drivers, coolers, filters, pressure gauges, control valves, etc.) piped and mounted on a flat steel fabricated base plate located adjacent to the compressor. The compressor manufacturer normally supplies both systems in order to have overall unit responsibility.

Depending on the application, lubrication and seal-oil systems may be furnished as combined into one system, or as one lubrication system having booster pumps to increase the pressure of only the seal oil to the required sealing level. In service involving heavily contaminated gases, separate lube-oil and seal-oil systems should be used.

The lubrication system may supply oil to both compressor and driver bearings (including gear), couplings (if continuously lubricated), as well as turbine governor, trip and throttle valve, and hydraulic control system.

A single lubricant shall be used in all system equipment, usually an oil, having approximate viscosities of 150 Saybolt Universal Seconds (SUS) at 100°F and 43 SUS at 210°F.

In addition to all the elements of a common pressurized lubrication system, the seal oil system requires a collection system for the oil. Depending on the gas composition, a degassing tank may be installed in the seal oil trap return line to remove the oil-entrained gas prior to return of the seal oil to the common oil reservoir. The flow past the outer sleeve passes through an atmospheric drain system and is returned to the reservoir. The relatively low flow through the inner sleeve is collected in a drain trap or continuous drainer and may be returned to the reservoir or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discarded, depending upon the degree and type of contamination which occurred while it was in contact with the internal gas.

Compressors using only liquid film seals should be provided with a seal-oil system which incorporates an overhead surge tank. The surge tank provides seal-oil capacity for coastdown of the machine and blowdown of the gas present in case of a compressor shutdown.

In combined seal-oil and lube-oil systems when large amounts of contaminants are present in the process gas, the seal-oil design may call for buffer gas injection to form a barrier between the compressed gas and the seal oil.

Fig. 13-46 shows clean sweet buffer gas being injected into the center of a labyrinth seal preceding the oil film seal with seal oil supplied between the two sleeves. Part of the seal oil flows across the inner sleeve and mixes with buffer gas and then drains into the seal oil trap. The other part of the seal oil flows across the outer sleeve, mixes with the bearing lube oil drain flow, and returns to the common lube- and seal-oil reservoir.

**FIG. 13-47**

**Pressure Control at Variable Speed**

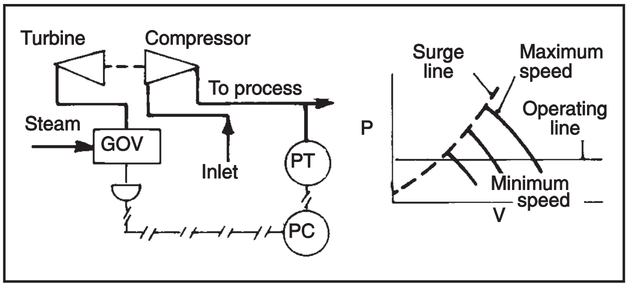



**FIG. 13-48**

**Volume Control at Variable Speed**

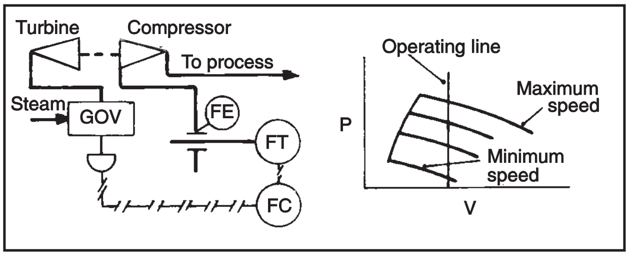
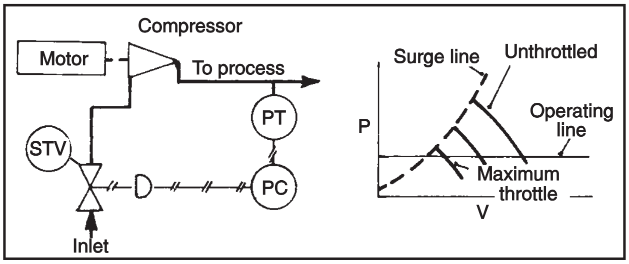



**FIG. 13-49**

**Pressure Control at Constant Speed**




**Drivers** — Centrifugal compressors can be driven by a wide variety of prime movers including electric motors, steam turbines, gas combustion turbines, and gas-expander turbines. Each driver has its own design parameters. A motor drive presents limitations in operation of the compressor due to constant and low speed. The constant speed restriction is minimized by suction or discharge throttling. The low speed restriction is corrected by introduction of a speed increasing gear. A steam turbine, on the other hand, has variable speed capability that allows more control of the compressor capacity or discharge pressure, and its high speed permits the compressor to be directly connected to the driver. In the case of a single-shaft gas turbine, the power output is limited at a reduced speed.

## CONTROL SYSTEMS

Centrifugal compressor controls can vary from the very basic manual recycle control to elaborate ratio controllers. The driver characteristics, process response, and compressor operating range must be determined before the right controls can be selected.

The most efficient way to match the compressor characteristic to the required output is to change speed in accordance with the fan laws (affinity laws, see Equations 13-23 and 13-24):

**FIG. 13-50**

**Volume Control at Constant Speed**

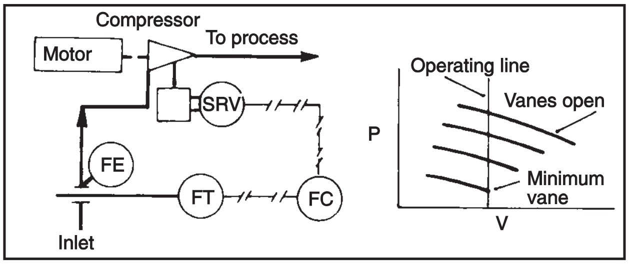
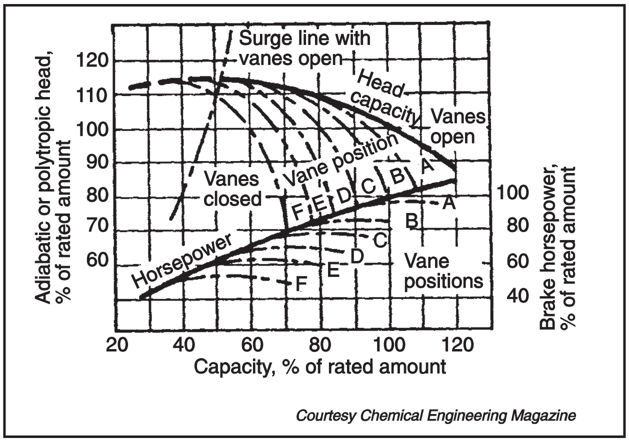



**FIG. 13-51**

**Effect of Adjustable Inlet Guide Vanes on Compressor Performance**



$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \frac{\sqrt{H_1}}{\sqrt{H_1}} \quad \textbf{Eq 13-44}$$

One of the principal advantages of using steam or gas turbines as drivers for compressors is that they are well suited to variable-speed operation. With such drivers, the speed can be controlled manually by an operator adjusting the speed governor on the turbine or, alternatively, the speed adjustment can be made automatically by a pneumatic or electric controller that changes the speed in response to a pressure or flow signal.

## Pressure Control at Variable Speed

The control system operates as follows:

The pressure transmitter (PT) in Fig. 13-47 senses the process discharge pressure. It converts this signal to a signal proportional to the process pressure and sends it to the pressure controller (PC).

The pressure controller amplifies the transmitter signal and sends a modified signal to the final control element. Depending on system requirements the controller may require additional correction factors called integral (reset) and rate.

The final element in this case is speed control. This varies the turbine-governor speed setting within a predetermined range.

As the load decreases, the discharge pressure will rise. An increase in process pressure above the set-point value will cause the signal to reach the governor and reduce the speed, maintaining the desired system discharge pressure.

## Volume Control at Variable Speed

If the nature of the process requires constant volume delivered, then the arrangement shown in Fig. 13-48 would be used.

Here, the flow transmitter (FT) senses the process flow, converts the signal to a signal proportional to the process flow, and sends it to the flow controller (FC).

The flow controller amplifies the transmitter signal and sends a modified signal to the final element. Integral (reset) and rate correction factors may be needed.

The final element is speed control, which is accomplished by a mechanism that varies the turbine-governor speed setting. An increase in flow over set point would cause a signal to reach the governor and reduce the speed to maintain the desired system flow.

When using electric motors as constant speed drivers (Fig. 13-49), the centrifugal compressor is normally controlled by a suction throttling device such as butterfly valve or inlet guide vanes. Throttling the suction results in a slightly lower suction pressure than the machine is designed for, and thus requires a higher total head if the discharge pressure remains constant. This can be matched to the compressor head-capacity curve, i.e., higher head at reduced flow. In throttling the inlet, the density of the gas is reduced, resulting in a matching of the required weight flow to the compressor inlet-volume capabilities at other points on the head/capacity curve.

## Pressure Control at Constant Speed

The control system shown in Fig. 13-49 has the pressure signal sensed and amplified in a similar manner as described in the scheme for variable speed control (Fig. 13-47).

The final element is a suction throttle valve (STV) that reduces the flow of gas into the compressor.

A process pressure increase over a set value would cause a signal to reach the suction throttle valve (STV) and would partially close the valve in order to reduce the inlet pressure.

## Volume Control at Constant Speed

The control scheme for this arrangement is shown in Fig. 13-50.

The flow transmitter (FT) senses the process flow using an orifice or venturi as the primary flow element (FE), converts this to a signal that is proportional, and sends this signal to the flow controller (FC). The flow controller amplifies the transmitter signal and sends a modified signal to the final element. Reset and derivative controller actions may be required.

The final element is the compressor guide-vane mechanism. The guide vanes are adjusted by means of a positioning cylinder. This cylinder is operated by a servo-valve (SRV) that receives a signal from the flow controller.

Here, an increase in flow above the set point causes a signal to reach the final element, which will result in the required degree of closing of the guide vanes to decrease flow.

**Adjustable Inlet Guide Vanes** — The use of adjustable inlet guide vanes is the most efficient method of controlling a constant speed compressor. The vanes are built into the inlet of the 1st stage, or succeeding stages, and can be controlled through the linkage mechanism either automatically or manually.

The vanes adjust the capacity with a minimum of efficiency loss and increase the stable operating range at design pressure. This is accomplished by pre-rotation of the gas entering the impeller which reduces the head-capacity characteristics of the machine. Fig. 13-51 illustrates the effect of such control at various vane positions.

Prior to control selection, the economics of inlet guide vanes must be considered because of their higher initial cost, complex mechanism, maintenance, and requirement for frequent adjustment.

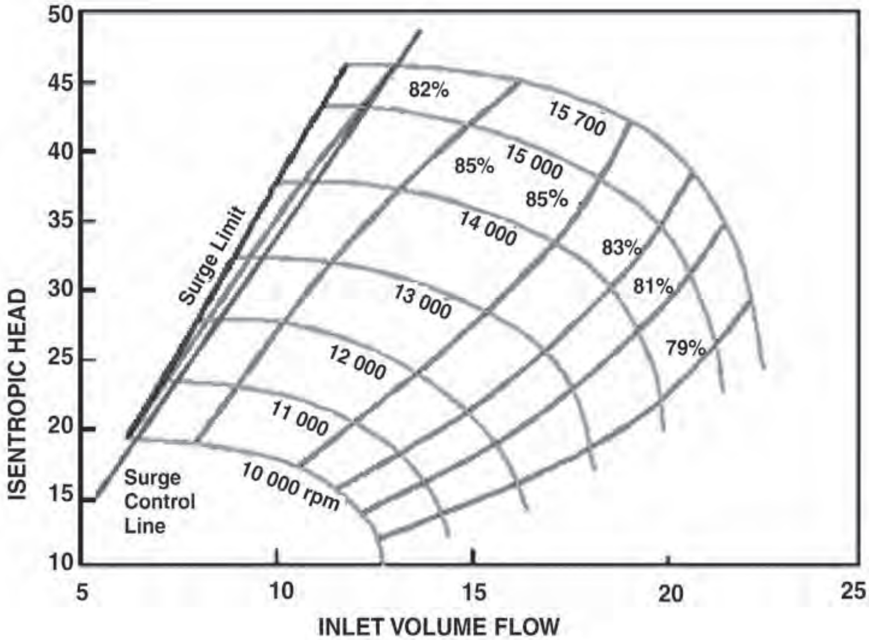
## Anti-surge Control

Surge Control systems are by nature surge avoidance systems. In general, the control system sensors measure the gas flow through the compressor and the head it generates. To determine compressor head, pressure and temperatures at suction and discharge are measured. The knowledge of head and flow allows the comparison of the present operating point of the compressor with the predicted surge line (Fig. 13-52). If the process forces the compressor to approach the surge line, a recycle valve in a recycle line is opened. This allows the actual operating point of the compressor to move away from surge ( Kurz and White, 2004).

One of the complications is, that the calculation of head and flow from pressure differentials over a flow element, and suction and discharge pressures and temperatures (as described earlier), requires the knowledge of the gas composition. In many applications, the gas composition can change. However, by normalizing the flow and the head appropriately (White and Kurz,2006) a surge limit line can be defined that is invariant to changes in gas composition.

A key issue in surge control is the accuracy of the flow measurement. It is therefore recommended to use properly installed

**FIG. 13-52**
**Typical Compressor Map (Variable Speed)**




Courtesy of Solar Turbines Incorporated

orifices or venture flow meters. Using the pressure differential between compressor flange and impeller eye is also a very effective method. Properly installed ultrasonic flow meters have also been used successfully. It is not recommended to use pitot type or elbow flow meters for flow measurements in surge control systems because the signals tend to be weak, with a low signal-to-noise ratio.

A surge avoidance system determines the compressor operating point using the pressure, temperature and flow data provided by the instrumentation. The system compares the compressor operating point to the compressor's surge limit. The difference between the operating point and the surge limit is the control error. A control algorithm (P+I+D) acts upon this difference, or "error," to develop a control signal to the recycle valve. When opened, a portion of the gas from the discharge side of the compressor is routed back to the suction side and head across the compressor is prevented from increasing further. When the operating point reflects more flow than the required protection margin flow, the surge control valve moves toward the closed position and the compressor resumes normal operation.

There are 5 essentials for successful surge avoidance:

1. A precise surge limit model: It must predict the surge limit over the applicable range of gas conditions and characteristics.
2. An appropriate control algorithm: It must ensure surge avoidance without unnecessarily upsetting the process.
3. The right instrumentation instruments must be selected to meet the requirements for speed, range, and accuracy.
4. Recycle valve correctly selected for the compressor: the valves must fit the compressor. They must be capable of large and rapid, as well a small and slow, changes in capacity.
5. Recycle valve correctly selected for the system volumes: The valve must be fast enough and large enough to ensure the surge limit is not reached during a shutdown. The piping system is the dominant factor in the overall system response. It must be analyzed and understood. Large volumes will preclude the implementation of a single valve surge avoidance system.

It must be understood that the anti-surge control system must be designed to operate under three, very different, scenarios:

1. Unit Startup: In this condition the recycle valve is typically kept at a fixed position to allow the compressor to start, and ultimately reach the discharge pressure necessary to open the check valve, and feed gas in to the process.
2. Process Control: with a properly sized recycle valve, a centrifugal compressor can stay on-line even at a no-flow condition. Well-designed surge control systems can allow reduction of the process flow to zero while keeping the compressor on line. This will also make the transition from fully closed recycle valve to an increasingly open recycle valve smooth and without upset to the process.
3. Emergency shutdown: During certain emergency situations, the compression units have to be shut down instantly. To that end, the fuel supply, electricity supply,

or steam supply to the driver are cut instantly. In this situation, the compressor will decelerate rapidly under its inertia. Typical a compressor may lose 30% of its speed in the first second. Because the speed reduction also reduces the head-making capability of the compressor, the recycle valve has to open quickly to relieve the pressure on the discharge side of the compressor.

In some instances it is necessary to use multiple loops or multiple valves in parallel to accomplish a system that both allows the necessary accuracy in flow control for process control, as well as the fast reaction for an emergency shutdown.

A typical anti-surge control system is shown in Fig. 13-53

The usual method for surge avoidance (“anti-surge control”) consists of a recycle loop that can be activated by a fast acting valve (“anti-surge valve”) when the control system detects that the compressor approaches its surge limit.

## Vibration Control System

This control system may be provided to monitor the driver behavior at the shaft bearings for detection of excessive lateral vibration and axial movement and for protection against possible machinery failure through alarm and/or shutdown devices.

The system may protect not only the compressor but also the driver, such as a steam or gas turbine, that usually runs at the same high speed as the compressor. When a speed increasing or reducing gear unit is furnished between the compressor and driver, also consider monitoring vibration at the gear shaft bearings.

The main system components are: variation transducer(s), signal amplifier(s) with d-c power supply, and vibration monitor and/or analyzer.

Vibration transducers fall into three categories: displacement probe, velocity pick-up, and accelerometer.

The displacement probe is most commonly used for equipment with high value, as it can measure shaft vibration relative to bearing housing. Output signal from each transducer is small and, therefore, it must be amplified before being transmitted to a vibration monitor or analyzer.

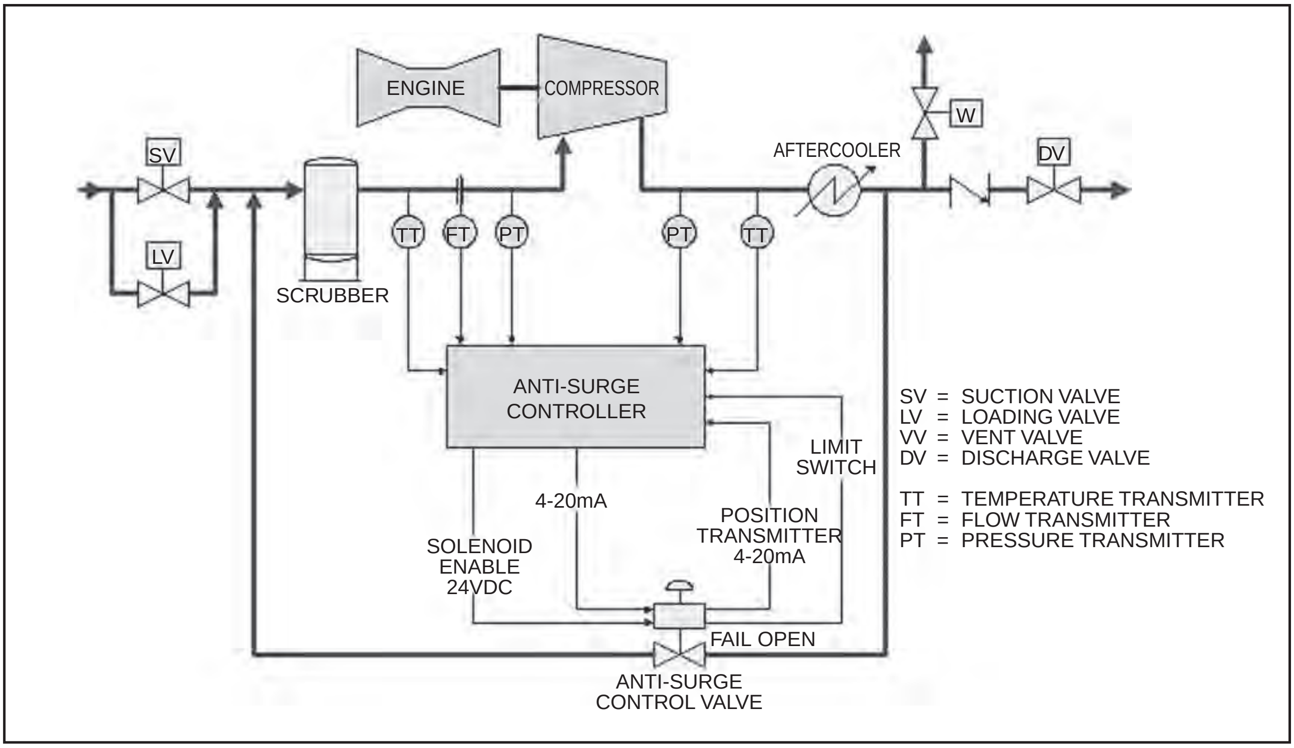
Fig. 13-54 shows a vibration severity chart for use as a guide in judging vibration levels as a warning of impending trouble.

For more information on vibration monitoring systems, see API Standard 670, Noncontacting Vibration and Axial Position Monitoring System, and API Standard 678, Accelerometer-Based Vibration Monitoring System.

Torsional analysis is also recommended for centrifugal compressors. The analysis is not as complex as for that required for reciprocating compressors due to the limited operating envelop of centrifugal compressor, and the fact that the energy sources are not as great as those within reciprocating compressors. Reference the discussion of torsional analysis in the Reciprocating Compressor section for additional information.

**FIG. 13-53**

**Example Anti-Surge Control System courtesy of Solar Turbines Incorporated**



# Howden advanced compressor technologies

**Reciprocating compressors**

**Packaged screw compressor systems**

**Multistage centrifugal compressors**

**Centrifugal compressors / turbo blowers**

**Rotary positive displacement blowers**

**Steam turbines**

**The widest range of custom-designed compression products, engineered for performance, reliability and process optimization in the most challenging applications.**

- Howden bareshaft and packaged screw compressors and single stage turbo blowers and compressors
- Thomassen heavy duty API618 reciprocating compressors
- Burton Corblin reciprocating, diaphragm, hybrid and Periflow® compressors
- Howden ČKD multistage centrifugal compressors
- Roots rotary positive displacement blowers and centrifugal compressors
- Turblex/HV-Turbo centrifugal compressors
- Kuhnle, Kopp & Kausch centrifugal compressors and turbo blowers
- Kuhnle, Kopp & Kausch / Howden steam turbines
- Spencer single and multistage centrifugal blowers, gas pressure boosters and vaccum systems

Revolving Around You™

## OPERATIONAL CONSIDERATIONS

### Rotor Dynamics and Critical Speeds

The demand for smooth-running turbomachinery requires careful analysis of rotor dynamics taking into account bearing performance, flexibility, critical speed, and rotor response.

Equally important is to analyze the dynamic behavior of the compressor for sudden changes in load due to start-up, shutdown, or loss of power supply.

**FIG. 13-54**

**Vibration Severity Chart[1]**

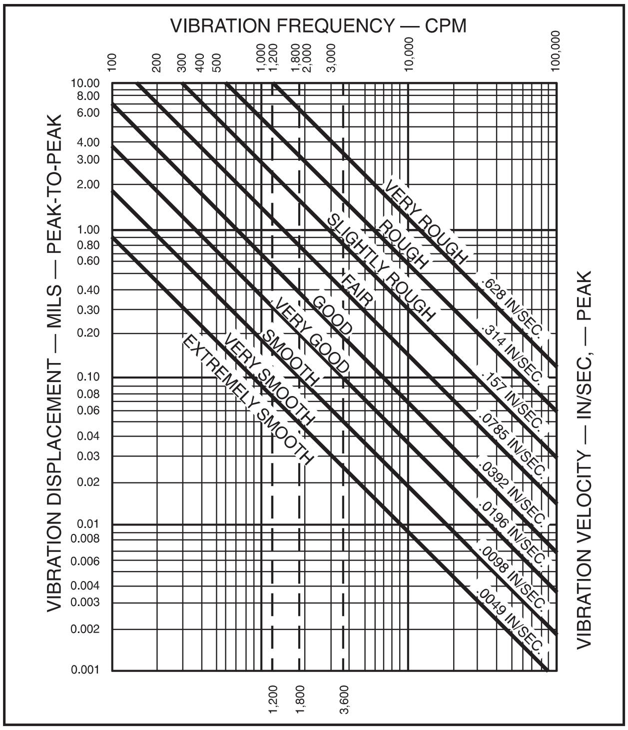
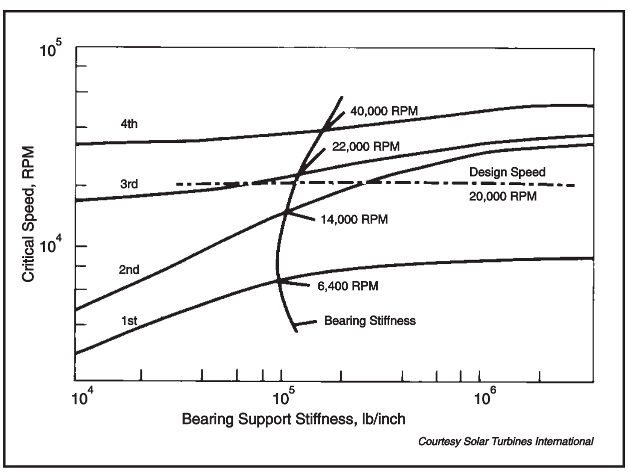



**FIG. 13-55**

**Undamped Critical Speed Map**




Successful rotor design is the result of accurate calculation of critical speeds. A critical speed occurs at a condition when the rotor speed corresponds to a resonant frequency of the rotor-bearing support system. Under no circumstances should the compressor be allowed to run at a critical speed for a prolonged length of time as the rotor vibrations amplified by this condition can cause machinery failure.

### Critical Speed Map

A critical speed map is one of various methods used to predict the operational behavior of the rotor. First, the critical speeds for a given rotor geometry are calculated for a range of assumed bearing-support stiffness values. The result is a map like that shown in Fig. 13-55. The bearing stiffness characteristics are determined from the geometry of the bearing support system, and cross-plotted on the critical speed map.

The map depicts the values of the undamped critical speeds and how they are influenced by bearing stiffness. The intersections of the bearing stiffness curve and the critical speed lines represent the undamped critical speeds. The intersection points generally indicate margins between the criticals and the operating speed range.

However, the use of this map is very limited because it is based on a simplified undamped, circular synchronous analysis with no cross-coupled or unbalance effects. It is a good trending tool showing a machine's basic dynamic characteristics. It may not accurately depict peak response frequencies.

The critical speed map is used extensively because it enables determination of bearing or support stiffness by correlating test-stand data.

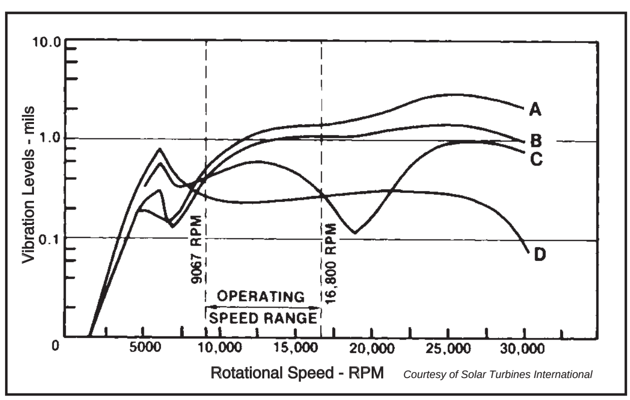
### Unbalance Response Analysis

This method predicts rotor-bearing system resonances to greater accuracy than the critical speed map. Here, bearing support stiffness and damping are considered together with synchronous vibration behavior for a selected imbalance distribution. A computer is normally required to solve the resulting differential equations. Satisfactory results depend on the accurate input of bearing stiffness and damping parameters.

Several runs are usually made with various amounts and locations of unbalance. The plot of results of a typical unbalance response study is shown in Fig. 13-56. Each curve represents the rotor behavior at a particular station or axial location such

**FIG. 13-56**

**Unbalanced Response Plot**



as those corresponding to the midspan, bearings, and overhangs.

No rotor can be perfectly balanced and, therefore, it must be relatively insensitive to reasonable amounts of unbalance.

The unbalance-response results predict the actual amplitudes that permit calculations of the unbalance sensitivity. This is expressed in mils of vibration amplitude per ounce-inch or gram-inch of unbalance.

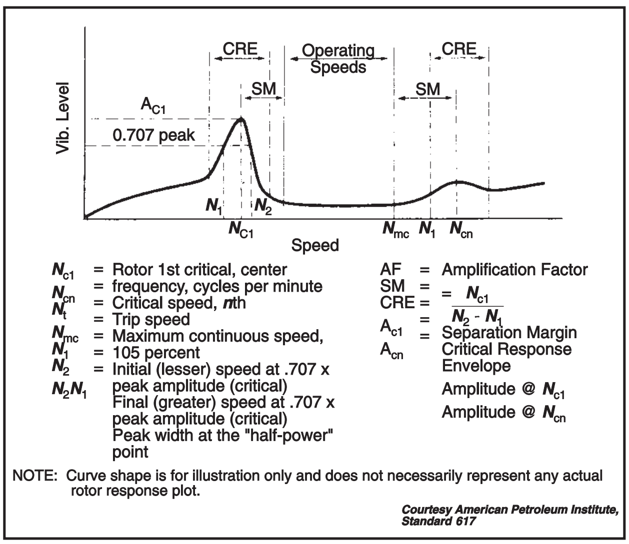
The peaks of the response curves represent the critical speed locations. Fig. 13-57 shows limits of placement of critical speeds as specified in the API Standard 617, Centrifugal Compressors for General Refinery Services.

Critical speeds should not encroach upon operating speed ranges, and the separation margin of encroachment (SM) from all lateral modes is required to be at least:

1. Twenty (20) percent over the maximum continuous speed for rigid shaft rotor systems.
2. Fifteen (15) percent below any operating speed and twenty (20) percent above the maximum continuous speed for flexible shaft rotor system.

**FIG. 13-57**

**Rotor Response Plot**




## Field Performance

Once the compressor has been installed, it is often desirable to measure its performance. The following parameters needs to be determined:

1. Inlet conditions (at the compressor inlet flange):
   a. Flow (scfm, acfm, or lb/min)
   b. Gas composition
   c. Pressure, psia
   d. Temperature, °F
2. Intermediate conditions (if applicable)
   a. Pressure at intermediate nozzles, psia
   b. Temperature at intermediate inlet flange, °F
   c. Flow (actual, mass flow, std flow) and gas composition at intermediate inlet flange if side stream is added or liquid drop-out occurs in the interstage cooler.
3. Discharge Conditions (at the compressor discharge flange):
   a. Pressure, psia
   b. Temperature, °F
4. Control setting (depending on the type of compressor controls)
   a. Compressor speed, rpm
   b. Guide vane setting
5. Driver Power
   a. If available, determine driver power output independently of compressor power measurement

## Troubleshooting

Operational troubles occurring in service may be due to a variety of causes.

If the trouble cannot be traced to adverse gas flow conditions or liquid "slugs" present in the system, Fig. 13-58 can be used as a guide for troubleshooting frequently encountered problems.

Careless operation and maintenance needs little comment. Lack of proper care of any machine is bound to result in a succession of minor troubles eventually leading to a major breakdown.

## INTEGRALLY GEARED COMPRESSORS

An integrally geared compressor utilizes a central driven bull gear with typically 2–4 high speed pinion-driven shafts. One or two impellers can be mounted on each pinion shaft. See Figures 13-59 and 13-60. This forms a compact unit for the multistage compression of a wide range of gases.

Integrally geared compressors offer the following potential advantages:

- low power consumption due to different impeller speeds, tailored aerodynamics and optimized auxiliaries.
- wide operating range and improved part-load efficiencies due to adjustable inlet guide vanes at the first or at all compression stages.
- multiservice capability.
- packaged designs available. A package includes the compressor, process coolers, lube oil console, process piping and all tubing and wiring.

Design requirements of integrally geared compressors are covered by API Standard 617.

## AXIAL COMPRESSORS

Axial compressors are basically high-flow, low-pressure machines, in contrast to the lower flow, high-pressure centrifugal compressors (the axial compressors used in gas turbines are often designed for higher pressures and compression ratios). Axial compressors are generally smaller and significantly more efficient than comparable centrifugal compressors. The characteristic feature of an axial compressor, as its name implies, is the axial direction of flow through the machine. An axial flow compressor requires more stages than a centrifugal due to the lower pressure rise per stage. In general, it takes approximately twice as many stages to achieve a given pressure ratio as would be required by a centrifugal. Although the axial compressor requires more stages, the diametral size of an axial is typically much lower than for a centrifugal. The axial compressor's capital cost is usually higher than that of a centrifugal but may be justified based on efficiency and size.

The axial compressor utilizes alternating rows of rotating and stationary blades to transfer the input energy from the rotor to the gas in order to generate an increase in gas pressure. A multistage axial flow compressor has two or more rows of rotating blades operating in series on a single rotor in a single casing. The casing contains the stationary vanes (stators) for directing the air or gas to each succeeding row of rotating blades. These stationary vanes, or stators, can be fixed or variable angle, or a combination of both.

A cross-sectional view of a typical axial flow compressor is shown in Fig. 13-61.

**Performance Capabilities** — The volume range of the axial compressor starts at approximately 30,000 cfm with a typical upper end of the flow range at 400,000 cfm. Much larger axial machines have been built. As can be seen in Fig. 13-3, the flow range for the axial overlaps the higher end of the range for

**FIG. 13-58**
**Probable Causes of Centrifugal Compressor Trouble**

| Trouble | Probable Cause(s) |
|---|---|
| **Low Discharge Pressure** | 1. Compressor not up to speed.<br>2. Excessive compressor inlet temperature.<br>3. Low inlet pressure.<br>4. Leak in discharge piping.<br>5. Excessive system demand from compressor. |
| **Compressor Surge** | 1. Inadequate flow through the compressor.<br>2. Change in system resistance due to obstruction in the discharge piping or improper valve position.<br>3. Deposit buildup on rotor or diffusers restricting gas flow. |
| **Low Lube Oil Pressure** | 1. Faulty lube oil pressure gauge or switch.<br>2. Low level in oil reservoir.<br>3. Oil pump suction plugged.<br>4. Leak in oil pump suction piping.<br>5. Clogged oil strainers or filters.<br>6. Failure of both main and auxiliary oil pumps.<br>7. Operation at a low speed without the auxiliary oil pump running (if main oil pump is shaft-driven).<br>8. Relief valve improperly set or stuck open.<br>9. Leaks in the oil system.<br>10. Incorrect pressure control valve setting or operation.<br>11. Bearing lube oil orifices missing or plugged. |
| **Shaft Misalignment** | 1. Piping strain.<br>2. Warped bedplate, compressor or driver.<br>3. Warped foundation.<br>4. Loose or broken foundation bolts.<br>5. Defective grouting. |

| Trouble | Probable Cause(s) |
|---|---|
| **High Bearing Oil Temperature**<br><br>Note:<br>Lube oil temperature leaving bearings should never be permitted to exceed 180°F. | 1. Inadequate or restricted flow of lube oil to bearings.<br>2. Poor conditions of lube oil or dirt or gummy deposits in bearings.<br>3. Inadequate cooling water flow to lube oil cooler.<br>4. Fouled lube oil cooler.<br>5. Wiped bearing.<br>6. High oil viscosity.<br>7. Excessive vibration.<br>8. Water in lube oil.<br>9. Rough journal surface. |
| **Excessive Vibration**<br><br>Note:<br>Vibration may be transmitted from the coupled machine. To localize vibration, disconnect coupling and operate driver alone. This should help to indicate whether driver or driven machine is causing vibration. | 1. Improperly assembled parts.<br>2. Loose or broken bolting.<br>3. Piping strain.<br>4. Shaft misalignment.<br>5. Worn or damaged coupling.<br>6. Dry coupling (if continuously lubricated type is used).<br>7. Warped shaft caused by uneven heating or cooling.<br>8. Damaged rotor or bent shaft.<br>9. Unbalanced rotor or warped shaft due to severe rubbing.<br>10. Uneven build-up of deposits on rotor wheels, causing unbalance.<br>11. Excessive bearing clearance.<br>12. Loose wheel(s) (rare case).<br>13. Operating at or near critical speed.<br>14. Operating in surge region.<br>15. Liquid "slugs" striking wheels.<br>16. Excessive vibration of adjacent machinery (sympathetic vibration). |
| **Water In Lube Oil** | 1. Condensation in oil reservior.<br>2. Leak in lube oil cooler tubes or tube-sheet. |

typical centrifugal compressor coverage. At the lower end of the axial's flow range, a thorough evaluation of axial vs centrifugal must normally be made. However, at the higher end flows, the axial compressor often becomes the obvious choice. As stated previously, the physical size of the axial is far smaller than the comparable centrifugal machine that would be required, and the efficiency of the axial is usually better. In many high flow applications, the axial is often a better match for the drivers that would typically be selected.

Because of the low pressure rise per stage, axial compressors are always manufactured as multistage machines. Axial compressors are in general low pressure machines. Typical discharge pressures are usually less than approximately 100 psig. They are very commonly utilized in refineries and other industrial processes for high volume, low pressure air supply applications. The most common application of axial compressors, besides aircraft jet engine use, is in gas turbines. In gas turbine applications, the axial air compressor is often designed to operate at final discharge pressures of up to around 500 psig.

Horsepower requirements for axial flow compressors in process service typically range from 3,000 to 65,000 HP for single casing units, depending on flow and pressure ratio requirements. Efficiencies for axial compressors are high, especially for larger machines, and can reach 90% (adiabatic).

Design requirements for centrifugal compressors are covered by API Standard 617.

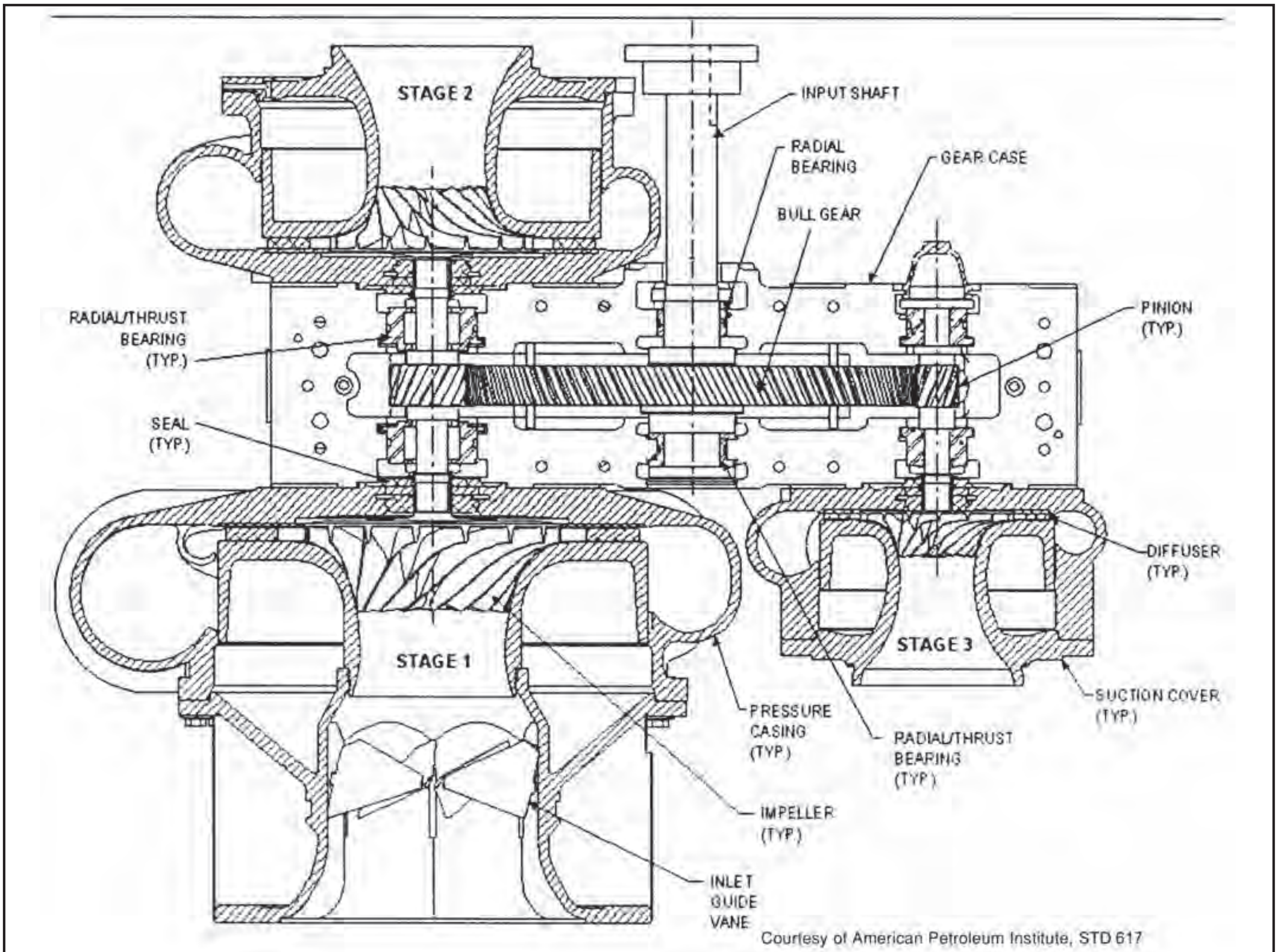
## SCREW COMPRESSORS

Screw compressors, also known as helical lobe compressors, fall into the category of rotary positive displacement compressors. Fig. 13-62 shows a cutaway cross-section of a typical rotary screw compressor.

Rotary screw compressors are available in oil-free (dry) or oil-injected designs. Oil-free compressors typically use shaft-mounted gears to keep the two rotors in proper mesh without contact. Applications for oil-free compressors include all processes that cannot tolerate contamination of the compressed gas or where lubricating oil would be contaminated by the gas. Oil-injected screw compressors are generally supplied without timing gears. The injected lubricant provides a layer separating the two screw profiles as one screw drives the other. Oil-injected machines generally have higher efficiencies and utilize

**FIG. 13-59**

**Typical Integrally Geared Compressor Showing Nomenclature of Key Parts**



Courtesy of American Petroleum Institute, STD 617

**FIG. 13-60**

**Typical Integrally Geared Compressor Arrangement Showing Nomenclature of Key Elements**

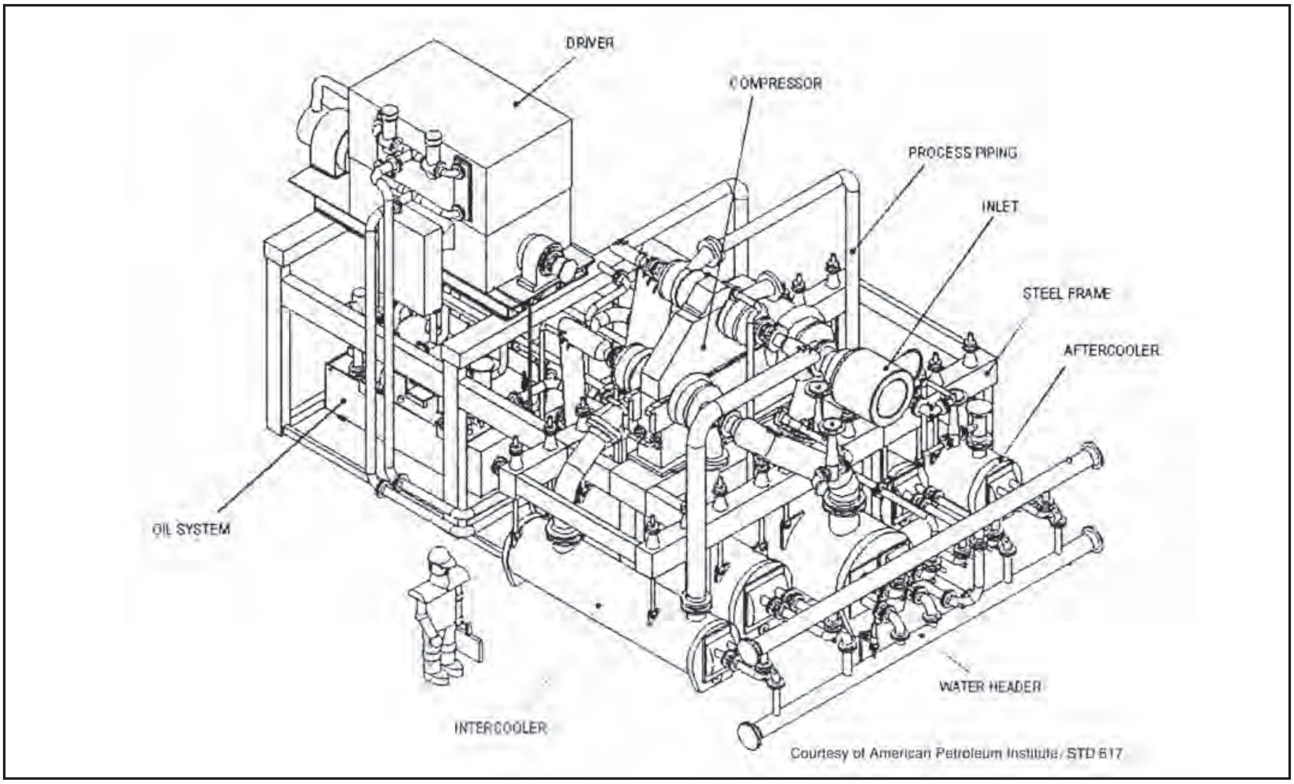
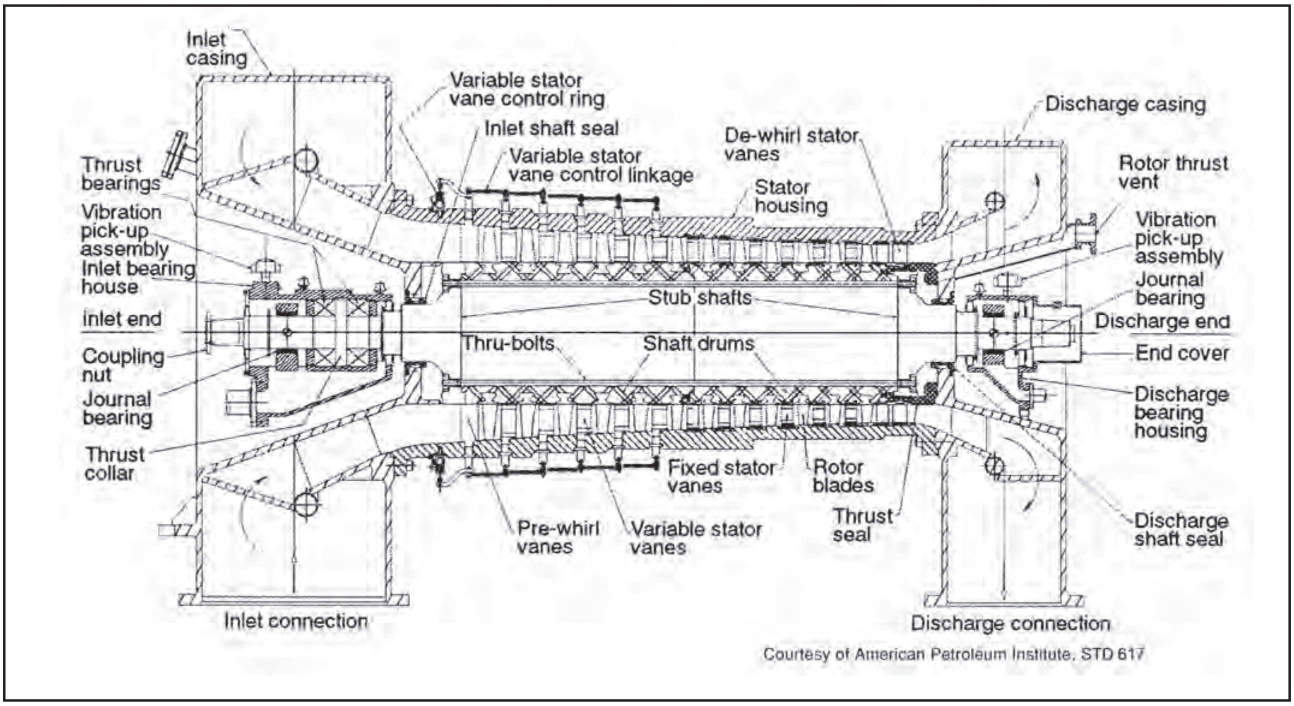



**FIG. 13-61**

**Typical Axial Compressor Showing Nomenclature of Key Parts**



# SAVE GAS AND PROTECT THE ENVIRONMENT
## WITH LMF INNOVATIVE TAILOR-MADE SOLUTIONS

### Compressor Design

- Gas and pressure tight up to 20barg/300psig
- TUV certified "emission-free compression of natural gas"
- Tightness ensured during operation and standstill
- ISO 13631 and API 618 compliant with lubricated or non-lubricated cylinders
- More than 30 years of experience with pressure tight compressors

### Applications

- Mobile and stationary compressor units for the evacuation of pipelines, turbines, filter housings
- Leakgas and vapor recovery units for your zero emission plant
- Patented design offers maximum flexibility and resource-saving benefits:
    - reduced evacuation time
    - energy saving
    - less residual release of gas
- Evacuation pressure of below 1 barg/15psig possible
- Energy independent and fully automatic operation

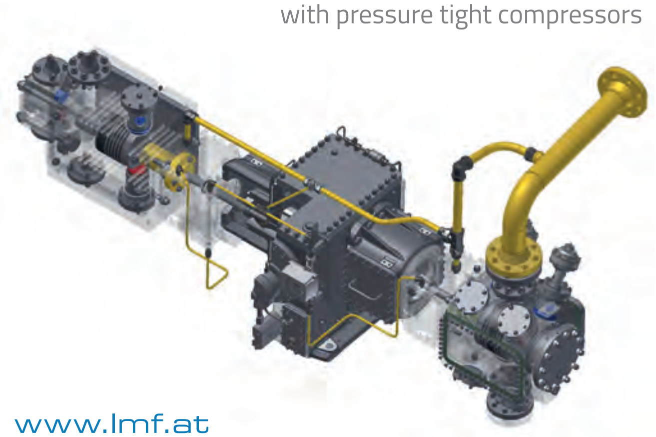


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**FIG. 13-62**
**Rotary Screw Compressor**

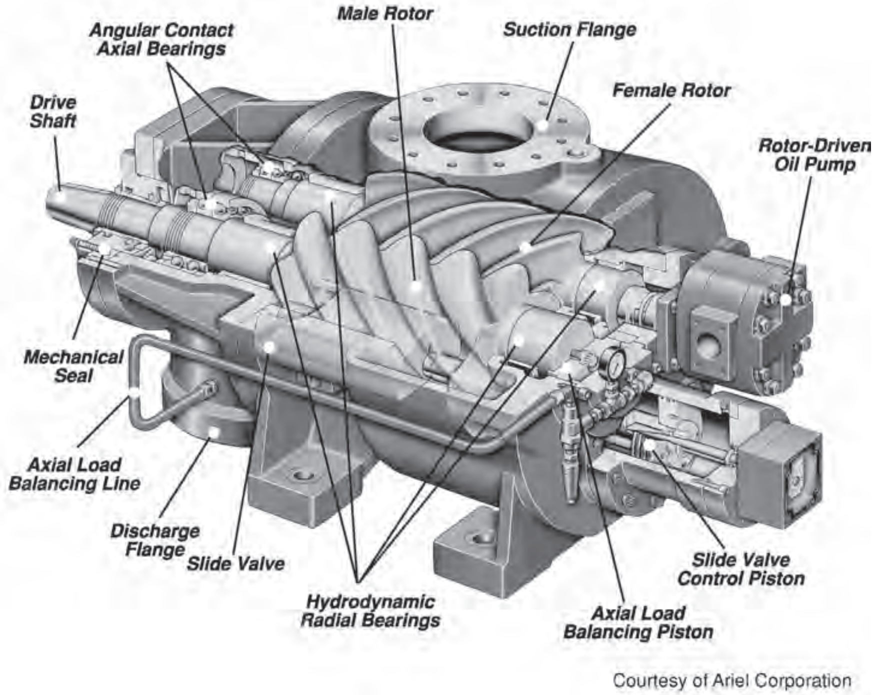
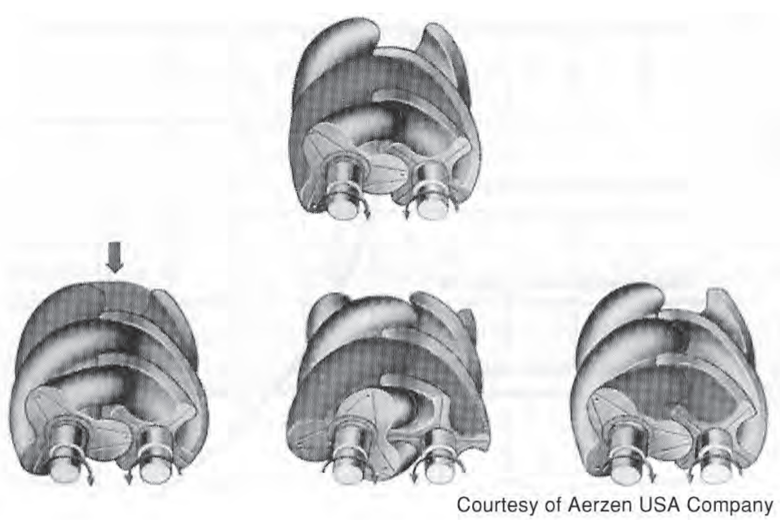



**FIG. 13-63**
**Working Phases of Rotary Screw Compressor**




(a) Suction intake. Gas enters through the intake aperture and flows into the helical grooves of the rotors which are open.

(b) Compression process. As rotation of the rotors proceeds, the air intake aperture closes, the volume diminishes and pressure rises.

(c) Discharge. The compression process is completed, the final pressure attained, the discharge commences.

the oil for cooling as well, which allows for higher compression ratios in a single screw compressor stage.

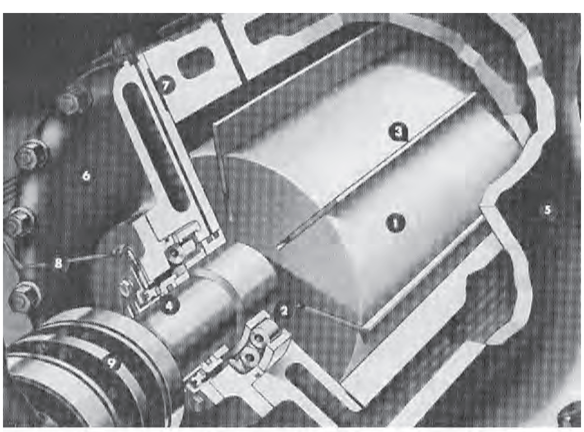
If an oil-injected compressor is used, the downstream oil separation is critical and often the cause for operating problems. A standard design should have primary separation and secondary separation using coalescing filters. Depending upon the process service, the oil content of the compressed vapor may need to be removed down to 100 ppb levels.

Although originally intended for air compression, rotary screw compressors are now compressing a large number of gases in the hydrocarbon processing industries. In particular, screw compressors are widely used in refrigeration service and are gaining in popularity in the gas production business in booster and gas gathering applications.

Gas compression is achieved by the intermeshing of the rotating male and female rotors. Power is applied to the male rotor and as a lobe of the male rotor starts to move out of mesh with the female rotor a void is created and gas is taken in at the inlet port. As the rotor continues to turn, the intermesh space is increased and gas continues to flow into the compressor until the entire interlobe space is filled. Continued rotation brings a male lobe into the interlobe spacing compressing and moving the gas in the direction of the discharge port. The volume of gas is progressively reduced as it increases in pressure. Further rotation uncovers the discharge port and the compressed gas starts to flow out of the compressor. Continued rotation then moves the remaining trapped gas out while a new charge is drawn into the suction of the compressor into the space created by the unmeshing of a new pair of lobes as the compression cycle begins again. Fig. 13-63 provides a sequence of drawings showing the compression process. Screw compressors are usually driven by constant speed motors, with capacity control normally achieved via an internal regulating device known as a slide valve. By moving the slide in a direction parallel to the rotors, the effective length of the rotors can be shortened. This provides smooth control of flow from 100 percent down to 10 percent of full compressor capacity.

**FIG. 13-64**

**Sliding Vane Compressor and Principal Components: Rotor and Shaft (1), Bearings (2), Blades (3), Mechanical Seals (4), Cylinder and Housing (5), Heads and Covers (6), Gaskets (7), Lube Supply Line (8), Coupling (9)**



Courtesy of A-C Compressor Corporation

Rotary screw compressors in use today cover a range of suction volumes from 180 to 35,000 acfm, with discharge pressures up to 750 psig. Typical adiabatic efficiency will be in the range of 70 to 80%.

Design requirements for screw compressors are covered under API Standard 619.

## ROTARY-SLIDING VANE COMPRESSORS

Rotary-sliding vane compressors (Fig. 13-64) are positive displacement machines. They have several applications, including vapor recovery and vacuum service. Each unit has a rotor eccentrically mounted inside a water jacketed cylinder. The rotor is fitted with blades that are free to move radially in and out of longitudinal slots. These blades are forced against the cylinder wall by centrifugal force. Fig. 13-65 illustrates how individual pockets are thus formed by the blades, and how the gas inside these pockets is compressed as the rotor turns. Oil is injected into the flow stream to lubricate the vanes, and is recovered via a downstream scrubber and recycled to the inlet.

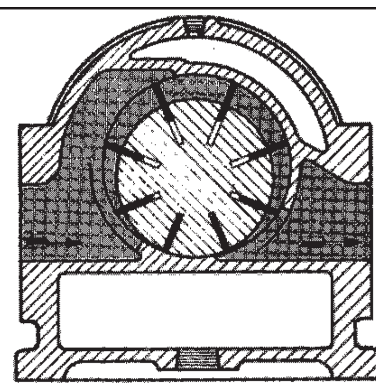
Sliding vane compressors are available in single- and multi-stage configurations. Typical single-stage capacities are ranging through 3200 cfm and 50 psig; two-stage compressors deliver pressures from 60 to 150 psig and flows up to approximately 1800 cfm. Most applications of rotary-sliding vane compressors in oil and gas service involve fairly small units, normally under 150 HP.

### Jet Pump Technology[1,2]

Jet pumps, also known as jet compressors, eductors or ejectors, are simple devices that use a high pressure (HP) fluid to increase the pressure of a lower pressure fluid (LP). In gas production, jet pumps have been successfully used in the following applications: boosting production of gas wells, preventing flaring of LP gas (vapor recovery), de-bottlenecking compressors, eliminating intermediate compressors, preventing HP wells from imposing back pressure on LP wells, and de-liquefication of liquid-loaded wells. In general, jet pumps are less efficient fluid movers as compared to a compressor or multi-phase compressor but their attractiveness is their low cost, tolerance to presence of some liquids in gas and their simplicity compared

**FIG. 13-65**

**Operating Principle of Sliding Vane Compressor**



Courtesy of A-C Compressor Corporation

to other systems such as compressors. If there is a local high pressure source, or a compressor with excess capacity or in a recycle mode, a jet pump can provide a cost effective solution to increase or maintain production or boost the pressure of low pressure (LP) processed gas. The primary components are a nozzle on the HP fluid side, a LP fluid inlet nozzle, a mixing tube and a diffuser, as shown in Fig 13-66.

In most gas production applications, the high pressure source is gas. The high pressure gas flows through the nozzle where some of the pressure (potential) energy is converted into kinetic energy (velocity). As a result, a low pressure zone is produced in front of the nozzle, at which point the low pressure fluid is introduced. The combined stream flows through the mixing tube to transfer momentum and energy between the two streams. The fluid is then expanded in a diffuser where the velocity of the fluid is reduced and pressure of the system is increased.

In vapor recovery applications, the high pressure source is sometimes a liquid. For example in well field applications, the produced water can be pumped up to high pressure and used to boost the LP gas pressure to gas pipeline pressures. The high pressure vapor and water stream are then separated, the vapor flows into the outlet gas pipeline, and the water is recycled for jet pump use.

The primary factors governing jet pump performance are the HP/LP pressure ratio (PR), and the LP/HP mass flow ratio. Other operating conditions, such as temperature and fluid physical properties will factor into the performance of the jet pump, but to a lesser extent. The resulting discharge pressure is primarily a function of the downstream production and process system.

Figure 13-67 provides typical performance of a jet pump under a range of gas pressures and flow ratios. In general, the LP pressure can be increased from a few percent up to five fold with a single jet pump. In "typical" applications the discharge pressure is 1.5 to 3 time greater than the low pressure source. The available LP/HP flow ratio is dependent upon the field installation and the HP source availability, but often it is 1:1 or less.

The performance of gas-gas jet pumps deteriorates if there are liquids present in the LP fluid. The reduction in performance is a result of the additional energy required to boost the pressure of the liquid phase, which has significantly more mass than the gas phase. In addition, increasing liquids in the LP stream can choke the flow of the jet pump due to the rapid decrease of the sonic velocity of the combined stream. The impact of liquids on performance is typically minimal up to 2 volume % liquid at operating pressure and temperature. Presence of liquids in the HP source is also problematic, as the liquids restrict flow through the nozzle. Gas-liquid separators, or other facility separators such as a test separator or a compact separator, may be used to separate the phases to achieve acceptable jet pump performance.

**FIG. 13-66**
**General Configuration of a Jet Pump**

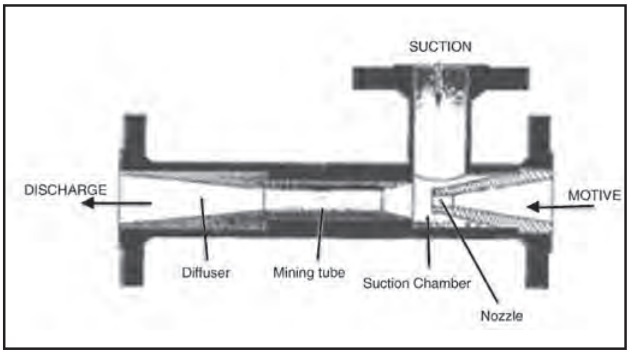
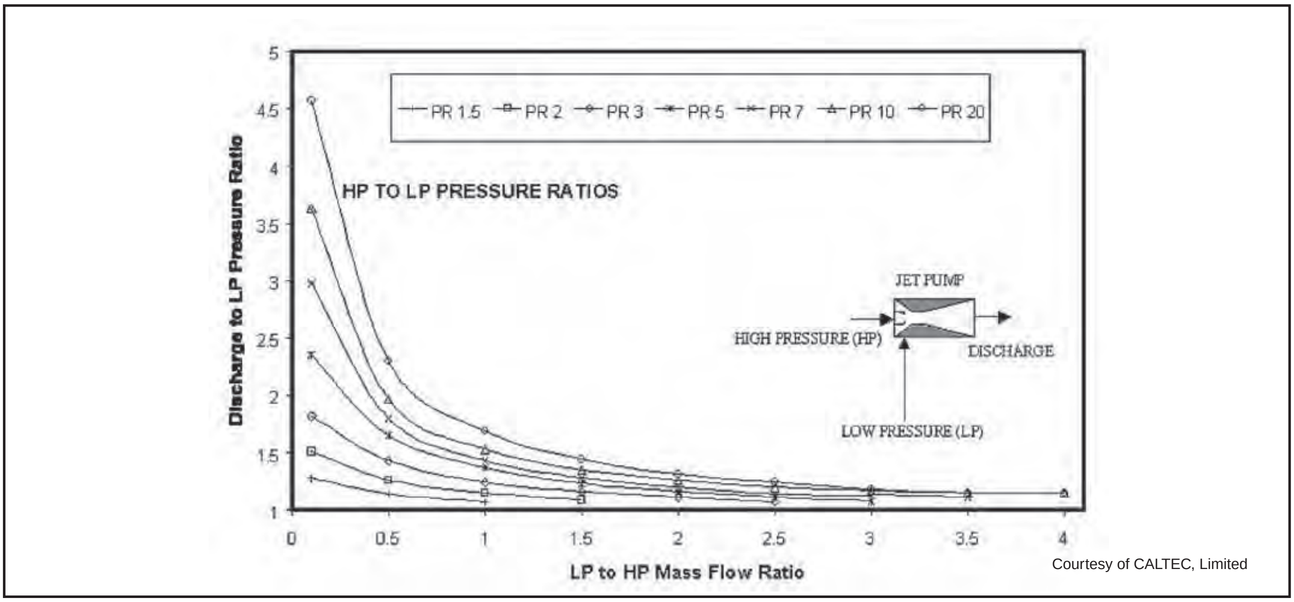



**FIG. 13-67**
**HP to LP Pressure Ratios**




Courtesy of CALTEC, Limited

# Turboexpanders

The use of turboexpanders in gas processing plants began in the early sixties. By 1970, most new gas processing plants for ethane or propane recovery were being designed to incorporate the particular advantages characteristic of an expander producing usable work and lower temperatures. This is due to the expander following an isentropic path as compared to an isenthalpic path of a JT valve, and thereby provide more effective cooling for a given pressure drop. The trend in the gas processing industry continues toward increased use of the turboexpander.

Current turboexpander process applications include: Hydrocarbon Dewpoint, NGL and LPG Recovery, LNG, Nitrogen Rejection, Helium Recovery, Air Separation Units, and Nitrogen Refrigeration Cycles. Section 16, Hydrocarbon Recovery, provides descriptions of a number of common turboexpander process applications for hydrocarbon recovery.

Selection of a turboexpander process cycle is indicated when one or more of the following conditions exist:

1. “Free” pressure drop in the gas stream.

2. High ethane recovery requirements (i.e., over 30% ethane recovery).

3. Compact plant layout requirement.

4. Flexibility of operation (i.e., easily adapted to wide variation in pressure and products).

There are multiple factors in addition to the ones listed above that affect a final process selection. If two or more of the above conditions are coexistent, generally a turboexpander process selection will be the best choice.

Fig. 13-68 shows a typical low temperature turboexpander process for recovering ethane and heavier hydrocarbons from a natural gas stream.

Fig. 13-69 represents the pressure-temperature diagram for this expander process. The solid curve represents the plant inlet gas. The solid line on the right is the dew point line. At a fixed pressure and, if the temperature of the gas is to the right of this dew point line, the gas is 100 percent vapor. If the gas is cooled, liquid starts to condense when the temperature reaches the dew point line. As cooling continues, more liquid is condensed until the bubble point line is reached — the solid line on the left. At this point, all of the gas is liquid. Additional cooling results in colder liquid.

Downstream of the gas treating facilities, the inlet gas is represented by point 1 on both Fig. 13-68 and 13-69. As the gas is cooled by the gas/gas exchangers and demethanizer side exchanger, its temperature moves along the dotted line to point 2 (Fig. 13-69). At 2, the gas enters the expander inlet separator where the condensed liquid is separated from the vapor. This vapor now has its own pressure-temperature diagram, as represented by the dashed curve. At the expander inlet, the gas is on its dew point line.

As the gas flows through the expander, its pressure-temperature path is shown by the dashed line from point 2 to point 3. Point 3 represents the outlet of the expander. The importance of using the expander as a driver for a compressor can be seen in Fig. 13-69. If the gas had been expanded without doing any driver work, the expansion path would be from point 2 to point 4. This is called a Joule-Thomson, or constant enthalpy expansion. The outlet temperature and pressure would be higher than that accomplished in the expander (nearly isentropic) expansion process.

Note that the pressure at Point 4 is not as low as that attained by flow through the expander (Point 3). This is because it has been assumed for this example that, without the expander running (therefore the brake compressor also not running), the process cannot restore the demethanizer overhead vapor to the residue gas pressure using the separate recompressor alone.

Also, because the path to Point 4 is adiabatic without the gas doing work, the gas does not cool to as low a temperature as the path to Point 3. That is, the path (2) to (3) is isentropic expansion producing work and thereby cooling the gas more than the simple isenthalpic (J-T) expansion path.

The higher temperature at Point 4 results in a reduction of product recovery. The use of the expander brake compressor to boost the residue gas pressure will allow a lower expansion pressure without the use of more residue compression.

## THERMODYNAMICS

A turboexpander (often just referred to as ‘expander’) recovers useful work from the expansion of a gas stream. The expander operates isentropically in the ideal case and produces something less than the theoretical work in the real case. In the process of producing work, the expander lowers the bulk stream temperature which can result in partial liquefaction of the bulk stream. A simple schematic of an expander is given in Fig. 13-70.

An example calculation of an expander operating on pure methane is provided to demonstrate the thermodynamic principles of expanders.

Gas inlet conditions ($t_1$, $P_1$) to the expander are generally set by upstream conditions. The outlet pressure $P_2$ from the expander is often set by the desired NGL recovery and recompressor power considerations. Fig. 13-71 gives an example calculation.

Outlet conditions for the expander processing a multi-component stream must be determined by trial-and-error calculations if one were to do them by hand.

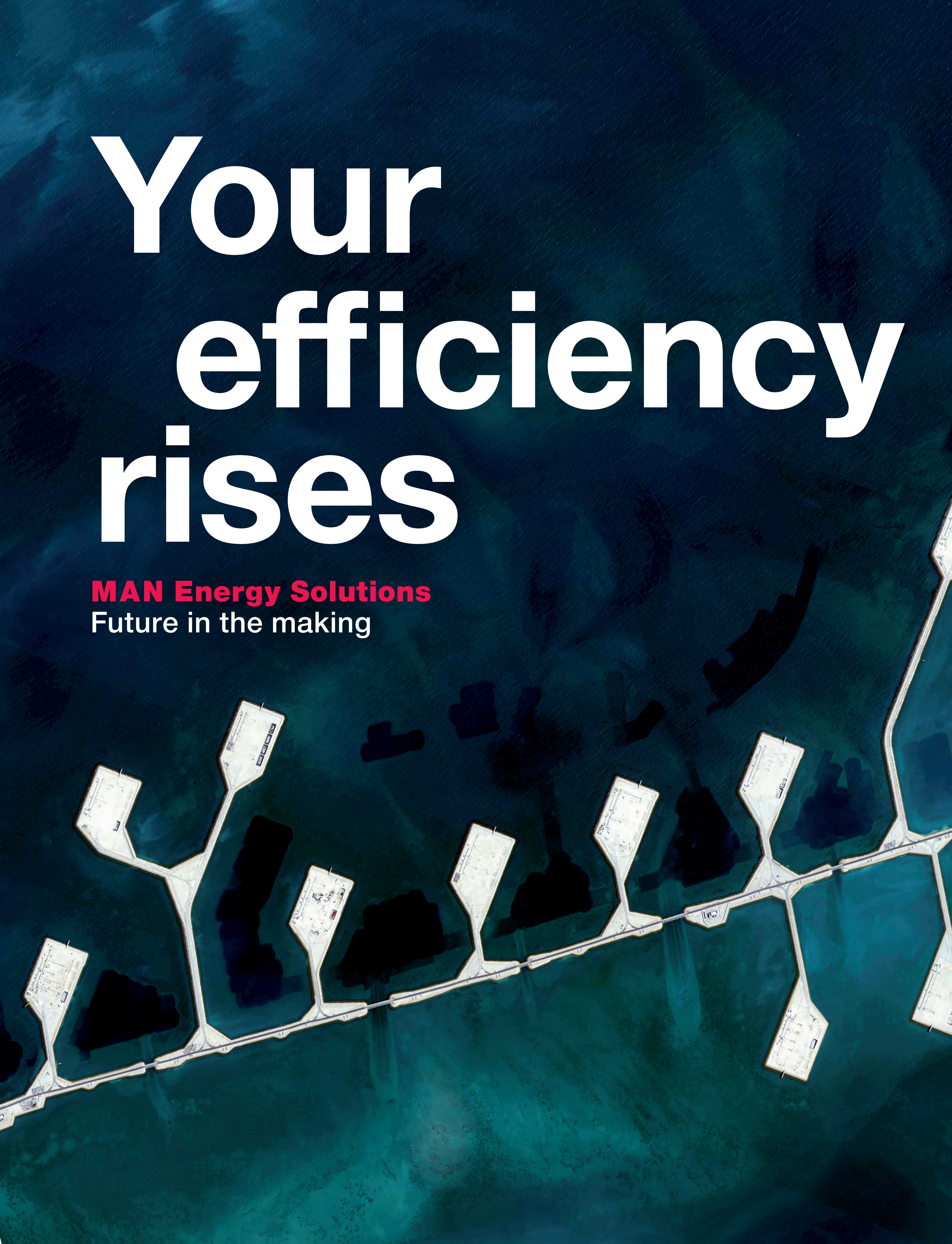
For multicomponent streams, such as natural gas, the hand calculations are iterative, tedious, and are only close approximations for expander performance. Expander and compressor performance is typically modeled using current process simulators.

In many applications the loading device for the turboexpander is a centrifugal compressor. Shaft and bearing losses in the order of 2% are usually deducted to calculate net power input to the driven end from the expander.

## MECHANICAL

Mechanical design of the turboexpander is the business of several manufacturers. Any specific information must come from such supplier.

Of the various general turbine types available, the radial reaction turbine design is dominant in cryogenic turboexpander natural gas plant applications. These units operate over wide ranges of inlet flow and pressure conditions, by utilizing vari-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able inlet guide vanes. They operate at very high rotating speeds and thus are subject to the design and operating cautions common to similar sophisticated rotating equipment.

The most common configuration is a turboexpander-compressor where the expander power is used to compress gas in the process. In this case, the compressor wheel operates on the same shaft as the expander wheel. Other applications of the power recovery are expander-pump or expander-generator drives. These normally require gearing to reduce the expander speed to that required for the driven unit.

Since power recovery and refrigeration effect are primary benefits of expander applications, rotating speeds are set to optimize the expander efficiency. This will usually result in a compromise in the compressor end design and lower compressor efficiencies. Usual efficiencies quoted for radial type units are 80 to 83% for the expander and 68 to 70% for the compressor.

Some areas requiring extra attention in the installation of turboexpanders are listed below. The list is by no means comprehensive, but these items require more than the normal amount of concern in designing the installation of a turboexpander unit for cryogenic operation.

1. The expander inlet gas stream must be free of solid or liquid entrainment. Liquids are removed in a high pressure separator vessel. An inlet screen of fine mesh is usually required for solids removal. Monitoring of the pressure drop across this screen is recommended. Formation of solids (ice, carbon dioxide, amines, heavy oils) will often occur here first and can be detected by an increase in pressure drop across the screen.

2. Source of the seal gas, particularly during start-up, is an important consideration. The stream must be clean, dry, sweet, and of sufficient pressure to meet the system requirements.

3. Normally a quick closure shutoff valve is required on the expander inlet. Selection of this valve and actuator type must take into account start-up, operating, and shutdown conditions.

4. Vibration detection instrumentation is useful but not mandatory. Its application is normally an owner and vendor option and influenced by operating economics.

5. Loading of the flanges by the process piping system must be within prescribed limits to avoid distortion of the case, resulting in bearing or wheel rubbing problems.

6. Failures due to mechanical resonance have occurred in turboexpanders. Even though the manufacturer will exert his best efforts at the manufacturing stage to avoid this problem, in-plant operation may uncover an undesirable resonance. This must be solved in conjunction with the manufacturer and may involve a redesign of the wheels, bearing modifications, vane or diffuser redesign, etc.

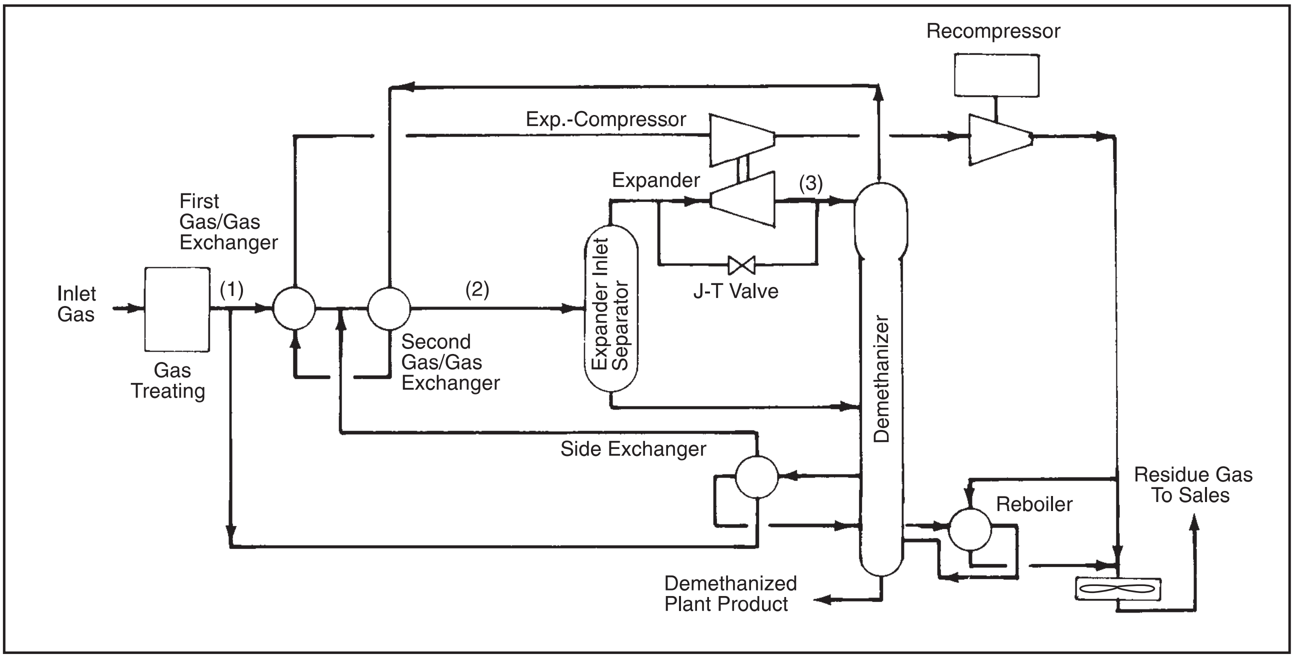
The installation of a turboexpander-compressor unit also requires the proper design of a lube system, instrumentation, etc., in common with other industrial rotating equipment. It is common practice to install a turboexpander-compressor with no special anti-surge instrumentation for the compressor unit. This is acceptable if it can be determined that the gas flow through the compressor is balanced with flow through the expander and the two will vary simultaneously.

## Auxiliary Systems

Both lubricated and non-lubricated turboexpander designs are available.

**Lubrication System** — The lubrication system circulates cooled and filtered lube oils to the turboexpander bearings as shown on Fig. 13-72. The principle components of the system

**FIG. 13-68**
**Example Expander Process**



are monitored on the lube console and normally consist of two electric motor-driven lube oil pumps, an oil cooler, a dual filter valve, a bladder type with switching coastdown accumulator, and a pressurized reservoir with mist eliminator.

The lube oil pumps (one stand-by) must maintain a constant flow to the radial and thrust bearings. Absence of oil, or improper filtration, can cause bearing damage. Most manufacturers recommend a light turbine oil (315 SSU at 100°F) for best machine performance.

The lube oil cooler is an integral part of the system to reject heat that is generated across the bearings. It can be of a fan air cooled type or shell and tube design, water cooled. If the cooling water is scale forming, duplicate coolers (one stand-by) are recommended.

Lube oil filtration is extremely important due to close tolerances between bearing surfaces.

**FIG. 13-69**
**Pressure-Temperature Diagram for Expander Process**

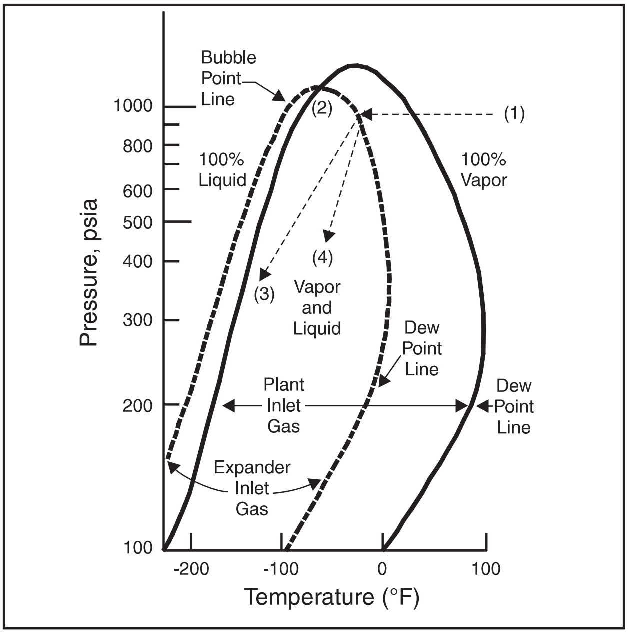
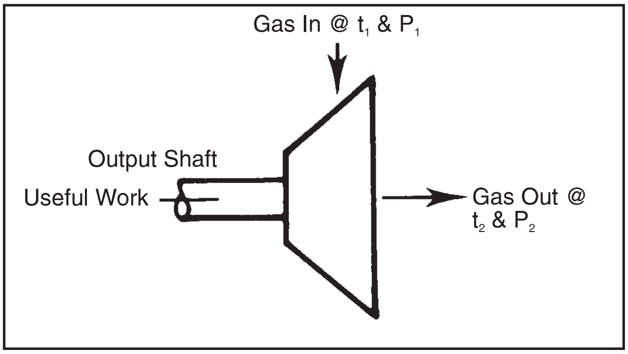



**FIG. 13-70**
**Simple Expander**




The lube-oil reservoir serves as a surge tank to enhance pump suction as well as to serve as a degassing drum permitting process seal gas to be released from the oil. If necessary, the reservoir should be equipped with a heater to bring the oil up to temperature for a "cold" start.

**Seal Gas System** — The seal gas system prevents loss of process gas and assures protection against entry of lube oil into process gas areas. To accomplish this, a stream of "seal gas" is injected into each labyrinth shaft seal at a pressure higher than that of the process gas. The leaking seal gas is collected in the oil reservoir, then returned through a mist eliminator to the fuel gas system, or put back into the compressor suction end.

The system for seal gas injection consists of a liquid collector, electric heater (if required), twin filters, and differential pressure regulators.

If recompression is necessary for the gas processing plant, sales gas is ideal for use as seal gas. If no recompression is provided, a stream can be taken from the expander inlet separator, warmed and used as seal gas. A minimum seal gas temperature (about 70°F) is required to prevent oil thickening.

Seal gas filtration is essential because of close clearances provided between the shaft and seals.

**FIG. 13-71**
**Expander Example Calculation**

Flow: 60 MMscfd $\quad T_1 = -60°F$, $P_1 = 900$ psia, $P_2 = 300$ psia

Composition: 100% $C_1$

$$60 \; \frac{\text{MMscfd//1d/1 lbmol}}{\text{24 hr/379.5 scf}} = 6588 \; \frac{\text{lbmol/16 lb}}{\text{hr/lbmol}} = 105408 \; \frac{\text{lb}}{\text{hr}}$$

Using Fig. 24-17 for Enthalpy & Entropy values.

At Inlet conditions

$$h_1 = 295 \; \frac{\text{BTU}}{\text{lb}}, \qquad s_1 = 1.0 \; \frac{\text{BTU}}{\text{lb°F}}$$

At $P_2 = 300$ psia and assuming 100% efficiency (ideal)

$$s_2 = 1.0 \; \frac{\text{BTU}}{\text{lb°F}} \qquad T_{2\ \text{ideal}} \cong -160°F$$

$$h_{2\ \text{ideal}} \cong 260 \; \frac{\text{BTU}}{\text{lb}}$$

$$\Delta h_{\text{ideal}} = (295 - 260) \frac{\text{BTU}}{\text{lb}} = 35 \; \frac{\text{BTU}}{\text{lb}}$$

Assume 80% expander efficiency:

$$\Delta h_{\text{actual}} = (0.80) \cdot \left(35 \; \frac{\text{BTU}}{\text{lb}}\right) = 28 \; \frac{\text{BTU}}{\text{lb}}$$

$$T_{2\ \text{actual}} \cong -157°F$$

$$\text{Work produced} = \left(\frac{28 \;\text{BTU}}{\text{lb}}\right) \cdot \left(105408 \; \frac{\text{lb}}{\text{hr}}\right)$$

$$= 2951424 \; \frac{\text{BTU}}{\text{hr}}$$

$$\text{Horsepower} = \frac{2951424 \; \frac{\text{BTU}}{\text{hr}}}{2545 \; \frac{\text{BTU}}{\text{HP}}} = 1160 \text{ HP}$$

Seal gas flow requirements are determined by the expander manufacturers as a part of their performance rating.

## Control Systems

**Process** — Control of the process streams begins with proper dehydration and filtering. Generally a final protective screen upstream of the expander is designed into the piping system to form a protective barrier against carbon dioxide or water freezing.

As a further protection against water freezing, methanol injection connections are incorporated into the system upstream of the expander.

**Machine** — The expander speed is established by the manufacturer, given the process conditions. The expander manufacturer determines the wheel diameter and specific speed for maximum efficiency.

As plant operating conditions change, the expander speed may change. Fig. 13-73 shows the change in efficiency as a function of change in design flow rate.

Gas entering the expander is directed by adjustable nozzles into the impeller. About one-half of the pressure drop across the expander takes place in the nozzles, imparting kinetic energy to the gas which is converted to shaft horsepower by the expander wheel. Pressure reductions are normally limited to 3-4 ratios. Greater ratios reduce expander efficiency to the extent that 2-stage expansion may be advisable.

The adjustable inlet nozzles function as pressure control valves. A pneumatic operator takes a split range signal (3 to 9 psi) to stroke the nozzles. On increasing flow beyond the full open nozzle position, a 9 to 15 psi signal from a pressure controller opens a bypass control valve. This valve is called the J-T (Joule-Thomson) valve.

Thrust bearing force imbalance is caused by difference in pressures between the expander discharge and compressor suction. With a differential of the order of 20 psi, the thrust loads are usually within the capabilities of the thrust bearings. At higher pressure differentials, it is essential that steps be taken to control the thrust loads against each other, thereby the net thrust load will not exceed the thrust bearing capacity.

This is done by providing a force-measuring load-meter on each thrust bearing, Fig. 13-74, and a thrust control valve which controls the thrust by control of pressure behind the thrust balancing drums or behind one of the seals. These two load-meters indicate thrust bearing oil film pressure (proportional to bearing load) and the third shows the pressure behind the balancing drum as controlled by the valve in its vent as a means of adjusting the thrust load.

Vibration comes from an unbalanced force on one of the rotating components, or it could come from an outside source such as pipe vibration or gas pulsation.

Most expanders are supplied with monitoring and shutdown devices for shaft vibration. These devices are set to shut down the expander before damage occurs.

**Lube Oil** — The lube oil must be filtered. Most systems use a primary and secondary filtering system. Controls are provided to ensure oil flow to bearings at proper pressure and temperature. Two (2) lube oil pumps are furnished, the second pump serving as a standby. The standby oil pump is controlled automatically to cut in to provide oil pressure upon failure of the main pump or reduction in pressure for other reasons.

Generally an oil flow bypass valve is included to permit excess flow to bypass the expander bearings and return to the reservoir.

For temperature control, the oil must be cooled to prohibit heat buildup which occurs through the bearings. Also, a temperature control bypass is included in the circuit for an extra measure of control to keep the oil from getting too cool.

**Seal Gas** — Use a suitable gas stream with filtering and pressure control to maintain proper gas pressure at the shaft seals.

If the seal gas is delivered from a cold supply point (expander inlet separator) then a means of heating the gas is necessary.

The seal gas should be introduced before the lube oil system is started because there might be a pressure upset which would put enough oil into the process to cause a problem.

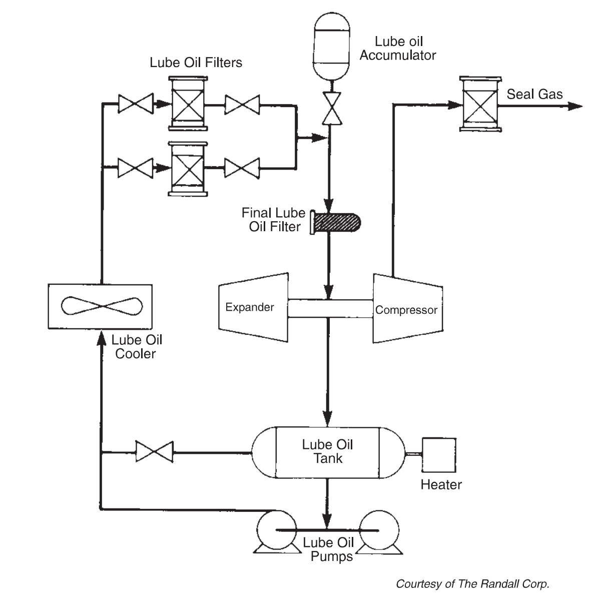
Each of the main rotating components (radial bearings, thrust bearings, and shaft seals) can be damaged or eroded by improper oil filtration, lack of oil flow, improper gas dehydration, and improper seal gas filtration.

**Shutdown** — A number of conditions during the operation of expanders justify prompt shutdown to avoid serious damage.

Some of these conditions are:

- High Vibration
- Low Lube Oil Flow
- High Inlet Separator Level
- High Inlet Screen Pressure Drops
- High Thrust
- High Lube Oil Temperature

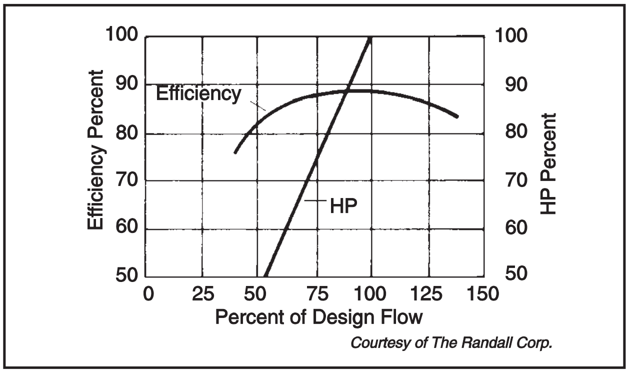
**FIG. 13-72**
**Lube Oil Schematic**




*Courtesy of The Randall Corp.*

**FIG. 13-73**

**Example Change in Efficiency with Flow Rate**




- Low Lube Oil Pressure
- High Speed

Two primary actions of a shutdown signal are to block gas flow to the expander and the compressor. This is accomplished by actuating quick acting shutdown valves at the expander inlet and outlet and the compressor inlet. Simultaneously, a pressurized bladder supplies oil to the bearings during the expander coast down. The expander bypass valve (J-T) opens automatically and is positioned by the split-range pressure controller to keep the plant on-line in the J-T mode.

**Field Performance** — Field measurements can be made to check efficiencies and horsepower of the expander. The process of calculations is just the reverse of selecting a machine performance.

Knowing the gas composition, mass flow (lbs/hr), inlet and outlet conditions (pressure, temperature) for the expander, the actual difference in enthalpy can be determined for each unit.>

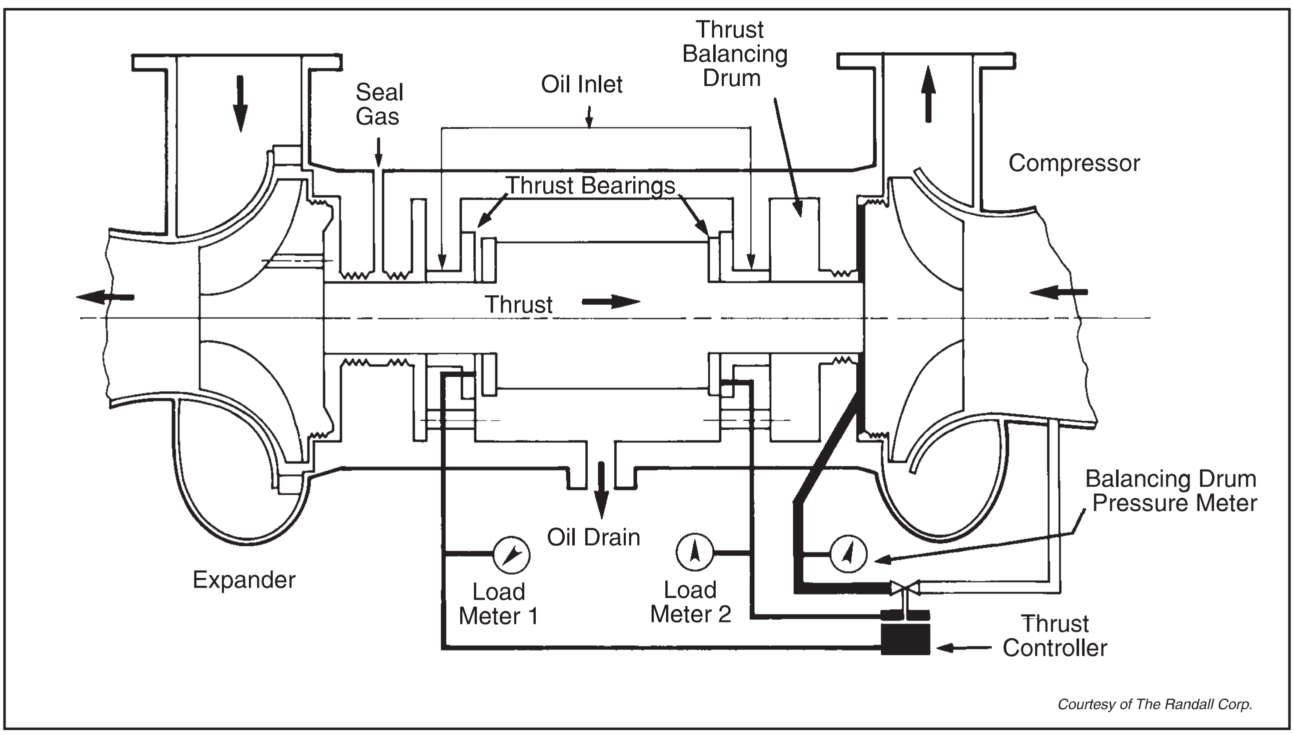
Thus: $\Delta h_{actual} = h_{t_2P_2} - h_{t_1P_1}$

$$\eta = \frac{\Delta h\ \text{actual}}{\Delta h\ \text{ideal}}$$

$$EP_{actual} = \frac{\Delta h\ \text{actual} \cdot \text{lbs/hr}}{2{,}545}$$

**FIG. 13-74**

**Typical Expander/Compressor Cross-Section with Thrust Balancing Schematic**



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# NATURAL GAS ENGINES

## 2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Engine Model | Bore (mm) | Stroke (mm) | Displacement Per Cylinder (L/cyl) | Number Of Cylinders & Configuration L: In-Line V: Vee-Type H: Horizontal O: Opposed | Output Per Cylinder Range (kW) | | Rated Speed Range (rpm) | | Maximum Brake Mean Effective Pressure (bar) | Output Range hp | | Output Range kW | | Rating System & Standard No. ISO - SAE - DIN - Other |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | min | max | min | max | | min | max | min | max | |
| ARROW ENGINE CO. | * | K-6 | 102 | 116 | 0.9 | 1L | | 4.5 | 400 | 800 | 7.1 | | 6 | | 5 | EPA Certified |
| | | C Series | 127 to 190.5 | 159 to 216 | 2 to 10.8 | 1L, 2L | | 8.7 to 49 | 300 to 450 | 600 to 800 | 5.6 to 6 | | 9 to 65.7 | | 7 to 49 | EPA Certified |
| | | L-795 | 190.5 | 228 | 6.5 | 2L | | 24.3 | 300 | 600 | 3.7 | | 65 | | 49 | |
| | | A Series | 98 to 150 | 118 to 150 | 0.9 to 2.65 | 3L, 6L, 12L | | 6 to 35.8 | 900 to 1200 | 1200 to 2200 | 5.8 to 9 | | 22 to 215 | | 16 to 160 | |
| CATERPILLAR INC. | * | GCM34 | 340 | 420 | 38.13 | 16V, 12V | 381.25 | 381.25 | 750 | 750 | 16 (232) | 6135 | 8180 | 4575 | 6100 | |
| | | G Series | 121 to 300 | 152 to 300 | 1.75 to 21.2 | 8 to 8L; 8 to 20V | 17.8 to 233 | 26.2 to 249 | 1000 | 1000 to 1800 | 10 to 13.3 (142 to 193) | 95 to 1875 | 211 to 5350 | 71 to 1398 | 157 to 3990 | |
| | | CG137 | 137 | 152 | 2.250 | 8V, 12V | 37.25 | 37.25 | 1800 | 1800 | 11 (160.1) | 400 | 600 | 298 | 447 | |
| COOPER MACHINERY SERVICES | * | Ajax DPC Series | 330 to 381 | 406 | 36.15 to 46.33 | 2L, 3L, 4L | 78 to 110 | 110 to 158 | 265 | 440 | 4.3 to 4.8 | 104 to 592 | 148 to 846 | 82 to 441 | 110 to 630 | ANSI PTC 17-1974 |
| | | Ajax E-565 | 216 | 254 | 9.29 | L | 21 | 30 | 315 | 525 | 3.67 | 28 | 40 | 21 | 30 | ANSI PTC 17-1974 |
| | | Superior Series | 254 | 267 | 13.52 | 12V, 16V | 82, 83 | 124 | 600 | 900 | 12.26 | 1333 to 1766 | 2000 to 2650 | 994 to 1317 | 1491 to 1976 | ANSI PTC 17-1974, DEMA |
| | | Cooper-Bessemer Series | 457 | 508 | 26.55 | 6V to 16 V | 299 | 374 | 270 | 330 | 8.13 | 2400 to 6400 | 3000 to 8000 | 1789 to 4772 | 2237 to 5965 | ANSI PTC 17-1974, DEMA |
| CUMMINS INC. | * | KTA19GCE | 159 | 159 | 3.17 | 6L | 31.5 | 52.2 | 1200 | 1800 | N/A | 254 | 420 | 189 | 313 | SAE J1995 |
| | | G855 | 140 | 152 | 2.33 | 6L | 13.0 | 23.3 | 1000 | 1800 | N/A | 104 | 188 | 78 | 140 | SAE J1995 |
| | | QSL9G | 114 | 145 | 1.48 | 6L | 11.7 | 21.7 | 1200 | 1800 | N/A | 90 | 175 | 67 | 130 | SAE J1995 |
| | | G8.3 | 114 | 135 | 1.38 | 6L | 6.8 | 16.8 | 1000 | 2200 | N/A | 55 | 135 | 41 | 101 | SAE J1995 |
| | | G5.9 | 102 | 120 | 0.98 | 6L | 3.3 | 12.3 | 1000 | 2200 | N/A | 27 | 99 | 20 | 74 | SAE J1995 |
| | | GTA Series | 102 to 159 | 120 to 159 | 0.98 to 3.17 | 6L, 12V | 96 to 35.4 | 14.4 to 47.2 | 1000 to 1350 | 1800 | N/A | 48 to 570 | 116 to 760 | 36 to 425 | 87 to 567 | SAE J1995 |
| INNIO - WAUKESHA GAS ENGINES | * | 275GL+/16V | 275 | 300 | 17.82 | 16V | 175 | 233 | 750 | 1000 | 15.7 | | | 2796 | 3729 | |
| | | 275GL+/12V | 275 | 300 | 17.82 | 12V | 175 | 233 | 750 | 1000 | 15.7 | | | 2097 | 2796 | |
| | | VHP/P9394 S5 | 238 | 216 | 7.92 to 9.58 | 6L, 12V, 16V | 38.2 to 87.4 | 85.75 to 118 | 800 to 1000 | 1200 | 9.6 to 12.3 | | | 368 to 1399 | 552 to 1865 | |
| | | VGF Series | 152 | 165 | 3 | 6L, 8L, 12V, 16V | 19.8 to 38.3 | 55 | 1100 to 1400 | 1800 | 12.2 | | | 119 to 530 | 330 to 880 | |
| WÄRTSILÄ | * | W Series | 340 to 500 | 400 to 580 | 32.17 to 114 | 9L, 12V to 20V | 480 to 1045 | 460 to 1070 | 500, 720 | 514, 750 | 20 to 23 | 5431 to 22,931 | 12,337 to 25,828 | 4050 to 18,810 | 9200 to 19,260 | ISO 3046; Gas Engine or Dual-Fuel |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# MECHANICAL DRIVE GAS TURBINES

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Number | Continuous Output At ISO Conditions | | Heat Rate | | Pressure Ratio | Maximum Output Shaft Speed (rpm) |
|---|---|---|---|---|---|---|---|---|
| | | | bhp | kW | Btu/hph | kJ/kWh | | |
| BAKER HUGHES | Inside Front Cover, 65 | NovaLT12 | 17,400 | 12,975 | 6914 | 9783 | 19 | 8900 |
| | | NovaLT16-2 | 23,467 | 17,499 | 6784.5 | 9600 | 19 | 7800 |
| | | PGT25 | 31,195 | 23,262 | 6793 | 9612 | 17.9 | 6500 |
| | | PGT25+ | 41,750 | 31,133 | 6280 | 8886 | 21.5 | 6100 |
| | | PGT25+G4 | 45,160 | 33,676 | 6280 | 8886 | 23.2 | 6100 |
| | | PGT25+G5 | 51,390 | 38,322 | 6187 | 8755 | 25.1 | 6100 |
| | | LM2500 | 31,235 | 23,292 | 6797 | 9618 | 18.7 | 3600 |
| | | LM2500+ | 41,775 | 31,152 | 6523 | 9230 | 23.7 | 3600 |
| | | LM2500+G4 | 45,077 | 33,614 | 6450 | 9127 | 24.3 | 3600 |
| | | LM2500+G5 | 50,420 | 37,598 | 6070 | 8589 | 25 | 3600 |
| | | LM6000PC | 59,662 | 44,490 | 6028 | 8530 | 30.1 | 3600 |
| | | LM6000PF | 59,655 | 44,485 | 6018 | 8515 | 30 | 3600 |
| | | LM6000PG | 70,787 | 52,786 | 6042.4 | 8550 | 33.8 | 3930 |
| | | LM6000PF+ | 72,238 | 53,868 | 6052 | 8564 | 33.7 | 3930 |
| | | LM9000 | 98,565 | 73,500 | 5783 | 8183 | 33 | 3429 |
| | | LMS100-PB+ | 147,512 | 110,000 | 5783 | 8183 | 44 | 3429 |
| | | LMS100-PB | 140,475 | 104,752 | 5719 | 8092 | 42 | 3429 |
| | | Frame 5-2C | 38,000 | 28,337 | 8701 | 12,312 | 9,1 | 4670 |
| | | Frame 5-2D | 45,570 | 34,000 | 8410 | 11,900 | 11 | 4670 |
| | | Frame 5-2E | 45,300 | 33,800 | 6884 | 9741 | 17,4 | 5714 |
| | | Frame 6B | 59,004 | 44,000 | 7591 | 10,741 | 12.7 | 5163 |
| | | Frame 7EA | 122,033 | 91,000 | 7501 | 10,614 | 13 | 3600 |
| | | Frame 9E | 177,012 | 132,000 | 7754 | 10,972 | 13.1 | 3000 |

# MECHANICAL DRIVE GAS TURBINES

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Number | Continuous Output At ISO Conditions | | Heat Rate | | Pressure Ratio | Maximum Output Shaft Speed (rpm) |
|---|---|---|---|---|---|---|---|---|
| | | | bhp | kW | Btu/hph | kJ/kWh | | |
| GTR & PC "ZORYA"-"MASHPROEKT" | * | UGT3000 (DE76) | 4500 | 3360 | 8210 | 11,615 | 13.5 | 9700 |
| | | UGT6000 (DT71) | 8720 | 6500 | 8080 | 11,430 | 14.0 | 8200 |
| | | UGT8000 (DT70) | 11,130 | 8300 | 7665 | 10,845 | 16.0 | 8200 |
| | | UGT10000 (DN70) | 14,080 | 10500 | 7070 | 10,000 | 18.5 | 4800/6500 |
| | | UGT15000 (DG90) | 22,400 | 16,700 | 7270 | 10,285 | 19.5 | 5200 |
| | | UGT16000 (DJ59L2) | 22,400 | 16,700 | 7955 | 11,250 | 13.0 | 5200 |
| | | UGT25000 (DU80) | 34,870 | 26,000 | 7070 | 10,000 | 21.5 | 5000 |
| | | UGT25000 (DN80) | 35,800 | 26,700 | 6975 | 9865 | 21.5 | 3700 |
| | | UGT32000 (DU32) | 43,990 | 32,800 | 6525 | 9230 | 22.8 | 5500 |
| MAN ENERGY SOLUTIONS | 118, 119 | MGT6000 | 9250 - 11,130 | 6900 - 8300 | 7270 - 7480 | 10,290 - 10,590 | 15 | 12,600 |
| | | THM 1304-10N | 14,080 | 10,500 | 8370 | 11,840 | 10 | 9450 |
| | | THM 1304-12N | 16,090 | 12,000 | 8210 | 11,610 | 11 | 9450 |
| MITSUBISHI | * | MFT-8 | 35,910 | 26,780 | 6582 | 9313 | 21 | 5000 |
| | | ASE-40 | 3038 | 2265 | 10,259 | 14,518 | 8.4 | 15,400 |
| SIEMENS ENERGY | * | SGT Series | 7644 to 54,994 | 5700 to 41,100 | 6121 to 7656 | 8661 to 10,832 | 13.8 to 24.3 | 6405 to 13,650 |
| | | SGT-A35 Series | 37,464 to 51,092 | 27,940 to 38,100 | 6289 to 6819 | 8893 to 9648 | 20.6 to 25.2 | 3600 to 5093 |
| SOLAR TURBINES INCORPORATED | Prime Movers Tab | Titan 250 | 31,900 | 23,790 | 6360 | 9000 | 24.1 | 7000 |
| | | Titan 130 | 23,470 | 17,500 | 6800 | 9620 | 16.1 | 8855 |
| | | Mars 100 | 15,900 | 11,860 | 7395 | 10,465 | 17.1 | 9500 |
| | | Mars 90 | 13,220 | 9860 | 7655 | 10,830 | 16.3 | 9500 |
| | | Taurus 70 | 11,110 | 8290 | 7190 | 10,170 | 16.5 | 11,605 |
| | | Taurus 60 | 7700 | 5740 | 7950 | 11,250 | 12.2 | 14,300 |
| | | Centaur 50 | 6150 | 4590 | 8485 | 12,000 | 10.3 | 16,500 |
| | | Centaur 40 | 4700 | 3500 | 9100 | 12,870 | 10.3 | 15,500 |
| | | Saturn 20 | 1590 | 1185 | 10,360 | 14,655 | 6.7 | 22,300 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# MECHANICAL DRIVE STEAM TURBINES

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Type | Output Range kW min | Output Range kW max | Output Range hp min | Output Range hp max | Cycle Type: Back Pressure | Cycle Type: Condensing | Frame Configuration: E = Extraction, I = Injections | Frame Configuration: SF = Single Flow, DF = Double Flow | Number Frame Sizes | Maximum Inlet Steam Pressure bar | Maximum Inlet Steam Pressure PSI | Maximum Inlet Temperature °C | Maximum Inlet Temperature °F | Maximum Steam Flow kg/s | Maximum Steam Flow lb/s | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| BAKER HUGHES | Inside Front Cover, 65 | SNC | 2000 | 100,000 | 2,680 | 134,100 | X | | I | SF | | 140 | 2030 | 565 | 1050 | 180 | 400 | 2000 | 16,000 |
| | | SANC | 2000 | 100,000 | 2,680 | 134,100 | X | | E/I | SF | | 140 | 2030 | 565 | 1050 | 180 | 400 | 2000 | 16,000 |
| | | SC | 2000 | 150,000 | 2,680 | 201,153 | | X | I | SF | | 140 | 2030 | 565 | 1050 | 220 | 485 | 2000 | 16,000 |
| | | SAC | 2000 | 150,000 | 2,680 | 201,153 | | X | E/I | SF | | 140 | 2030 | 565 | 1050 | 220 | 485 | 2000 | 16,000 |
| | | SDFC | 5000 | 80,000 | 6,705 | 107,282 | | X | I | DF | | 30 | 435 | 300 | 570 | 100 | 660 | 2000 | 16,000 |
| | | SGNC | 2000 | 35,000 | 2,680 | 46,900 | X | | I | SF | | 30 | 435 | 300 | 570 | 180 | 400 | 2000 | 16,000 |
| | | SGC | 2000 | 60,000 | 2,680 | 80,500 | | X | I | SF | | 30 | 435 | 300 | 570 | 180 | 400 | 2000 | 16000 |
| | | SGDFC | 5000 | 60,000 | 6,705 | 80,500 | | X | I | DF | | 30 | 435 | 300 | 570 | 100 | 660 | 2000 | 16,000 |
| | | C | 500 | 6000 | 670 | 8050 | | X | E/I | SF | | 90 | 1305 | 500 | 932 | | | 3000 | 15,000 |
| | | P | 500 | 6000 | 670 | 8050 | X | | E/I | SF | | 90 | 1305 | 500 | 932 | | | 3000 | 15,000 |
| | | BFPT | 5000 | 30,000 | 6705 | 40,230 | | X | | SF | | 280 | 4060 | 575 | 1070 | | | 3000 | 6,000 |
| | | MC | 2000 | 45,000 | 2000 | 60,300 | | X | E/I | SF | | 140 | 2030 | 540 | 1004 | | | 3000 | 15,000 |
| | | MP | 2000 | 40,000 | 1350 | 53,600 | X | | E/I | SF | | 140 | 2030 | 540 | 1004 | | | 3000 | 15,000 |
| ELLIOTT GROUP | 61, 137, Inside Back Cover | YR | 1 | 2600 | 1.00 | 3500 | X | X | | SF | 5 | 103 | 1500 | 538 | 1000 | 15 | 34 | 500 | 7100 |
| | | K, R, Q, N | 745 | 130,000 | 1000 | 175,000 | X | X | E/I | SF/DF | 4 | 151 | 2200 | 565 | 1050 | 303 | 670 | 1500 | 16,000 |
| | | MYR | | 10,400 | | 14,000 | X | X | E | SF | 5 | 62 | 900 | 482 | 950 | 18 | 40 | 500 | 8500 |
| | | E, B | 336 | 8950 | 450 | 12000 | X | X | | SF/DF | 2 | 65 | 950 | 510 | 950 | 29 | 63 | 2000 | 14,500 |
| HOWDEN | 104, 105 | KK&K BASE AF | 30 | 750 | 40 | 1000 | yes | no | no | SF | 2 | 101 | 1485 | 500 | 930 | 3 | 7 | | 5000 |
| | | KK&K BASE BF | 2 | 350 | 3 | 475 | yes | no | no | SF | 3 | 101 | 1485 | 500 | 930 | 4 | 9 | | 4500 |
| | | KK&K MONO | 300 | 6000 | 400 | 8000 | yes | yes | no | SF | 10 | 131 | 1925 | 530 | 985 | 40 | 88 | | 25,000 |
| | | KK&K TWIN | 1000 | 12,000 | 1350 | 16,100 | yes | yes | E, I | SF, DF | combinations of KK&K MONO frames | 131 | 1925 | 530 | 985 | 45 | 99 | 1500 | 3000 |
| | | KK&K MONO CBA | 300 | 4500 | 400 | 6000 | yes | no | no | SF | 1 | 53 | 769 | 440 | 824 | 40 | 88 | | 9000 |

# MECHANICAL DRIVE STEAM TURBINES

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Type | Output Range kW min | Output Range kW max | Output Range hp min | Output Range hp max | Cycle Type: Back Pressure | Cycle Type: Condensing | Frame Configuration: E = Extraction I = Injections | Frame Configuration: SF = Single Flow DF = Double Flow | Number Frame Sizes | Maximum Inlet Steam Pressure bar | Maximum Inlet Steam Pressure PSI | Maximum Inlet Temperature °C | Maximum Inlet Temperature °F | Maximum Steam Flow kg/s | Maximum Steam Flow lb/s | Speed Range (rpm) min | Speed Range (rpm) max |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| MAN ENERGY SOLUTIONS | 118, 119 | MST 010 | 500 | 1500 | 670 | 2010 | X | X | E, I | SF | | 45 | 653 | 450 | 842 | | | 8000 | 16,500 |
| | | MST 020 | 1000 | 5000 | 1340 | 6700 | X | X | E, I | SF | | 130 | 1885 | 530 | 986 | | | 13,000 | 13,000 |
| | | MST 040 | 3000 | 15,000 | 4020 | 20,100 | X | X | E, I | SF | | 140 | 2030 | 540 | 1004 | | | 4164 | 14,206 |
| | | MST 050 | 5000 | 30,000 | 6700 | 40,200 | X | X | E, I | SF | | 140 | 2030 | 540 | 1004 | | | 3559 | 11,365 |
| | | MST 060 | 15,000 | 55,000 | 20,100 | 73,700 | X | X | E, I | SF | | 140 | 2030 | 540 | 1004 | | | 2546 | 10,166 |
| | | MST 080 | 25,000 | 75,000 | 33,500 | 100,500 | X | X | E, I | SF | | 140 | 2030 | 540 | 1004 | | | 2038 | 7274 |
| | | MST 100 | 40,000 | 140,000 | 53,600 | 187,600 | X | X | E, I | SF | | 140 | 2030 | 540 | 1004 | | | 1536 | 5819 |
| | | MST 120 | 70,000 | 180,000 | 93,800 | 241,200 | X | X | E, I | SF | | 140 | 2030 | 540 | 1004 | | | 1536 | 4655 |
| MITSUBISHI | * | EBL or EBH | 2000 | 80,000 | 2700 | 107,300 | X | | E | SF | 10 | 142 | 2060 | 560 | 1040 | 167 | 368 | 2600 | 25,000 |
| | | MXL or MXH | 2000 | 80,000 | 2700 | 107,300 | | X | I | SF | 10 | 142 | 2060 | 560 | 1040 | 83 | 183 | 2600 | 19,000 |
| | | EL or EH | 2000 | 120,000 | 2700 | 160,000 | | X | E | SF | 10 | 142 | 2060 | 560 | 1040 | 220 | 485 | 2600 | 19,000 |
| | | BL or BH | 2000 | 80,000 | 2700 | 107,300 | X | | | SF | 10 | 142 | 2060 | 560 | 1040 | 167 | 368 | 2600 | 25,000 |
| MITSUBISHI HEAVY INDUSTRIES COMPRESSOR INTERNATIONAL | * | | | 150,000 | | 201,000 | X | X | E,I | SF, DF | | 170 | 2465 | 565 | 1050 | 14 | 30 | | 20,000 |
| SHIN NIPPON | * | C | 50,000 | 50,000 | 67,000 | 67,000 | | X | E/I | SF/DF | 7 | 130 | 1885 | 540 | 1004 | | | | 14,000 |
| | | B | 50,000 | 50,000 | 67,000 | 67,000 | X | | E/I | SF | 6 | 130 | 1885 | 540 | 1004 | | | | 16,000 |
| SIEMENS ENERGY | * | SST Series | | 20,000 to 200,000 | | | X | X | E, I | SF/DF | 2, 4, Modular | 30 to 165 | 430 to 2395 | 400 to 565 | 750 to 1050 | | | | 8000 to 18,000 |
| | | D-R Series | | 750 to 25,000 | | 1000 to 33,500 | X | X | E, I | SF | 25 | 63 to 125 | 915 to 1508 | 482 to 550 | 890 to 1022 | | | | 6000 to 15,000 |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# ELECTRIC MOTORS

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Frame Size | | Output Range (kW) | | Poles | Voltage Range (kV) | | Motor Type | Frequency Hz | VFD Operation | Speed Range For VFD Operation | | Motor Efficiency (%) At Rated Operating Point | Normal Power Factor (cos Phi) At Rated Operating Point | Explosion Proof Available | A/A=Air/Air A/W=Air/Water R=Rib-Coated O=Open |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | min | max | min | max | (2, 4, 6) | min | max | (IM, SM) | (50, 60) | (Y/N) | min | max | | | (Y/N) | |
| BALDOR | * | Varies | 143 to 183 | 184 to 286 | 1 | 4 to 22 | 4 | 200 | 230 to 575 | | 60 | N | | 89 | 85 | | Y | A/A |
| CATERPILLAR INC. | * | CN Series | | 450 to 560 | | 1119 to 3729 | 6, 8 | | 4/4.16 | IM | 60 | Y | 600 | 900, 1200 | | | | WPII |
| | | CN2785 | | 630 | | 2700 | 8 | | 10 | IM | 50 | Y | | 750 | | | | IC611 |
| ELIN MOTOREN GMBH | * | HKG (IC411) | 355 | 560 | 200 | 2500 | 2 to 18 | | 11 | IM | 50, 60 | Y | | | 98 | | Y | R |
| | | HKR (IC511) | 710 | 1250 | 400 | 8000 | 2 to 24 | | 15 | IM | 50, 60 | Y | | | 98 | | Y | A/A |
| | | HKL (IC611) | 500 | 1250 | 400 | 16,000 | 2 to 24 | | 15 | IM | 50, 60 | Y | | | 98 | | Y | A/A |
| | | HKH(IC7A0W7) | 355 | 560 | 130 | 2500 | 2 to 8 | | 11 | IM | 50, 60 | Y | | | 98 | | Y | W |
| | | MKH (IC7A0W7) | 200 | 630 | 37 | 3000 | 2 to 10 | 0 | 1 | IM | 50, 60 | Y | 0 | 10,000 | | | Y | W |
| | | HKM (IC81W ) | 450 | 1250 | 300 | 35,000 | 2 to 24 | | 15 | IM | 50, 60 | Y | | | 98 | | Y | A/W |
| | | HRM Slip Ring | 500 | 1250 | 500 | 25,000 | 2 to 8 | | 15 | IM | 50, 60 | | | | 98 | | Y | A/W |
| | | Special Motors | 355 | 1400 | | 35,000 | 2 to 30 | | 15 | IM | 50, 60 | Y | | | 98 | | Y | |
| GE POWER CONVERSION | * | FL | 280 | 450 | 200 | 800 | 2 to 8 | 3 | 11 | IM | 50, 60 | Y | 1 | 3600 | 97 | 0.92 | N | R |
| | | F3 | 355 | 560 | 500 | 2500 | 2 to 12 | 3 | 11 | IM | 50, 60 | Y | 1 | 5000 | 97.5 | 0.92 | N | A/A A/W O |
| | | AKG | 355 | 560 | 200 | 1800 | 2 to 12 | 3 | 11 | IM | 50, 60 | Y | 1 | 3600 | 97.5 | 0.9 | Y | R |
| | | N Series | 450, 630 | 1250, 1600 | 500 to 4000 | 8000 to 40,000 | 2 to 30 | 3 | 14 | IM | 50, 60 | Y | 1 | 3600, 5000 | 97.8 | 0.93 | N | A/A A/W O |
| | | MS | 800 | 1600 | 7500 | 50,000 | 4 to 30 | 3 | 14 | SM | 50, 60 | Y | 1 | 1800 | 98 | 1 | N | A/A A/W O |
| | | TM | 1000 | 1800 | 15,000 | 100,000 | 2 | 6 | 11 | SM | N/A | Y | 500 | 6500 | 98.5 | 1 | N | A/W |
| | | MGV | 450 | 800 | 1500 | 20,000 | 2 | 3 | 9 | IM | N/A | Y | 3000 | 15,000 | 97.5 | 0.85 | N | INTEGRATED |
| HITACHI INDUSTRIAL PRODUCTS, LTD. | * | HyMD | 400 | 710 | 265 | 8500 | 2 to 12 | 3 | 14 | IM | 50, 60 | Y | 5 | 60 | 95 | 90 | Y | A/A, A/W, O |
| | | JF2000 | 250 | 500 | 55 | 2250 | 2 to 12 | 3 | 6 | IM | 50, 60 | Y | 5 | 60 | 95 | 90 | Y | R |
| TECO WESTINGHOUSE | * | PDH | 215T | 405T | 7.5 | 74.9 | 4 | 230 | 600 | IM | 60 | Y | 3 | 120 | 92.4 | 88 | Y | R |
| TMEIC | * | High-Speed Custom | 400 | Special | 500 to 10,000 | 70,000 to 100,000 | 2 | 2 | 14 | IM | 50 and up | Y | 3000 | 12000 | 98 | 1 | Y | TEAAC, TEWAC |
| | | High-Power Custom | 630 | Special | 10,000 | 25,000 to 70,000 | Any | 2 | 14 | IM or SM | 50, 60 | Y | 1 | 3600 | 98 | 1 | Y | TEAAC, TEWAC |
| | | TM 21 Series | 315 | 1200 | 160 | 25,000 | Any | 2 | 14 | IM | 50, 60 | Y | 1 | 3600 | 98 | 1 | Y | TEFC, WPII, TEAAC, TEWAC |
| WEG | * | W22 | 215T | 405T | 7.5 | 74.9 | 4 | 230 | 600 | IM | 60 | Y | 3 | 120 | 92.4 | 83 | Y | R |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

# VARIABLE SPEED DRIVES

2022 BASIC SPECIFICATIONS

| MANUFACTURER | Catalog Page Reference | Model Designation | Output Range (kW) | | Max. Motor Voltage (kV) | Max. Output Frequency (Hz) | Rectifier Type (6, 12, 18, 24, 36 Pulse, AFE) | Drive Type (VSI, CSI, LCI) | Step Number Of Inverter Output (6,12, 24 Pulse System) | Semiconductors | | Motor Type (IM, SM) | Cooling A=Air W=Water |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | min | max | | | | | | Rectifier | Inverter | | |
| GE POWER CONVERSION | * | MV Series | | 3800 to 27,000 | 3.3 to 10 | 300 | 12 Puls to 36 Puls, AFE | VSI | 3 Level | Diode or IGBT | IGBT | IM or SM | A or W |
| | | 2xMV7927 | | 54,000 | 10 | 300 | 2 x 36 Puls | VSI | 5 Level | Diode | IGBT | IM or SM | W |
| | | 3xMV7927 | | 81,000 | 10 | 300 | 3 x 36 Puls | VSI | 7 Level | Diode | IGBT | IM or SM | W |
| | | SD Series | | 4000 to 80,000 | 1.5 to 11 | 100 | 6 or 12 Puls | LCI | 6 or 12 Puls | Thyristor | Thyristor | SM | A or W |
| | | MV Series | | 250 To 5500 | 4.1 to 6.6 | 75 to 90 | 36 Puls or AFE | VSI | 5 level | Diode or IGBT | IGBT | IM or SM | A |
| NIDEC ASI | * | Silcovert TN | 1300 | 21,600 | 3300 | 140 | 12p, 24p, AFE | VSI | 6 | Diode/IGBT | IGBT | IM | A, W |
| | | Silcovert GN | 10,000 | 24,000 | 3300 | 70 | 12p, 24p, AFE | VSI | 6 | Diode/IGCT | IGCT | IM, SM | W |
| | | Silcovert TH | 290 | 42,400 | 2400 - 7200 | 300 | 18p, 24p, 30p, 36p | VSI | 6 | Diode | IGBT | IM, SM | A, W |
| | | Silcovert TH+ | 1400 | 60,200 | 10.000 - 13.800 | 300 | 24p, 30p, 36p | VSI | 6 | Diode | IGBT | IM, SM | A, W |
| | | Silcovert FH | 400 | 2500 | 3.300 - 6.600 | 100 | AFE transformerless | VSI | 6 | IGBT | IGBT | IM | A |
| | | Silcovert S | 1500 | 45,000 | 3.300 - 6.600 | 70 | 6p, 12p, 24p | LCI | 6, 12 | Thyristor | Thyristor | SM | A, W |
| VOITH TURBO | * | Variable speed planetary gear | 1000 | 50,000 | any motor pos-sible | motor is oper-ated direct online | N/A | | N/A | N/A | N/A | IM, SM | A, W |
| | | Geared variable speed coupling | 1000 | 30,000 | any motor pos-sible | motor is oper-ated direct online | N/A | | N/A | N/A | N/A | IM, SM | A, W |
| | | Variable speed coupling | 100 | 10,000 | any motor pos-sible | motor is oper-ated direct online | N/A | | N/A | N/A | N/A | IM, SM | A, W |

* This company is not represented in the 2022 Sourcing Guide with a section describing its products.

The following section covering Prime Movers for Mechanical Drives has been reproduced, by permission, from the **GPSA Engineering Data Book**, 14th edition, published by GPSA. The complete GPSA Engineering Data Book can be ordered by visiting GPSAmidstreamsuppliers.org/.

# Prime movers for mechanical drives

"Prime movers for mechanical drives" is a common term for machines made for transferring mechanical energy to pumps and compressors, including:

- Steam turbines
- Gas turbines
- Electrical motors
- Internal combustion engines

Special considerations for the use of prime movers as drives for generators are not included in this chapter.

## STEAM TURBINE TYPES

Mechanical drive steam turbines are major prime movers for compressor, blower, and pump applications. Steam turbines are available for a wide range of steam conditions, horsepower, and speeds. Typical ranges for each design parameter are:

| | |
|---|---|
| Inlet Pressure, psig | 30 – 2000 |
| Inlet Temperature, °F | saturated – 1000 |
| Exhaust Pressure, psig | saturated – 700 |
| Horsepower | 5 – 100,000 |
| Speed, rpm | 1800 – 14,000 |

Steam turbines used as process drivers are usually required to operate over a range of speeds in contrast to a turbine used to drive an electric generator which runs at nearly constant speed. Significant hardware differences exist between these two applications. Only variable speed process drivers will be covered here.

Mechanical drive steam turbines are categorized as:

- Single-stage or multi-stage
- Condensing or non-condensing exhausts
- Extraction or admission
- Impulse or reaction

### Single Stage/Multi-Stage

In a single-stage turbine, steam is accelerated through one cascade of stationary nozzles and guided into the rotating blades or buckets on the turbine wheel to produce power. A Rateau design has one row of buckets per stage (Fig. 15-2). A Curtis design has two rows of buckets per stage and requires a set of turning vanes between the first and second row of buckets to redirect the steam flow (Fig. 15-3). A multi-stage turbine utilizes either a Curtis or Rateau first stage followed by one or more Rateau stages.

Single-stage turbines are usually limited to about 2500 horsepower although special designs are available for larger units. Below 2500 horsepower the choice between a single and a multi-stage turbine is usually an economic one. For a given shaft horsepower, a single-stage turbine will have a lower capital cost but will require more steam than a multi-stage turbine because of the lower efficiency of the single-stage turbine.

### Condensing/Non-Condensing

The energy available in each pound of steam which flows through the turbine is a function of the overall turbine pressure ratio (inlet pressure/exhaust pressure) and inlet temperature. Condensing turbines are those whose exhaust pressure is below atmospheric. They offer the highest overall turbine pressure ratio for a given set of inlet conditions and therefore require the lowest steam flow to produce a given horsepower. A cooling medium is required to totally condense the steam.

Non-condensing or back-pressure turbines exhaust steam at pressures above atmospheric and are usually applied when the exhaust steam can be utilized elsewhere.

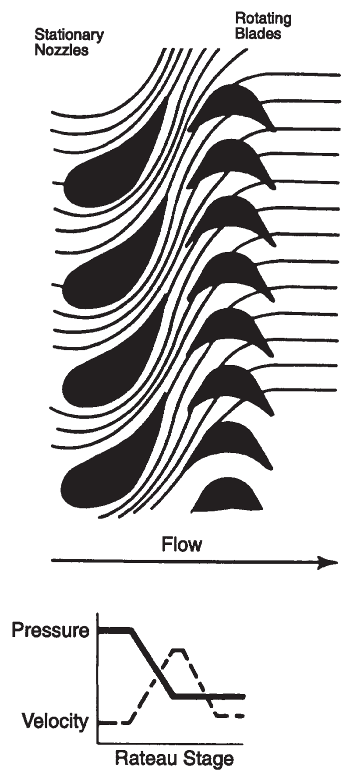
### Extraction/Admission

Some mechanical drive steam turbines are either extraction or admission machines. Steam is extracted from, or admitted to, the turbine at some point between the inlet and exhaust (Fig. 15-4). Admission or extraction units may be either controlled or uncontrolled. An uncontrolled turbine accepts or provides steam based only on the characteristics of the steam sys-

**FIG. 15-1**
**Nomenclature**

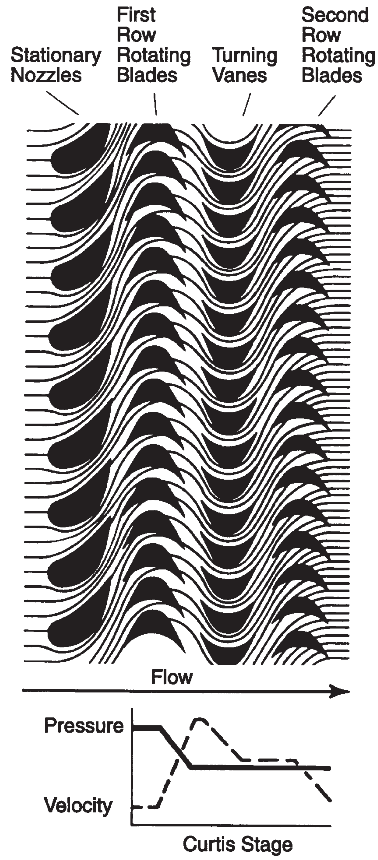
- A = area, sq in.
- ASR = actual steam rate, lb/(hp · hr)
- BMEP = brake mean effective pressure, psi
- D = diameter, in.
- F = steam low, lb/hr
- f = frequency, Hz
- h = specific enthalpy of superheated steam, Btu/lb
- $h_f$ = specific enthalpy of saturated water, Btu/lb
- $h_g$ = specific enthalpy of saturated steam, Btu/lb
- N = number of power strokes per min
- P = number of magnetic poles in motor
- s = specific entropy of superheated steam, Btu/(lb · °F)
- $s_f$ = specific entropy of saturated water, Btu/(lb · °F)
- $s_g$ = specific entropy of saturated steam, Btu/(lb · °F)
- S = piston stroke, ft
- TSR = theoretical steam rate, lb/(hp · hr)
- v = velocity, ft/sec
- ρ = density, lb/cu ft

**FIG. 15-2**
**Rateau Design**




Courtesy of General Electric

**FIG. 15-3**
**Curtis Design**




Courtesy of General Electric

tem to which the extraction or admission line is connected. A controlled turbine will control the flow of extraction or admission steam based on some process measurement such as pressure or flow. In general, if the horsepower associated with the extraction or admission flow is greater than 15% of the total turbine horsepower, a controlled extraction (or admission) turbine is used.

### Impulse/Reaction

Turbines are further categorized by the philosophy employed in the steam path design and are divided into two major design concepts: impulse and reaction. In an impulse turbine the pressure drop for the entire stage takes place across the stationary nozzle. In reaction designs, the pressure drop per stage is divided equally between the stationary nozzles and the rotating blades (Fig. 15-5). For given horsepower, speed and steam conditions, a reaction turbine will, in general, employ approximately three times more stages than an impulse turbine in the same turbine span. Most U.S. mechanical drive steam turbines are of the impulse type.

## STEAM TURBINE COMPONENTS

### Trip and Throttle Valve/Stop (Block) Valve

A trip-and-throttle valve or stop valve, or both, may be positioned between the steam supply and the turbine inlet control valve(s) (Fig. 15-6). During normal operation this valve remains fully open and its primary function is to shut off the steam supply in response to a trip (shutdown) signal. In addition a trip-and-throttle valve can be used to modulate the steam flow during start-up and can be either manually or hydraulically positioned from zero lift to 100% lift. The stop valve can only be positioned either in the closed or fully open positions. In order to minimize the pressure drop through the trip-and-throttle valve, maximum inlet velocities are usually limited to 150 ft/sec. Velocities above this level will usually result in high pressure drops which will reduce turbine efficiency.

### Inlet Control Valves

The primary function of the inlet control valve(s) is regulation of the steam flow to provide the appropriate horsepower and speed. These valves may also close in response to a shutdown signal. Throttling which occurs across the control valve(s) reduces the thermal performance of the turbine. This efficiency loss is a function of the control valve design and overall turbine pressure ratio. For a given amount of throttling, turbines with large pressure ratios suffer smaller efficiency losses than turbines with smaller pressure ratios (Fig. 15-7).

Multi-stage turbines may have a single inlet control valve or several control valves to regulate the inlet steam. Typical multi-valve steam turbines will have from three to eight control valves

**FIG. 15-4**
**Extraction/Admission Flow Turbines**

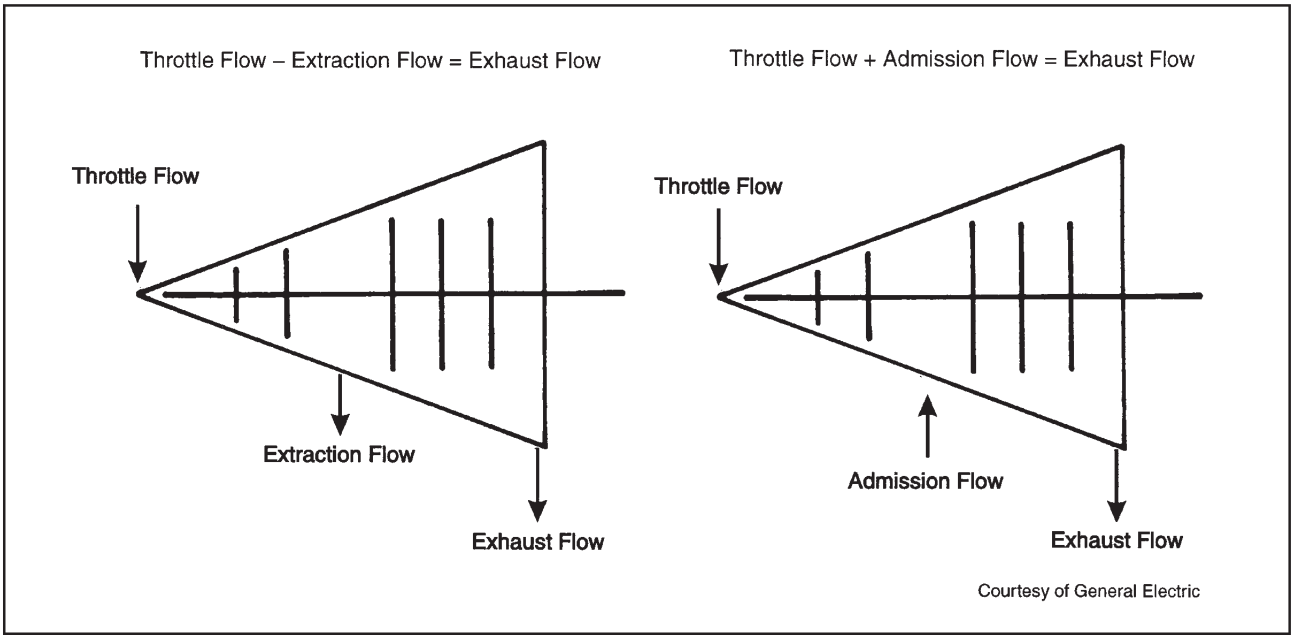



**FIG. 15-5**
**Turbine Types**

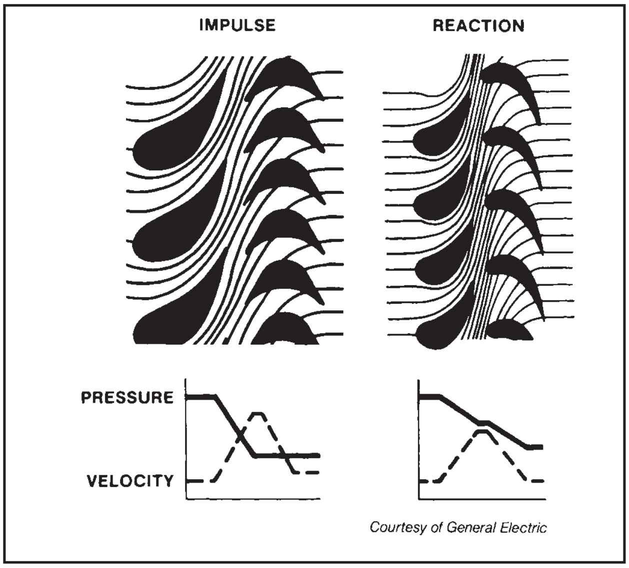
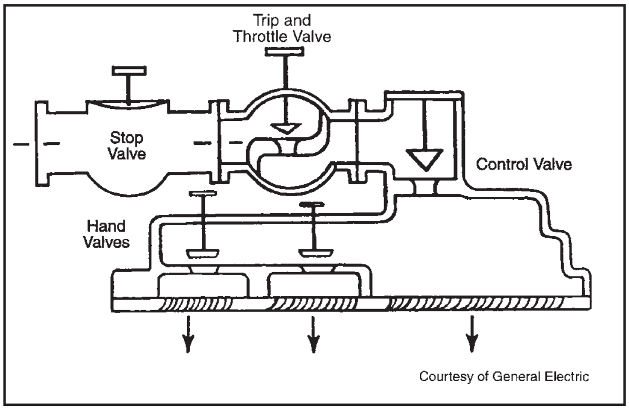



**FIG. 15-6**
**Single Valve with Hand Valves**




(Fig. 15-8). Multi-valve turbines have higher efficiencies at reduced loads because only the flow through one of the control valves is incurring a throttling loss (Fig. 15-9).

Turbines with a single control valve will often employ hand valves to improve efficiency at reduced loads. For the turbine shown in Fig. 15-6 both hand valves would be open at or near full load. As the load on the unit is reduced one or both of these hand valves can be closed to reduce throttling loss. Fig. 15-10 shows the efficiency advantage at reduced loads.

## Nozzles/Blades (Buckets)

On constant speed turbines a design objective is to avoid all bucket resonances at the operating speed. On variable speed turbines, although the design objectives remain the same, it is seldom possible to avoid all blade resonance because of the wide operating speed range. In these cases it is important to identify all blade resonance and to verify that all stresses are well below the material strength.

## Exhaust Casings

Turbine exhaust casings are categorized by pressure service (condensing or non-condensing) and number of rows of the last stage buckets (single flow, double flow, triple flow). Non-condensing exhausts are usually cast steel with most of the applications between 50 and 700 psig exhaust pressure. Most con-

**FIG. 15-7**

**Loss in Available Energy of Steam due to 10% Throttling**

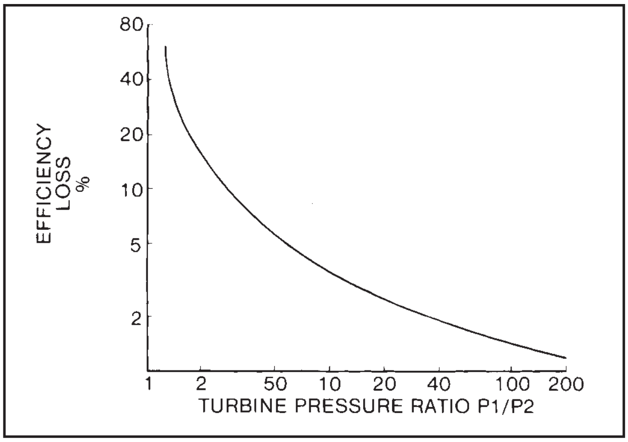
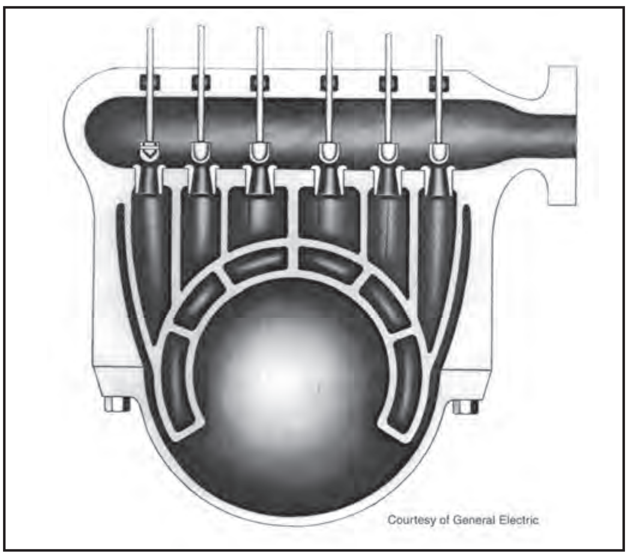



**FIG. 15-8**

**Multi-Valve Inlet**




densing exhausts are steel fabrications although some utilize cast iron construction. Maximum exhaust flange velocities are typically 450 ft/sec. Velocities above this level will usually result in substantial increases in exhaust hood losses and will decrease turbine efficiency.

## Moisture Protection

As steam expands through the turbine both the pressure and temperature are reduced. On most condensing and some non-condensing exhaust applications, the steam crosses the saturation line thereby introducing moisture into the steam path. The water droplets which are formed strike the buckets and can cause erosion of the blades. In addition, as the water is centrifuged from the blades, the water droplets strike the stationary components, also causing erosion. Where the moisture content is greater than 4%, moisture separators, which are internal to the turbine, can be used to remove a large percentage of the moisture, improving the turbine efficiency and reducing the impact erosion on the buckets. Stainless steel moisture shields can also be used to minimize the impact erosion of the stationary components.

**FIG. 15-9**

**Multi-Valve vs Single-Valve Performance Characteristic (Typical Non-Condensing Turbine) Turbine Pressure Ratio = 8.0**

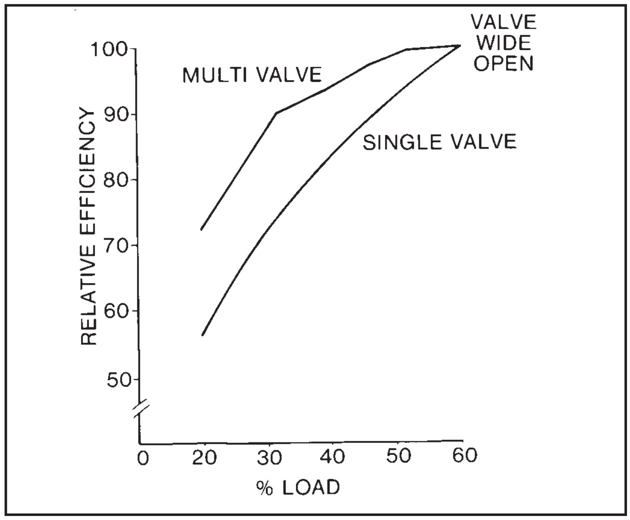
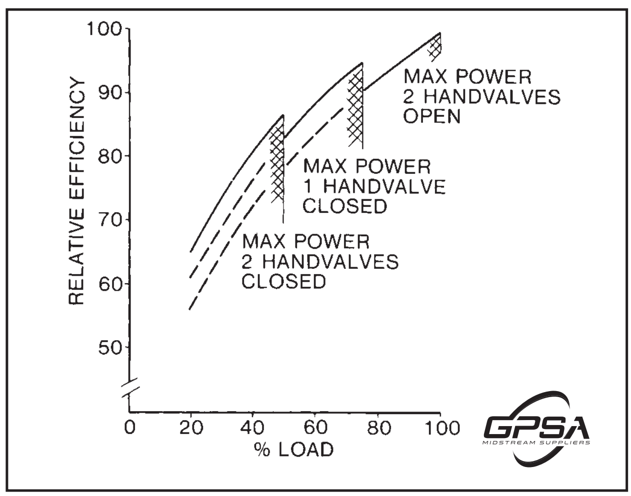



**FIG. 15-10**

**Single-Valve with Hand Valves Performance Characteristic (Typical Non-Condensing Turbine) Turbine Pressure Ratio = 8.0**




## Control Systems

Mechanical governors were the first generation control systems employed on mechanical drive turbines. Shaft speed is sensed by a fly-ball governor with hydraulic relays providing the input to the control valve. A second generation control system was developed and utilized analog control circuitry with the fly-ball governor replaced by speed pick-ups and the hydraulic relays with electronic circuit boards. A third generation control system was developed and replaced the electronic circuitry with digital logic. A microprocessor is used and the control logic is programmed into the governor. The major advantage of this system is the ability to utilize two governors simultaneously, each capable of governing the turbine alone. If the primary governor incurs a fault, the back-up governor assumes control of the turbine and provides diagnostic information to the operator.

## STEAM TURBINE EFFICIENCY

### Factors Affecting Efficiency

The objective of the steam turbine is to maximize the use of the available steam energy where the available steam energy is defined as the difference between the inlet and exhaust energies (enthalpies) for a 100% efficient constant entropy (i.e., isentropic) process. There are numerous loss mechanisms which reduce the efficiency from the isentropic such as throttling losses, steam leakage, friction between the steam and the nozzles/buckets, bearing losses, etc. Efficiency can range from a low of 40% for a low horsepower single-stage turbine to a high approaching 90% for a large multistage, multi-valve turbine.

### Techniques to Improve Efficiency

Various techniques are employed to maximize turbine efficiency, each designed to attack a specific loss mechanism. For example, the number of stages utilized can range from the fewest possible to develop the load reliably to the thermodynamically optimum selection. Spill bands can be utilized to minimize throttling losses. High efficiency nozzle/bucket profiles are available to reduce friction losses. Exhaust flow guides are available to reduce the pressure within the exhaust casing.

The specific features employed on a given application are usually based on the trade-off between capital investment and the cost to produce steam over the life of the turbine.

### Operation at Part Load

Most equipment driven by steam turbines are centrifugal machines where horsepower varies as the cube of speed. Part load efficiency varies as a function of speed, flow, and the number of stages. By assuming horsepower to vary as the cube of speed the turbine part load efficiency can be approximated as a percentage of the design efficiency (Fig. 15-11).

## EXAMPLES

Figs. 15-11 through 15-19 and 24-30 and 24-31 allow estimates to be made of steam rate, turbine efficiency, number of stages, and the inlet and exhaust nozzle diameters. The following examples illustrate the use of these figures:

**Example 15-1** — Given a steam turbine application with the following characteristics:

| | |
|---|---|
| Inlet Pressure | 600 psia |
| Inlet Temperature | 750°F |
| Exhaust Pressure | 2 psia |
| Required Horsepower | 6000 hp |
| Speed | 7000 rpm |

Determine:

- The actual steam rate (ASR).
- The inlet and exhaust nozzle diameters.
- The approximate number of stages.
- The steam rate at a partial load of 4000 hp and 6100 rpm.

**Solution Steps**

Using Figs. 24-30 and 31, the theoretical steam rate (TSR) may be determined from the difference in the inlet enthalpy and the theoretical exhaust enthalpy (i.e. isentropic exhaust

**FIG. 15-11**

**Part Load Efficiency Correction Factor vs Percent Power Multi-Valve Steam Turbines**

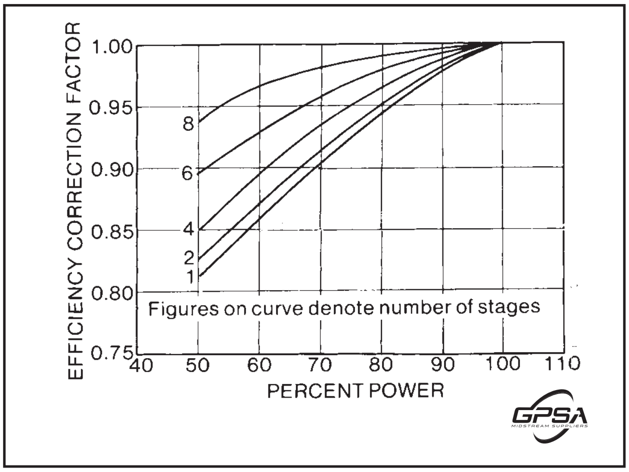
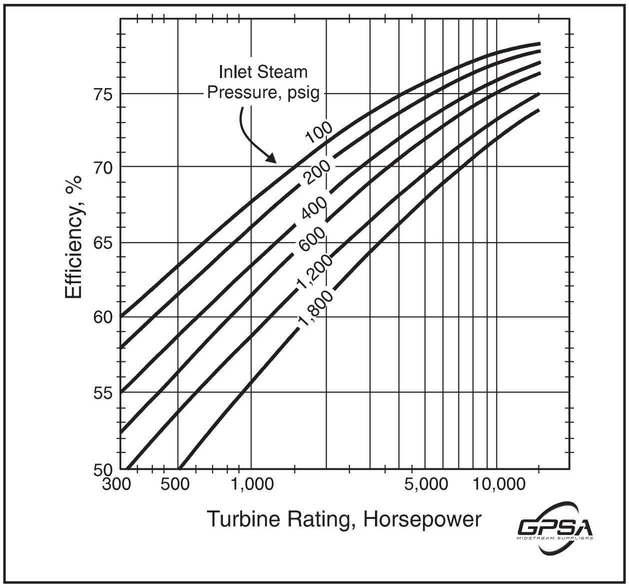



**FIG. 15-12**

**Basic Efficiency of Multi-Valve, Multi-Stage Condensing Turbines**



enthalpy), but first the inlet and exhaust states should be confirmed. Fig. 24-31 for superheated steam indicates that the inlet is superheated (i.e., 750°F is above the saturation temperature of 486.2°F), and gives an inlet entropy of 1.6109 Btu/(lb · °F). From Fig. 24-30, for saturated steam at the turbine exhaust pressure of 2 psia absolute, the liquid and vapor entropies are 0.1750 and 1.9200 Btu/(lb · °F). Since the inlet entropy is within this range, the theoretical exhaust must be two-phase. Had the exhaust-vapor entropy been equal to the inlet entropy, the exhaust would be single-phase vapor (i.e. at its dewpoint). Had the exhaust-vapor entropy been below the inlet entropy, the assumed two-phase exhaust would have been incorrect and Fig. 24-31 instead of 24-30 would be applicable.

Inlet conditions at 600 psia and 750°F (the average of the values at 700°F and 800°F on Fig. 24-31):

s = 1.6109 Btu/(lb · °F)

h = 1379.4 Btu/lb

Exhaust conditions at 2.0 psia:

$s_f$ = 0.1750 Btu/(lb · °F)

$s_g$ = 1.9200 Btu/lb · °F)

$h_f$ = 94.03 Btu/lb

$h_g$ = 1116.2 Btu/lb

Letting x equal the liquid fraction in the exhaust, and equating the inlet and exhaust entropies:

1.6109 = x (0.1750) + (1 – x)(1.9200)

x = 0.1771

1 – x = 0.8229 (vapor fraction in the exhaust)

Exhaust enthalpy = (0.1771)(94.03) + (0.8229)(1116.2)

= 935.2 Btu/lb

Enthalpy change = 935.2 – 1379.4

= –444.2 Btu/lb

Substituting Btu = (hp · hr)/2544:

Enthalpy change = (–444.2/2544) = (–1/5.727)(hp · hr)/lb

TSR = the absolute value of the inverse of the enthalpy change = 5.727 lb/(hp · hr)

Basic efficiency = 0.729 (Fig. 15-12)

Inlet saturation temperature = 486.2°F (first column Fig. 24-31)

Inlet superheat = 750 – 486 = 264°F

Superheat efficiency-correction factor = 1.03 (Fig. 15-14)

**FIG. 15-13**

**Basic Efficiency of Multi-Valve, Multi-Stage Non-Condensing Turbines**

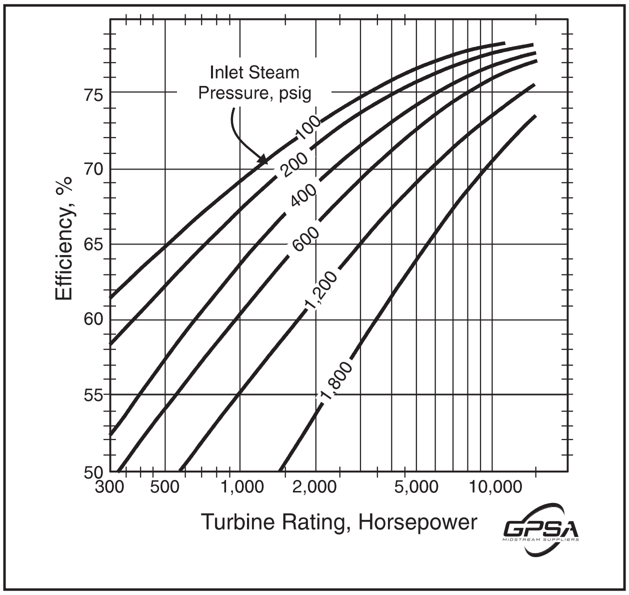



**FIG. 15-14**

**Superheat Efficiency Correction Factor for Condensing Turbines**

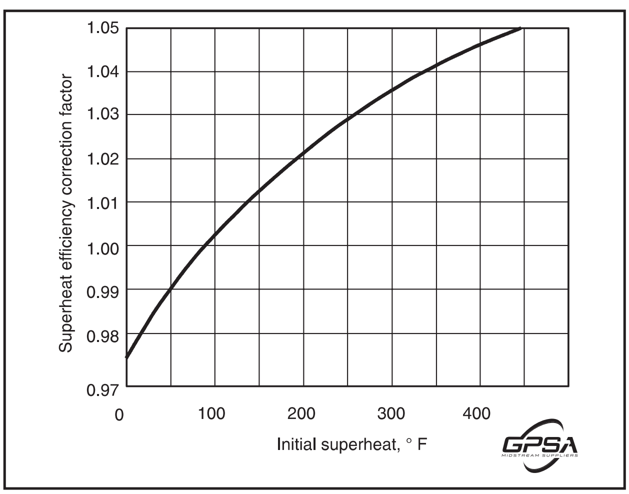
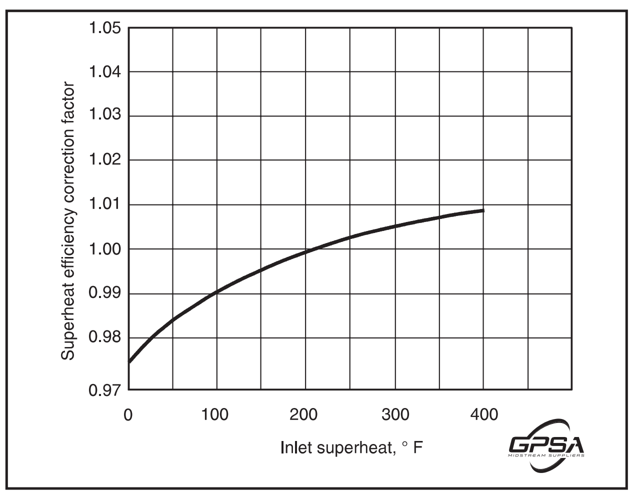



**FIG. 15-15**

**Superheat Efficiency Correction Factor for Non-Condensing Turbines**



Speed efficiency-correction factor = 0.957 (Fig. 15-16)

Corrected efficiency = (0.729)(1.03)(0.957) = 0.719

ASR = 5.727/0.719 = 7.97 lb/(hp · hr)

F = (6000 hp) 7.97 lb/(hp · hr)

= 47,800 lb/hr

The inlet and exhaust diameters may be estimated from the equation:

$$D = \sqrt{\frac{(0.051)\,(F)}{(\rho v)}} \qquad \textbf{Eq 15-1}$$

A reasonable rule of thumb for maximum velocity of the inlet steam is 150 (ft/sec).

$\rho = 0.88\ lb/ft^3$ @ 600 psia and 750°F

$$D = \sqrt{\frac{(0.051)\,(47{,}800)}{(0.88)\,(150)}}$$

D = 4.3 in.

A 4 in. NPS (minimum) inlet nozzle would be selected.

For exhaust sizing a maximum steam velocity of 450 ft/sec is a reasonable rule of thumb.

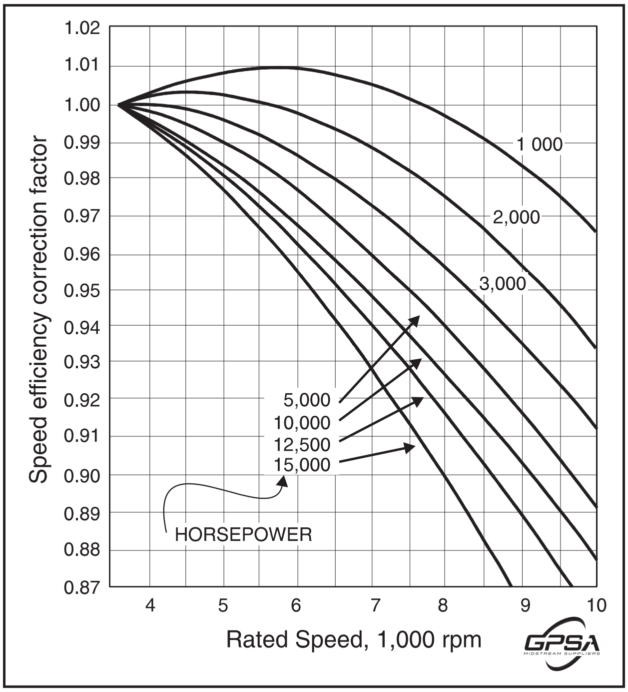
$\rho = 0.0057\ lb/ft^3$ @ 2 psia

$$D = \sqrt{\frac{(0.051)\,(47{,}800)}{(0.0057)\,(450)}}$$

D = 30.8 in.

**FIG. 15-16**

**Speed Efficiency Correction Factor for Condensing and Non-Condensing Turbines**




A 30 in. exhaust nozzle would be selected.

The number of stages may be estimated using Fig. 15-18. Drawing a horizontal line from the 7000 RPM indicates that between 1.5 and 2 stages per 100 Btu/lb of available energy would be acceptable.

Available Energy (theoretical)(i.e., the isentropic enthalpy change calculated above)

= 444.2 Btu/lb

Number of Stages

$$= \frac{(1.5)\,(444)}{(100)} = 7 \text{ (approximately)}$$

or, Number of Stages

$$= \frac{(2)\,(444)}{(100)} = 9 \text{ (approximately)}$$

Nine stages would provide increased efficiency but at additional cost.

At partial load of 4000 hp and 6,100 RPM and assuming seven stages from Fig. 15-11, a part load efficiency factor of approximately 0.96 is obtained. From Fig. 15-12, the basic efficiency at 4000 hp and 6,100 RPM is estimated to be 0.71.

Efficiency = (0.96) (0.71) = 0.68

Actual Steam Rate = 5.73/0.68 = 8.43 lb/(hp · hr)

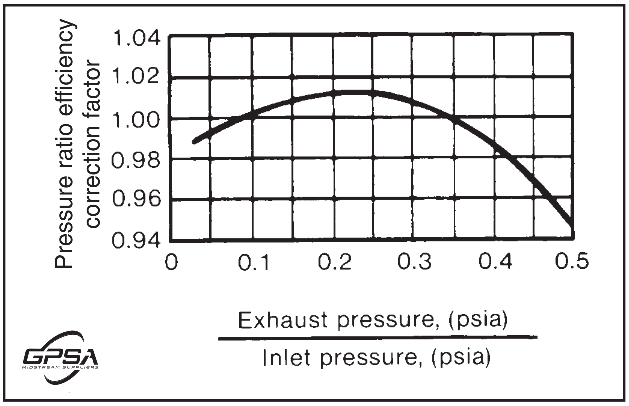
F = (4000) (8.43) = 33,700 lb/hr

**Example 15-2** — Determine the ASR and total steam requirements for a multi-stage turbine and a single-stage turbine at the following conditions:

| | |
|---|---|
| Inlet Pressure | 250 psig |
| Outlet Pressure | 100 psig |
| Inlet Temperature | 500°F |
| Horsepower | 900 hp |
| Speed | 5000 rpm |

**FIG. 15-17**

**Pressure Ratio Efficiency Correction Factor, Non-Condensing Turbines**



# CONVERSION FACTORS
## SI - METRIC/DECIMAL SYSTEM

### ABBREVIATIONS

| | |
|---|---|
| abs | absolute |
| ata | atmosphere absolute |
| Btu | British thermal unit |
| Btu/hr | British thermal unit/ hour |
| °C | Celsius |
| cfm | cubic foot/minute |
| cm | centimeter |
| cm$^2$ | square centimeter |
| cm$^3$ | cubic centimeter |
| cu.ft. | cubic foot |
| °F | Fahrenheit |
| ft/sec | foot/second |
| ft-lb | foot-pound |
| gal | gallon |
| hp | horsepower |
| in | inch |
| in. Hg | inch mercury |
| in. H$_2$0 | inch water |
| kcal | kilocalorie |
| kg | kilogram |
| kJ | kilojoule |
| kPa | kilopascal |
| kW | kilowatt |
| L | liter |
| m | meter |
| mm | millimeter |
| m$^2$ | square meter |
| m$^3$ | cubic meter |
| m$^3$/min | cubic meter/minute |
| mph | mile per hour |
| N | Newton |
| N/m$^2$ | Pascal |
| Nm$^3$/hr | normal* cubic meter/hour |
| psi | pound/square inch |
| psia | pound/square inch absolute |
| psig | pound/square inch gage |
| scf | standard* cubic foot |
| scfm | standard* cubic foot/minute |
| sq | square |

* "Normal" = 0°C and 1.01325 x 10$^5$ Pascals
* "Standard" = 59°F and 14.73 psia

### MILLIMETERS (mm) TO INCHES (in)
(1 millimeter = 0.03937 inch)

| mm | in | mm | in | mm | in | mm | in | mm | in |
|---|---|---|---|---|---|---|---|---|---|
| **1** | 0.039 | **21** | 0.827 | **41** | 1.614 | **61** | 2.402 | **81** | 3.189 |
| **2** | 0.079 | **22** | 0.866 | **42** | 1.654 | **62** | 2.441 | **82** | 3.228 |
| **3** | 0.118 | **23** | 0.906 | **43** | 1.693 | **63** | 2.480 | **83** | 3.268 |
| **4** | 0.157 | **24** | 0.945 | **44** | 1.732 | **64** | 2.520 | **84** | 3.307 |
| **5** | 0.197 | **25** | 0.984 | **45** | 1.772 | **65** | 2.559 | **85** | 3.346 |
| **6** | 0.236 | **26** | 1.024 | **46** | 1.811 | **66** | 2.598 | **86** | 3.386 |
| **7** | 0.276 | **27** | 1.063 | **47** | 1.850 | **67** | 2.638 | **87** | 3.425 |
| **8** | 0.315 | **28** | 1.102 | **48** | 1.890 | **68** | 2.677 | **88** | 3.465 |
| **9** | 0.354 | **29** | 1.142 | **49** | 1.929 | **69** | 2.717 | **89** | 3.504 |
| **10** | 0.394 | **30** | 1.181 | **50** | 1.968 | **70** | 2.756 | **90** | 3.543 |
| **11** | 0.433 | **31** | 1.220 | **51** | 2.008 | **71** | 2.795 | **91** | 3.583 |
| **12** | 0.472 | **32** | 1.260 | **52** | 2.047 | **72** | 2.835 | **92** | 3.622 |
| **13** | 0.512 | **33** | 1.299 | **53** | 2.087 | **73** | 2.874 | **93** | 3.661 |
| **14** | 0.551 | **34** | 1.339 | **54** | 2.126 | **74** | 2.913 | **94** | 3.701 |
| **15** | 0.591 | **35** | 1.378 | **55** | 2.165 | **75** | 2.953 | **95** | 3.740 |
| **16** | 0.630 | **36** | 1.417 | **56** | 2.205 | **76** | 2.992 | **96** | 3.779 |
| **17** | 0.669 | **37** | 1.457 | **57** | 2.244 | **77** | 3.032 | **97** | 3.819 |
| **18** | 0.709 | **38** | 1.496 | **58** | 2.283 | **78** | 3.071 | **98** | 3.858 |
| **19** | 0.748 | **39** | 1.535 | **59** | 2.323 | **79** | 3.110 | **99** | 3.898 |
| **20** | 0.787 | **40** | 1.575 | **60** | 2.362 | **80** | 3.150 | **100** | 3.937 |

### KILOGRAMS (kg) TO POUNDS (lb)
(1 kilogram = 2.20462 pounds)

| kg | lb | kg | lb | kg | lb | kg | lb | kg | lb |
|---|---|---|---|---|---|---|---|---|---|
| **1** | 2.204 | **21** | 46.297 | **41** | 90.390 | **61** | 134.482 | **81** | 178.574 |
| **2** | 4.409 | **22** | 48.502 | **42** | 92.594 | **62** | 136.687 | **82** | 180.779 |
| **3** | 6.614 | **23** | 50.706 | **43** | 94.799 | **63** | 138.891 | **83** | 182.984 |
| **4** | 8.819 | **24** | 52.911 | **44** | 97.003 | **64** | 141.096 | **84** | 185.188 |
| **5** | 11.023 | **25** | 55.116 | **45** | 99.208 | **65** | 143.300 | **85** | 187.393 |
| **6** | 13.228 | **26** | 57.320 | **46** | 101.413 | **66** | 145.505 | **86** | 189.598 |
| **7** | 15.432 | **27** | 59.525 | **47** | 103.617 | **67** | 147.710 | **87** | 191.802 |
| **8** | 17.637 | **28** | 61.729 | **48** | 105.822 | **68** | 149.914 | **88** | 194.007 |
| **9** | 19.843 | **29** | 63.934 | **49** | 108.026 | **69** | 152.119 | **89** | 196.211 |
| **10** | 22.046 | **30** | 66.139 | **50** | 110.231 | **70** | 154.324 | **90** | 198.416 |
| **11** | 24.251 | **31** | 66.343 | **51** | 112.436 | **71** | 156.528 | **91** | 200.621 |
| **12** | 26.455 | **32** | 70.548 | **52** | 114.640 | **72** | 158.733 | **92** | 202.825 |
| **13** | 28.660 | **33** | 72.753 | **53** | 116.845 | **73** | 160.937 | **93** | 205.030 |
| **14** | 30.865 | **34** | 74.957 | **54** | 119.050 | **74** | 163.142 | **94** | 207.235 |
| **15** | 33.069 | **35** | 77.162 | **55** | 121.254 | **75** | 165.347 | **95** | 209.439 |
| **16** | 35.274 | **36** | 79.366 | **56** | 123.459 | **76** | 167.551 | **96** | 211.644 |
| **17** | 37.479 | **37** | 81.571 | **57** | 125.663 | **77** | 169.756 | **97** | 213.848 |
| **18** | 39.683 | **38** | 83.776 | **58** | 127.868 | **78** | 171.961 | **98** | 216.053 |
| **19** | 41.888 | **39** | 85.980 | **59** | 130.073 | **79** | 174.165 | **99** | 218.258 |
| **20** | 44.093 | **40** | 88.185 | **60** | 132.277 | **80** | 176.370 | **100** | 220.462 |

### CONVERSION FACTORS

| TO CONVERT FROM ENGLISH | TO S.I. METRIC | MULTIPLY BY | TO OLD METRIC | MULTIPLY BY |
|---|---|---|---|---|
| sq. in. | mm$^2$ | **645.16** | cm$^2$ | **6.4516** |
| sq. ft. | m$^2$ | **0.0929** | m$^2$ | **0.0929** |
| lb/cu.ft. | kg/m$^3$ | **16.0185** | kg/m$^3$ | **16.0185** |
| lb$_f$ | N | **4.4482** | N | **4.4482** |
| lb$_f$/ft | N/m | **14.5939** | N/m | **14.5939** |
| Btu | kJ | **1.0551** | kcal | **0.252** |
| Btu/hr | W | **0.2931** | kcal/hr | **0.252** |
| Btu/scf | kJ/mm$^3$ | **37.2590** | kcal/nm$^3$ | **0.1565** |
| in | mm | **25.400** | cm | **2.540** |
| ft | m | **0.3048** | m | **0.3048** |
| yd | m | **0.914** | m | **0.914** |
| lb | kg | **0.4536** | kg | **0.4536** |
| hp | kW | **0.7457** | kW | **0.7457** |
| psi | kPa | **6.8948** | kg/cm$^2$ | **0.070** |
| psia | kPa abs | **6.8948** | bars abs | **0.0716** |
| psig | kPa gage | **6.8948** | ata | **0.070** |
| in. Hg | kPa | **3.3769** | cm Hg | **2.540** |
| in. H$_2$0 | kPa | **0.2488** | cm H$_2$0 | **2.540** |
| °F | °C = | **(°F -32) 5/9** | °C = | **(°F -32) 5/9** |
| °F (Interval) | °C (Interval) | **5/9** | °C (Interval) | **5/9** |
| ft-lb | N • m | **1.3558** | N • m | **1.3558** |
| mph | km/hr | **1.6093** | km/hr | **1.6093** |
| ft/sec | m/sec | **0.3048** | m/sec | **0.3048** |
| cu. ft. | m$^3$ | **0.0283** | m$^3$ | **0.0283** |
| gas (US) | L | **3.7854** | L | **3.7854** |
| cfm | m$^3$/min | **0.0283** | m$^3$/min | **0.0283** |
| scfm | nm$^3$/min | **0.0268** | nm$^3$/hr | **1.61** |

| TO CONVERT FROM OLD METRIC | TO S.I. METRIC | MULTIPLY BY |
|---|---|---|
| cm$^2$ | mm$^2$ | **100.** |
| kcal | kJ | **4.1868** |
| kcal/hr | W | **1.16279** |
| cm | mm | **10.** |
| kg/cm$^2$ | kPa | **98.0665** |
| bars | kPa | **100.** |
| atm | kPa | **101.325** |
| cm Hg | kPa | **1.3332** |
| cm H$_2$0 | kPa | **9.8064** |
| nm$^3$/hr | nm$^3$/min | **0.0176** |

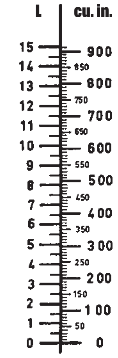
### TEMPERATURE CONVERSION TABLES*

| | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **0 TO 100** | | | 2.78 | **37** | 98.6 | 23.9 | **75** | 167.0 | 93 | **200** | 392 | 299 | **570** | 1058 | 510 | **950** | 1742 |
| -17.8 | **0** | 32 | 3.33 | **38** | 100.4 | 24.4 | **76** | 168.8 | 99 | **210** | 410 | 304 | **580** | 1076 | 516 | **960** | 1760 |
| -17.2 | **1** | 33.8 | 3.89 | **39** | 102.2 | 25.0 | **77** | 170.6 | 100 | **212** | 413 | 310 | **590** | 1094 | 521 | **970** | 1778 |
| -16.7 | **2** | 35.6 | 4.44 | **40** | 104.0 | 25.6 | **78** | 172.4 | 104 | **220** | 428 | 316 | **600** | 1112 | 527 | **980** | 1796 |
| -16.1 | **3** | 37.4 | 5.00 | **41** | 105.8 | 26.1 | **79** | 174.2 | 110 | **230** | 446 | 321 | **610** | 1130 | 532 | **990** | 1814 |
| -15.6 | **4** | 39.2 | 5.56 | **42** | 107.6 | 26.7 | **80** | 176.0 | 116 | **240** | 464 | 327 | **620** | 1148 | 538 | **1000** | 1832 |
| -15.0 | **5** | 41.0 | 6.11 | **43** | 109.4 | 27.2 | **81** | 177.8 | 121 | **250** | 482 | 332 | **630** | 1166 | | | |
| -14.4 | **6** | 42.8 | 6.67 | **44** | 111.2 | 27.8 | **82** | 179.6 | 127 | **260** | 500 | 338 | **640** | 1184 | **1000 TO 1630** | | |
| -13.9 | **7** | 44.9 | 7.22 | **45** | 113.0 | 28.3 | **83** | 181.4 | 132 | **270** | 518 | 343 | **650** | 1202 | 538 | **1000** | 1832 |
| -13.3 | **8** | 46.4 | 7.78 | **46** | 114.8 | 28.9 | **84** | 183.2 | 138 | **280** | 536 | 349 | **660** | 1220 | 543 | **1010** | 1850 |
| -12.8 | **9** | 48.2 | 8.33 | **47** | 116.6 | 29.4 | **85** | 185.0 | 143 | **290** | 554 | 354 | **670** | 1238 | 549 | **1020** | 1868 |
| -12.1 | **10** | 50.0 | 8.89 | **48** | 118.4 | 30.0 | **86** | 186.8 | 149 | **300** | 572 | 360 | **680** | 1256 | 554 | **1030** | 1886 |
| -11.7 | **11** | 51.8 | 9.44 | **49** | 120.0 | 30.6 | **87** | 188.6 | 154 | **310** | 590 | 366 | **690** | 1274 | 560 | **1040** | 1904 |
| -11.1 | **12** | 53.6 | 10.0 | **50** | 122.0 | 31.1 | **88** | 190.4 | 160 | **320** | 608 | 371 | **700** | 1292 | 566 | **1050** | 1922 |
| -10.6 | **13** | 55.4 | 10.6 | **51** | 123.8 | 31.7 | **89** | 192.2 | 166 | **330** | 626 | 377 | **710** | 1310 | 571 | **1060** | 1940 |
| -10.0 | **14** | 57.2 | 11.1 | **52** | 125.6 | 32.2 | **90** | 194.0 | 171 | **340** | 644 | 382 | **720** | 1328 | 577 | **1070** | 1958 |
| -9.44 | **15** | 59.0 | 11.7 | **53** | 127.4 | 32.8 | **91** | 195.8 | 177 | **350** | 662 | 388 | **730** | 1346 | 582 | **1080** | 1976 |
| -8.89 | **16** | 60.8 | 12.2 | **54** | 129.2 | 33.3 | **92** | 197.6 | 182 | **360** | 680 | 393 | **740** | 1364 | 588 | **1090** | 1994 |
| -8.33 | **17** | 62.6 | 12.8 | **55** | 131.0 | 33.9 | **93** | 199.4 | 188 | **370** | 698 | 399 | **750** | 1382 | 593 | **1100** | 2012 |
| -7.78 | **18** | 64.4 | 13.3 | **56** | 132.8 | 34.4 | **94** | 201.2 | 193 | **380** | 716 | 404 | **760** | 1400 | 599 | **1110** | 2030 |
| -7.22 | **19** | 66.2 | 13.9 | **57** | 134.6 | 35.0 | **95** | 203.0 | 199 | **390** | 734 | 410 | **770** | 1418 | 604 | **1120** | 2048 |
| -6.67 | **20** | 68.0 | 14.4 | **58** | 136.4 | 35.6 | **96** | 204.8 | 204 | **400** | 752 | 416 | **780** | 1436 | 610 | **1130** | 2066 |
| -6.11 | **21** | 69.8 | 15.0 | **59** | 138.2 | 36.1 | **97** | 206.6 | 210 | **410** | 770 | 421 | **790** | 1454 | 816 | **1500** | 2732 |
| -5.56 | **22** | 71.6 | 15.6 | **60** | 140.0 | 36.7 | **98** | 208.4 | 216 | **420** | 788 | 427 | **800** | 1472 | 821 | **1510** | 2750 |
| -5.00 | **23** | 73.4 | 16.1 | **61** | 141.8 | 37.2 | **99** | 210.2 | 221 | **430** | 806 | 432 | **810** | 1490 | 827 | **1520** | 2768 |
| -4.44 | **24** | 75.2 | 16.7 | **62** | 143.6 | 37.8 | **100** | 212.0 | 227 | **440** | 824 | 438 | **820** | 1508 | 832 | **1530** | 2786 |
| -3.89 | **25** | 77.0 | 17.2 | **63** | 145.4 | | | | 232 | **450** | 842 | 443 | **830** | 1526 | 838 | **1540** | 2804 |
| -3.33 | **26** | 78.8 | 17.8 | **64** | 147.2 | **100 TO 1000** | | | 238 | **460** | 860 | 449 | **840** | 1544 | 843 | **1550** | 2822 |
| -2.78 | **27** | 80.6 | 18.3 | **65** | 149.0 | 38 | **100** | 212 | 243 | **470** | 878 | 454 | **850** | 1562 | 849 | **1560** | 2840 |
| -2.22 | **28** | 82.4 | 18.9 | **66** | 150.8 | 43 | **110** | 230 | 249 | **480** | 896 | 460 | **860** | 1580 | 854 | **1570** | 2858 |
| -1.67 | **29** | 84.2 | 19.4 | **67** | 152.6 | 49 | **120** | 248 | 254 | **490** | 914 | 466 | **870** | 1598 | 860 | **1580** | 2876 |
| -1.11 | **30** | 86.0 | 20.0 | **68** | 154.4 | 54 | **130** | 266 | 260 | **500** | 932 | 471 | **880** | 1616 | 866 | **1590** | 2894 |
| -0.56 | **31** | 87.8 | 20.6 | **69** | 156.2 | 60 | **140** | 284 | 266 | **510** | 950 | 477 | **890** | 1634 | 871 | **1600** | 2912 |
| 0 | **32** | 89.6 | 21.1 | **70** | 158.0 | 66 | **150** | 302 | 271 | **520** | 968 | 482 | **900** | 1652 | 877 | **1610** | 2930 |
| 0.56 | **33** | 91.4 | 21.7 | **71** | 159.8 | 71 | **160** | 320 | 277 | **530** | 986 | 488 | **910** | 1670 | 882 | **1620** | 2948 |
| 1.11 | **34** | 93.2 | 22.2 | **72** | 161.6 | 77 | **170** | 338 | 282 | **540** | 1004 | 493 | **920** | 1688 | 888 | **1630** | 2966 |
| 1.67 | **35** | 95.0 | 22.8 | **73** | 163.4 | 82 | **180** | 356 | 288 | **550** | 1022 | 499 | **930** | 1706 | | | |
| 2.22 | **36** | 96.8 | 23.3 | **74** | 165.2 | 88 | **190** | 374 | 293 | **560** | 1040 | 504 | **940** | 1724 | | | |

Note: The numbers in **bold** face type refer to the temperature either in degrees Centigrade or Fahrenheit which is desired to convert into the other scale. If converting from Fahrenheit degrees to Centigrade degrees, the equivalent temperatures will be found in the left column; while if converting from degrees Centigrade to degrees Fahrenheit, the answer will be found in the column on the right.

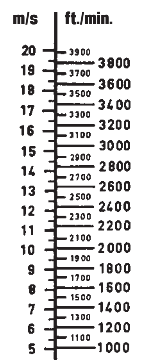
### VOLUME CONVERSION FACTORS

**1 L = 61.02 cu. in.**
**10 cu. in. = 0,164 L**



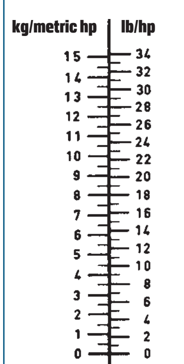

### PISTON SPEED CONVERSION FACTORS

**1 m/s = 196.9 ft./min.**
**100 ft./min. = 0,51 m/s**




### WEIGHT/ HORSEPOWER CONVERSION FACTORS

**1 kg/metric hp = 2.235 lb./hp**
**1 lb/hp = .4474 kg/metric hp**



**Solution Steps**

For a multi-stage turbine:

Examining Figs. 24-30 and 31 in the same way as in Example 15-1, the turbine inlet is superheated, and the exhaust is two-phase.

Inlet conditions at 250 psig (264.7 psia) and 500°F (interpolating linearly between 240 and 260 psia on Fig. 24-31):

s = 1.5873 Btu/(lb · °F)

h = 1261.8 Btu/lb

Exhaust conditions at 100 psig (i.e. 114.7 psia). From Fig. 24-30 interpolating linearly between 89.64 psia at 320°F and 117.99 psia at 340°F, get the following for 114.7 psia:

$s_f$ = 0.4872 Btu/(lb · °F)

$s_g$ = 1.5918 Btu/(lb · °F)

$h_f$ = 308.9 Btu/lb

$h_g$ = 1189.5 Btu/lb

Letting x equal the liquid fraction in the exhaust, and equating the inlet and exhaust entropies:

1.5873 = x (0.4872) + (1 – x) (1.5918)
x = 0.0041

1 – x = 0.9959 (fraction vapor in exhaust)

Exhaust enthalpy = (0.0041)(308.9) + (0.9959)(1189.5)
= 1185.9

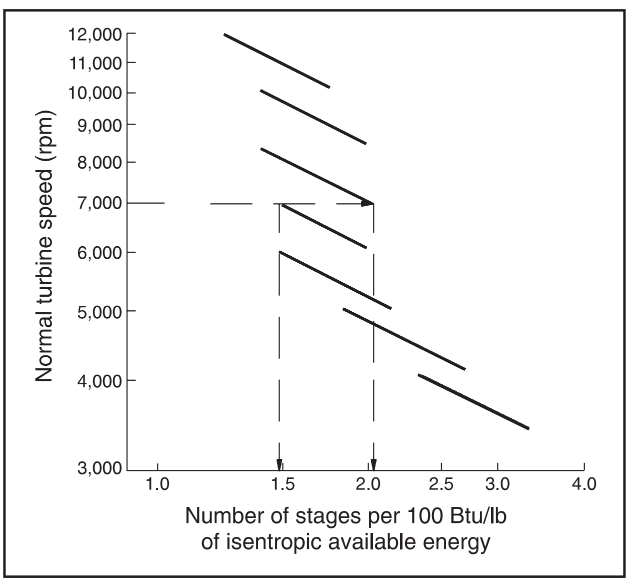
Enthalpy change = 1185.9 – 1261.8 = –75.9 Btu/lb

Substituting 1 Btu = (hp · hr)/2544:

Enthalpy change = (–75.9/2544) = (–1/33.5)(hp · hr)/lb)

**FIG. 15-18**

**Stages Required per 100 Btu/lb of Available Energy as a Factor of Normal Turbine Speed**




TSR = the absolute value of the inverse of the enthalpy change = 33.5 lb/(hp · hr)

Basic efficiency = 66% (Fig. 15-13)

Inlet saturation temperature = 406.0 °F (interpolating between 260 and 280 psia on Fig. 24-31)

Inlet superheat = 500 – 406 (Fig. 24-31) = 94°F

Efficiency-correction factor for superheat = 0.99 (Fig. 15-15)

Efficiency-correction factor for speed = 1.01 (Fig. 15-16)

Pressure ratio = (114.7 psia)/(264.7 psia) = 0.433

Efficiency-correction factor for pressure ratio = 0.97 (Fig. 15-17)

ASR = [33.5 lb/(hp · hr)]/[(0.66) (0.99) (1.01) (0.97)]

= 52.3 lb/(hp · hr)

F = [52.3 lb/(hp · hr)] (900 hp)

= 47,100 lb/hr

For a single-stage turbine

ASR = [75 lb/(hp · hr)] (0.93) (Fig. 15-19)

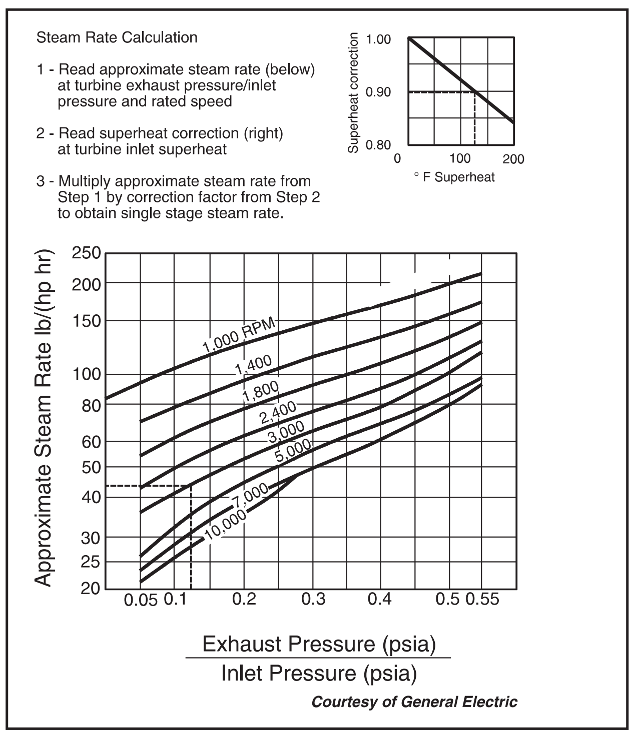
= 70 lb/(hp · hr)

F = [70 lb/(hp · hr)] (900 hp)

= 63,000 lb/hr

**FIG. 15-19**

**Single-Stage Application**



## GAS TURBINES

### General

Gas turbines are extensively used in all phases of the gas industry as a source of shaft power. They are used to drive compressors, generators, and other equipment required to produce, process, and transport natural gas. The main advantages of gas turbines are:

- Compact, light weight design.
- Minimal maintenance.
- Short installation time.

### Compact, Lightweight Design

The compact, lightweight design of gas turbines makes them ideally suited for offshore platform installations, portable generating sets, remote sites, or any application where size and weight are important considerations.

### Maintenance

Once installed, the gas turbine requires a minimum of routine maintenance. It is important to monitor the operating parameters of the turbine (pressures, temperatures, speed, vibration levels, etc.). This can often be done by an operator at a location remote from the actual turbine installation.

### Installation

The relatively light weight, compact size, and simple design of gas turbines make them an attractive choice where power must be quickly installed in the field. The gas turbine is often delivered on an integral one-piece baseplate with all auxiliary equipment installed and tested by the manufacturer. Thus, construction and start-up time are minimized.

## GAS TURBINE TYPES

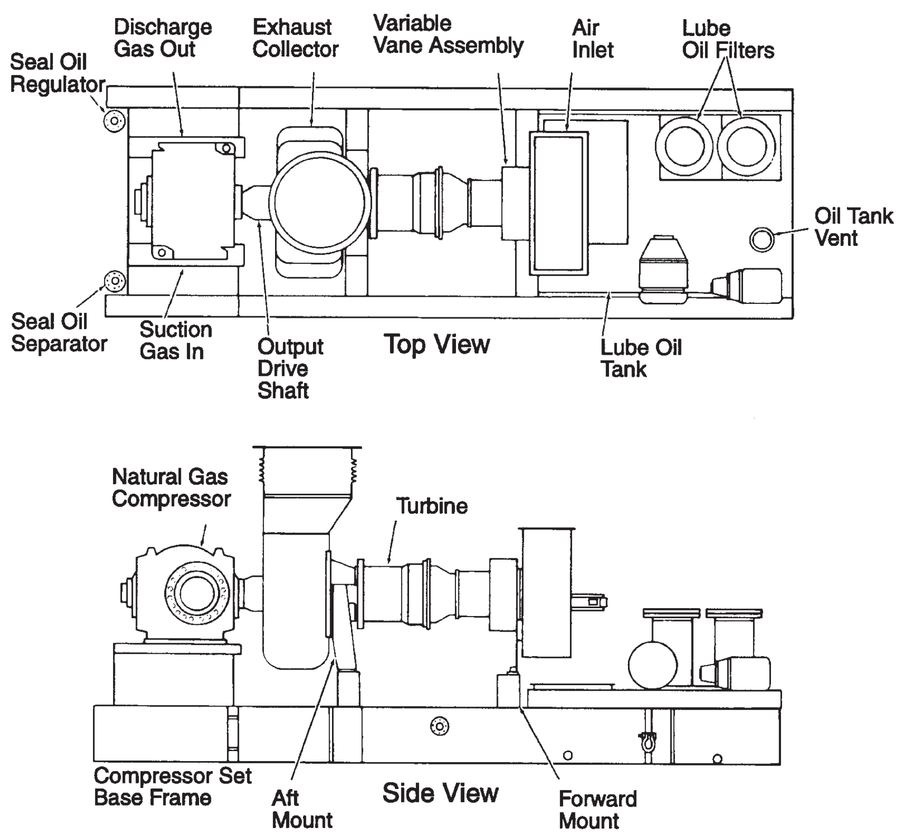
The gas turbine was first widely used as an aircraft power plant. However, as they became more efficient and durable, they were adapted to the industrial marketplace. Over the years the gas turbine has evolved into two basic types for high-power stationary applications: the industrial or heavy-duty design and the aircraft derivative design.

### Heavy Duty

The industrial type gas turbine is designed exclusively for stationary use. Where high power output is required, 35,000 hp and above, the heavy duty industrial gas turbine is normally specified. The industrial gas turbine has certain advantages

**FIG. 15-20**

**Typical Gas Turbine Skid Layout**




Courtesy of Solar Turbines Inc.

which should be considered when determining application requirements. Some of these are:

- Less frequent maintenance.
- Can burn a wider variety of fuels.
- Available in larger horsepower sizes.

## Aircraft Derivative

An aircraft derivative gas turbine is based on an aircraft engine design which has been adapted for industrial use. The engine was originally designed to produce shaft power and later as a pure jet. The adaptation to stationary use was relatively simple.

Some of the advantages of the aircraft derivative gas turbines are:

- Higher efficiency than industrial units.
- Quick overhaul capability.
- Lighter and more compact, an asset where weight limitations are important such as offshore installations.

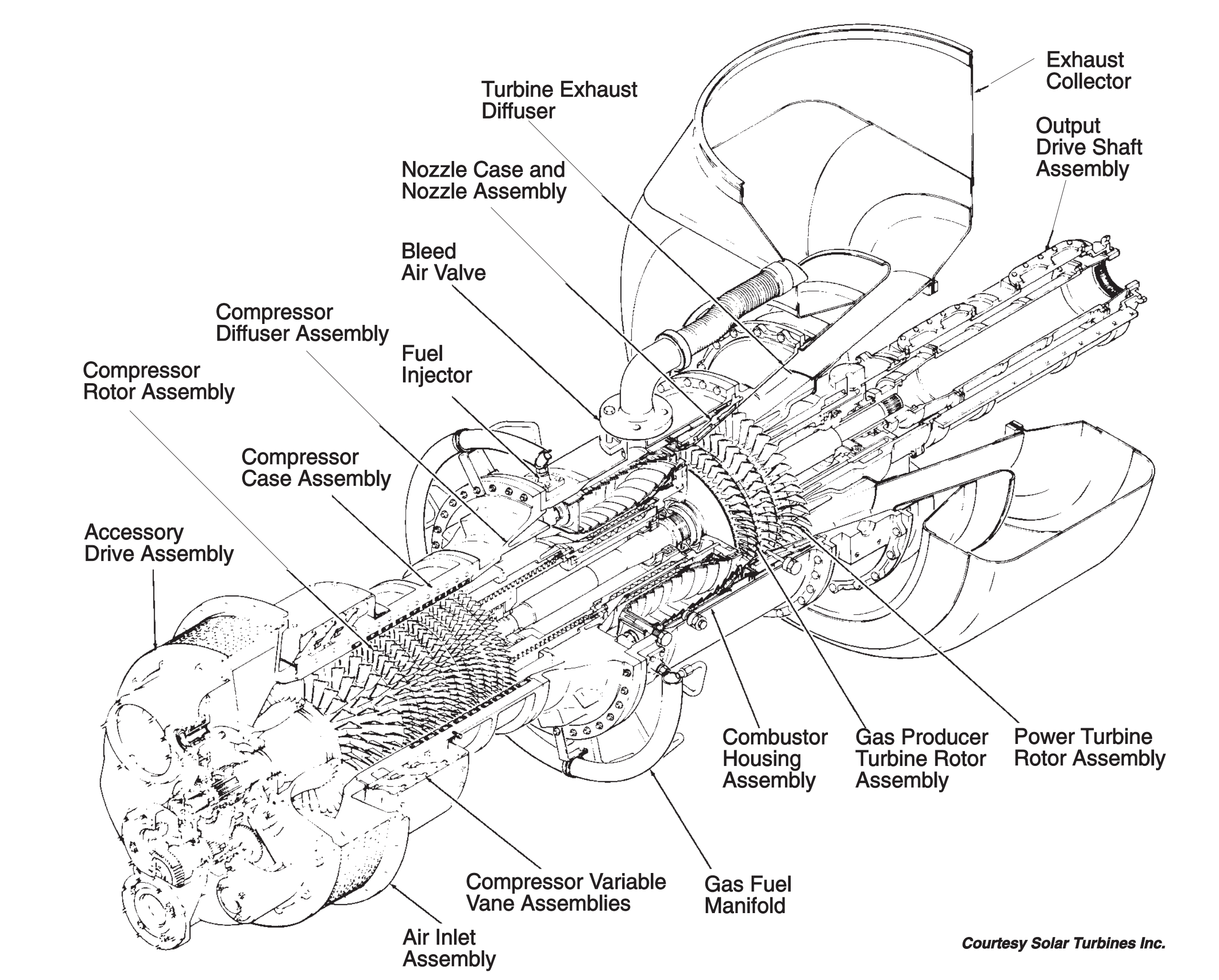
## Single Shaft/Split Shaft

Gas turbine designs are also differentiated by shaft configuration. In a single shaft design, all rotating components of the gas turbine are mounted on one shaft. In a split shaft design, the air compressor rotating components are mounted on one shaft, and the power turbine rotating components are mounted on another shaft. The driven equipment is connected to the power turbine shaft. The single shaft design is simpler, requiring fewer bearings, and is generally used where the speed range of the driven equipment is narrow or fixed (as in generator sets). It requires a powerful starting system since all the rotating components (including the driven equipment) must be accelerated to idle speed during the start cycle.

A split shaft design is advantageous where the driven equipment has a wide speed range or a high starting torque. The air compressor is able to run at its most efficient speed while the

**FIG. 15-21**

**Gas Turbine Internals**




*Courtesy Solar Turbines Inc.*

power turbine speed varies with the driven equipment. The split shaft design allows a much smaller starting system since only the air compressor shaft is accelerated during the start cycle.

## GAS TURBINE CYCLES

The basic gas turbine cycle is termed the Brayton cycle. The ideal Brayton cycle is a closed cycle consisting of an isentropic compression process; a constant pressure external heating process; an isentropic expansion process; and finally a constant pressure external cooling process which returns the working substance to the inlet state of the compression process. A schematic and TS diagram of the ideal Brayton cycle are shown in Fig. 15-22. The turbomachinery used in the process includes an axial flow or centrifugal compressor and an axial or radial flow turbine.

### Simple Open Cycle

The simple open cycle gas turbine takes atmospheric air into the compressor as the working substance. Following compression, the air enters the combustion chamber where the temperature is raised by the combustion of fuel. The gaseous combustion products are then expanded back to the atmosphere through a turbine. A diagram of this cycle is shown in Fig. 15-23. The turbine in this system derives enough power from the high temperature gas to drive both the compressor and load.

### Regenerative Ideal Brayton Cycle

The use of a regenerator in an ideal Brayton cycle acts to reduce the amount of available energy lost by external heat exchange. The system schematic is illustrated in Fig. 15-24. This available energy loss is due to irreversible heat input and is illustrated in Fig. 15-25. A heat exchanger or regenerator is placed in the system to transfer heat internally from the hot exhaust gas to the cooler air leaving the compressor. This preheating of the combustion air thus reduces the amount of external heat input needed to produce the same work output.

### Combined Cycle

Instead of using the hot exhaust gas for regeneration, this approach uses exhaust gas to generate steam. This steam can be used either as a supplement to the plant steam system or to generate additional horsepower in a Rankine cycle. In the basic Rankine cycle, the hot exhaust gas passes successively through the superheater, evaporator, and economizer of the steam generator before being exhausted to the atmosphere. The steam leaving the boiler is expanded through a steam turbine to generate additional power. The cycle is closed by the addition of a condenser and feed water pump completing a basic Rankine cycle. Since the steam cycle does not require any additional fuel to generate power, the overall thermal efficiency is increased. Fig. 15-26 shows schematically a typical installation and its TS diagrams.

## AUXILIARY SYSTEMS

### Lube Systems

Two types of oils are used in lubricating gas turbine equipment. They are mineral and fire-resistant synthetic based oils. The oil type used depends on the bearing construction of the particular turbine.

Babbitt type sleeve and thrust bearings, typical of heavy duty turbines, use a mineral based oil. Driven equipment such

**FIG. 15-22**
**Ideal Brayton Cycle**

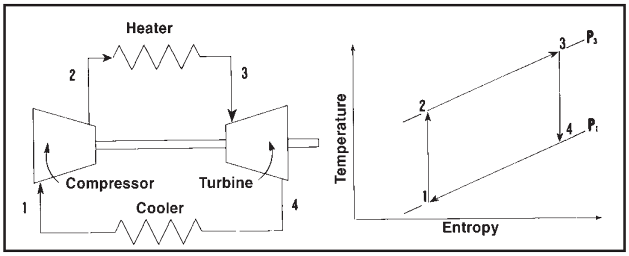



**FIG. 15-23**
**Simple Open Cycle**

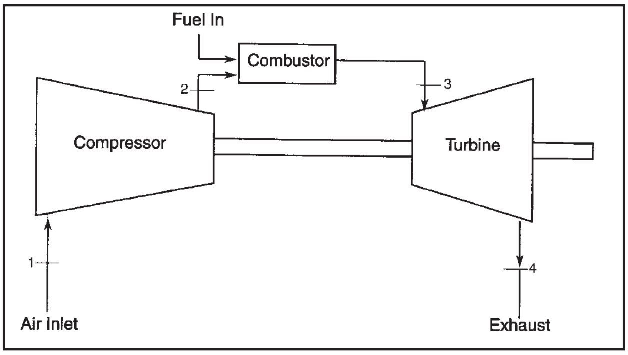



**FIG. 15-24**
**Regenerative Ideal Brayton Cycle**

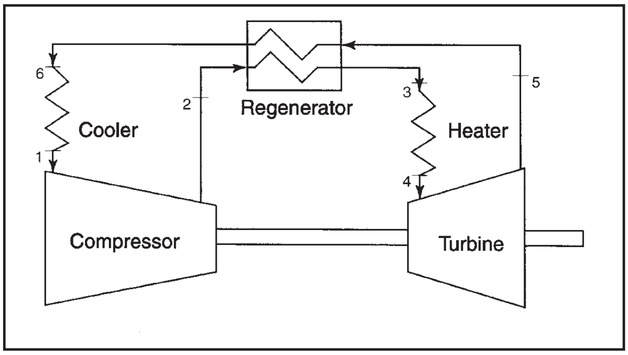
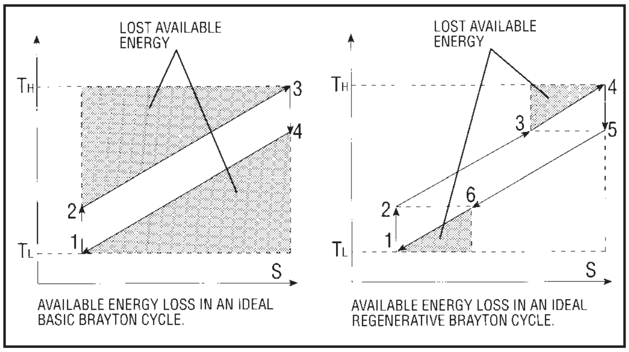



**FIG. 15-25**
**Ideal Brayton Cycle Available Energy**



as compressors, gear, and generators also use this type oil, thus a common, combined lube system can be provided for the train.

Aircraft derivative gas generators all incorporate anti-friction type ball and/or roller bearings. A synthetic oil is used in this service and is provided in a separate system from the mineral oil system used to lubricate the driven equipment. An oil scavenging system is also typical of these gas generators. Engine mounted pumps are used to scavenge oil from the main bearing pumps and return it to the reservoir.

## Air Filtration

The primary reason for inlet air filtration is to prevent unwanted dirt from entering the gas turbine. By reducing the contaminants which contribute to corrosion, erosion, and fouling, the gas turbine life is extended.

There are various types of filters. The main types are as follows:

**Inertial** — This type removes the larger particulates from the inlet air.

**Prefilters** — These are medium filters usually made of cotton fabrics or spun-glass fibers, used to extend the life of a high efficiency filter further downstream.

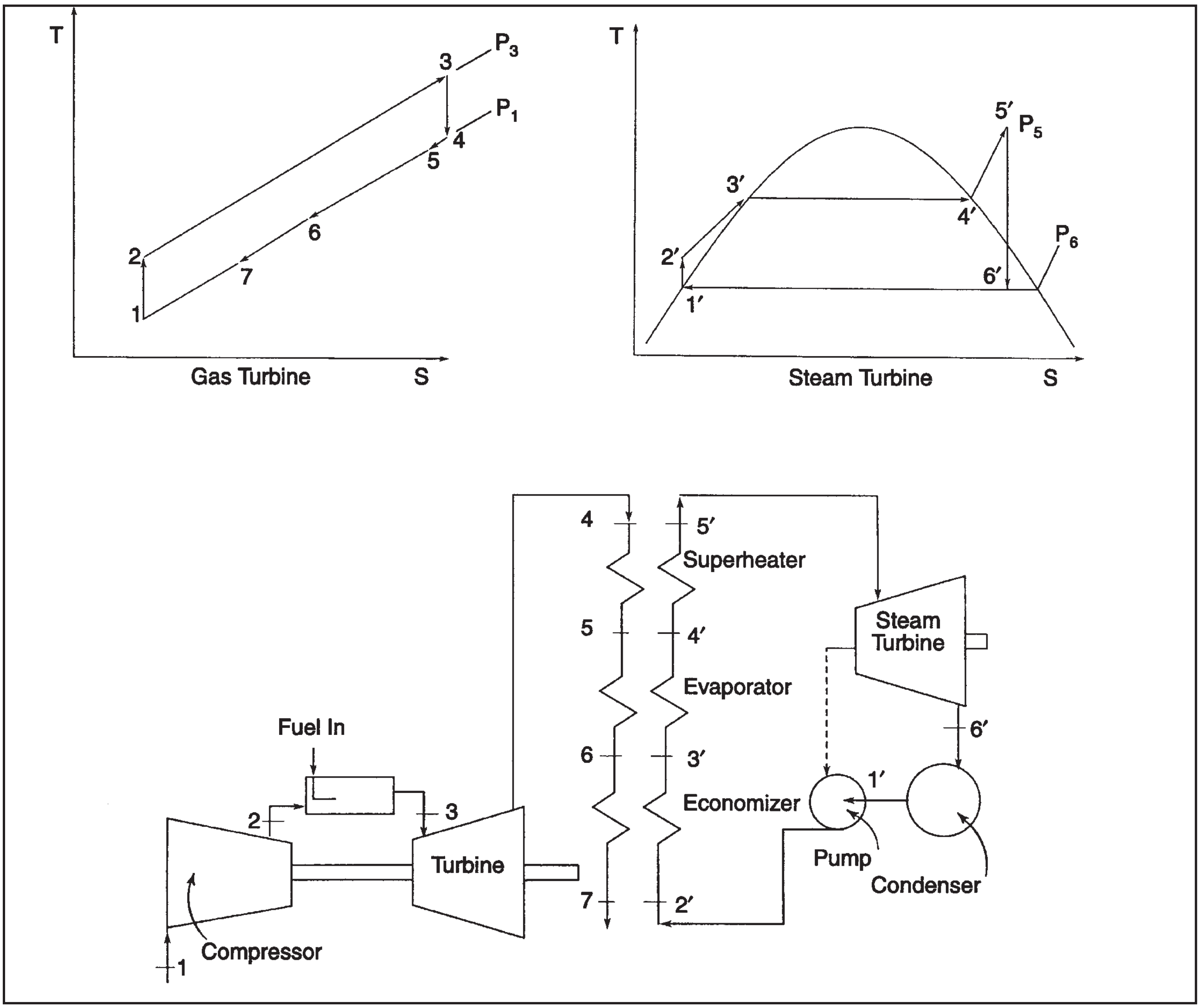
**Coalescers** — These filters are used to remove moisture from the inlet air system.

**High Efficiency Media** — These filters remove smaller dirt particles from the inlet air.

**Marine or Demister** — These filters are used in marine environments to remove both moisture and salt.

**Self-Cleaning** — These filters are composed of a number of

**FIG. 15-26**
**Combined Cycle**



high efficiency media filter "banks." Air is drawn through the media at a low velocity and, at a predetermined pressure drop across the system, a reverse blast of air removes built-up dirt on the filter and lowers the pressure drop. This filter can be used in any environment. It is particularly useful in colder climates where ice build-up is a problem. The reverse blast of air also removes any ice that has built up on the filter.

Another method of eliminating icing problems is to install an anti-icing system. In this system, heated air from the gas generator discharge is introduced through distribution manifolds immediately downstream of the inlet air silencer.

The selection of a filtration system is largely dependent on the site location and operating conditions. Fig. 15-27 suggests filtration for various types of environments.

Since filters do protect the gas turbine and help extend its useful life, some type of filtration is always recommended.

## Acoustics

The noise created by a gas turbine engine is considerable and must be reduced to protect plant personnel and minimize environmental impact. The main sources of noise in a gas turbine installation are the intake, the exhaust, and casing radiated noise.

The noise associated with the intake is characterized as high frequency noise. This type of noise is the loudest and most disturbing to the ear since it is in a range where hearing is most sensitive. The second most objectionable noise is produced by the gas generator and power turbine and is radiated from the casing. Although the exhaust noise contains more energy, the casing noise is more objectionable since it contains more noise in a frequency range where the ear is most sensitive. The exhaust noise is a low frequency noise which is only slightly audible. It does, however, possess a considerable amount of energy which results in a detectable pressure change.

A variety of methods can be used to attenuate noise. The most common are the use of silencers and enclosures. The inlet noise is the first area considered since this is where the largest amount of sound power is produced. Inlet noise is the loudest directly in front of the inlet opening. Consequently, the least expensive method for obtaining some of the required noise reduction is to place an elbow at the inlet. Additional silencing is usually necessary and can be attained by the use of acoustic baffles before the elbow.

Casing radiated noise can be reduced by using an acoustical enclosure over the turbine. If an enclosure is used, it is necessary to provide gas and fire detection and fire extinguishing equipment inside the enclosure.

The last major source of noise to be silenced is the gas turbine exhaust noise. Since most turbines exhaust vertically, there is generally no need for an elbow. However, a silencer with acoustic baffles is needed and the exhaust ducting should be sound insulated.

**FIG. 15-27**
**Gas Turbine Air Filtration**

| Type of Environment | Suggested Filtration |
|---|---|
| Rural Country | High Efficiency Media |
| Urban/Industrial | Inertial & High Efficiency Media |
| Desert | Inertial and Media or Self-Cleaning |
| Tropical | Inertial & Media |
| Arctic | High Efficiency Media with Anti-Icing or Self-Cleaning |
| Offshore | Demisters |

## GAS TURBINE PERFORMANCE

The performance of a gas turbine is usually expressed in terms of power and heat rate. Power is the net power available at the output shaft of the turbine after all losses and power take-offs have been subtracted.

Heat rate is a measure of thermal efficiency or the amount of heat energy (in the form of fuel) which must be input to the gas turbine to produce the output power. Heat rate is usually expressed in terms of Btu/(hp · hr) or Btu/(kW · hr) based on the lower heating value of the fuel. Heat rate and thermal efficiency are related as follows:

$$\text{Thermal efficiency} = \frac{2544}{\text{Heat Rate, } \dfrac{\text{Btu (LHV)}}{\text{hp} \cdot \text{hr}}}$$

$$= \frac{3414}{\text{Heat Rate, } \dfrac{\text{Btu (LHV)}}{\text{kW} \cdot \text{hr}}}$$

Power and heat rate both vary depending on environmental conditions such as ambient air temperature, altitude, barometric pressure, and humidity. Therefore, when performance is stated for a gas turbine, the ambient conditions must be defined. In order to compare different gas turbines, a set of standard conditions known as ISO (International Standards Organization) conditions have been defined as follows:

ISO Conditions: Ambient Temperature = 59°F = 15°C
Altitude = 0 ft (sea level)
Ambient Pressure = 29.92 in. Hg
Relative Humidity = 60%

All gas turbine performance is stated in ISO conditions. To arrive at site rated horsepowers, the ISO conditions must be corrected for the following:

Altitude (Fig. 15-28)
Inlet Losses (Fig. 15-29)
Exhaust Losses (Fig. 15-30)
Temperature (Fig. 15-31)
Humidity (below)

For changing relative humidity, the power output does not change, and the heat rate changes only slightly. For example, for an increase in relative humidity from 60 to 100 percent, a typical correction factor for the heat rate is 1.0016. For a decrease to zero percent, a typical correction factor is 0.9979.

Performance is also affected by other installation variables including power take-offs and type of fuel used. Inlet loss is the pressure drop which occurs as the outside air passes through the inlet filters and plenum. Similarly, exhaust loss is the pressure drop through the exhaust stack, silencers, and heat recovery equipment (if any) which creates a back pressure on the turbine. Power take-offs include any devices such as oil pumps, generators, etc. which are directly driven from the gas turbine output shaft and thus reduce the available output power. Sometimes it is necessary to correct power and/or heat rate for the

The system outlined here is the International System of Units (Systeme International d' Unites), for which the abbreviation SI is being used in all languages.

The SI system, which is becoming universally used, is founded on seven base units, these being:

| | | |
|---|---|---|
| Length | meter | m |
| Mass | kilogram | kg |
| Time | second | s |
| Electric current | ampere | A |
| Thermodynamic temperature | Kelvin | K |
| Luminous intensity | candela | cd |
| Amount of substance | mole | mol |

## POWER

The derived SI unit for power is the Watt (W), this being based on the SI unit of work, energy and quantity of heat – the Joule (J). One Watt (1 W) is equal to one Joule per second (1 J/s). One Watt is a very small unit of power, being equivalent to just 0.00134102 horsepower, so for engine ratings the kilowatt (kW) is used, 1 kW being equal to 1.341 hp and 1 hp being the equivalent of 0.7457 kW. The British unit of horsepower is equal to 1.014 metric horsepower (CV, PS, PK, etc.).

1 kW = 1.341 hp = 1.360 metric hp
1 hp = 0.746 kW = 1.014 metric hp
1 metric hp = 0.735 kW = 0.986 hp

## TORQUE

The derived SI unit for torque (or moment of force) is the Newton meter (Nm), this being based on the SI unit of force — the Newton (N) – and the SI unit of length – the meter (m). One Newton (1 N) is equivalent to 0.2248 pound-force (lbf) or 0.10197 kilogram-force (kgf), and one meter is equal to kilogram force (kgf) and one member is equal to 3.28084 feet (ft), so one Newton meter (1 N m) is equal to 0.737562 pound-force (lbf ft). or 0.101972 kilogram-force meter (kgf m).

1 Nm = 0.738 lbf ft = 0.102 kgf m
1 lbf ft = 1.356 Nm = 0.138 kgf m
1 kgf m = 9.807 Nm = 7.233 lbf ft

## PRESSURE AND STRESS

Although it has been decided that the SI derived unit for pressure and stress should be the Pascal (Pa), this is a very small unit, being the same as one Newton per square meter (1 N/m$^2$), which is only 0.000145 lbf/in$^2$ or 0.0000102 kgf/cm$^2$. So many European engine designers favor the bar as the unit of pressure, one bar being 100,000 Pascal (100 kPa), which is the equivalent of 14,504 lbf/in$^2$ or 1.020 kgf/cm$^2$, so being virtually the same as the currently accepted metric equivalent. On the other hand, for engine performance purposes, the millibar seems to be favored to indicate barometric pressure, this unit being one thousandth of a bar. Then again, there is a school that favors the kiloNewton per square meter (kN/m$^2$), this being the same as a kilopascal, and equal to 0.145 lbf/in$^2$ or 0.0102 kgf/cm$^2$.

1 bar = 14.5 lbf/in$^2$ = 1.0197 kgf/cm$^2$
1 lbf/in$^2$ = 0.069 bar
1 kgf/cm$^2$ = 0.98 bar

The American Society of Mechanical Engineers in 1973 published its Performance Test Codes for Reciprocating Internal Combustion engines. Known as PTC 17, this code is intended for tests of all types of reciprocating internal combustion engines for determining power output and fuel consumption. In its Section 2, Description and Definition of Terms, both the FPS and corresponding SI units of meas-urements are given.

## SPECIFIC CONSUMPTION

Fuel consumption measurements will be based on the currently accepted unit, the gram (g), and the Kilowatt Hour (kWh). Also adopted is heat units/power units so that energy consumption of an internal combustion engine referred to net power output, mechanical, is based on low unsaturated heat value of the fuel whether liquid or gaseous type. Thus the SI unit of measurement for net specific energy consumption is expressed: g/kWh.

1 g/kWh = 0.001644 lb/hph =
0.746 g/hph = 0.736 g/metric hph
1 lb/hph = 608.3 g/kWh
1 g/hph = 1.341 k/kWh
1 g/metric hph = 1.36 g/kWh

## HEAT RATE

Heat Rate is a product of Lower Heating Value (LHV) of Fuel (measured in Btu/lb or kJ/g for liquid fuel and Btu/ft$^3$ or kJ/m$^3$ for gas fuel) multiplied times (sfc) specific fuel consumption (measured in lb/hph or g/kWh).

**For Liquid Fuel**
Heat Rate (Btu/hph) = LVH (Btu/lb) X sfc (lb/hph)

**For Gaseous Fuel**
Heat Rate (Btu/hph) = LVH (Btu/ft$^3$) X sfc (ft$^3$/hph)

**To convert these units to SI units:**
Btu/hph X 1.414 = kJ/kWh
Or
Btu/kWh X 1.055 = kJ/kWh

## LUBRICATING-OIL CONSUMPTION

Although the metric liter is not officially an SI unit, its use will continue to be permitted, so measurement of lube-oil consumption will be quoted in liters per hour (liters/h).

1 liter/h = 0.22 Imp gal/h
1 Imp gal/h = 4.546 liters/h

## TEMPERATURES

The SI unit of temperature is Kelvin (K), and the character is used without the degree symbol (°) normally employed with other scales of temperature. A temperature of zero degree Kelvin is equivalent to a temperature of -273.15°C on the Celsius (centigrade) scale. The Kelvin unit is identical in interval to the Celsius unit, so direct conversions can be made by adding or subtracting 273. Use of Celsius is still permitted.

0 K = 273°C; absolute zero K
1°C = 273 K

## WEIGHTS AND LINEAR DIMENSIONS

For indications of "weight" the original metric kilogram (kg) will continue to be used as the unit of mass, but it is important to note that the kilogram will no longer apply for force, for which the SI unit is the Newton (N), which is a kilogram meter per second squared. The Newton is that force which, when applied to a body having a mass of one kilogram, gives it an acceleration of one meter per second squared.

"Weight" in itself will no longer apply, since this is an ambiguous term, so the kilogram in effect should only be used as the unit of mass. Undoubtedly, though, it will continue to be common parlance to use the word "weight" when referring to the mass of an object.

The base SI unit for linear dimensions will be the meter, with a wide range of multiples and sub-multiples ranging from exa ($10^{18}$) to atto ($10^{-18}$): A kilometer is a meter x $10^3$, for example, while a millimeter is a meter x $10^{-3}$.

To give an idea of how currently used units convert to SI units, the tables below give examples.

**KILOWATTS (kW) TO HORSEPOWER (hp)**
**(1 Kw = 1.34102 hp)**

| kW | hp | kW | hp | kW | hp | kW | hp | kW | hp |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.341 | 21 | 28.161 | 41 | 54.982 | 61 | 81.802 | 81 | 108.623 |
| 2 | 2.682 | 22 | 29.502 | 42 | 56.323 | 62 | 83.143 | 82 | 109.964 |
| 3 | 4.023 | 23 | 30.843 | 43 | 57.664 | 63 | 84.484 | 83 | 111.305 |
| 4 | 5.364 | 24 | 32.184 | 44 | 59.005 | 64 | 85.825 | 84 | 112.646 |
| 5 | 6.705 | 25 | 33.526 | 45 | 60.346 | 65 | 87.166 | 85 | 113.987 |
| 6 | 8.046 | 26 | 34.867 | 46 | 61.687 | 66 | 88.507 | 86 | 115.328 |
| 7 | 9.387 | 27 | 36.208 | 47 | 63.028 | 67 | 89.848 | 87 | 116.669 |
| 8 | 10.728 | 28 | 37.549 | 48 | 64.369 | 68 | 91.189 | 88 | 118.010 |
| 9 | 12.069 | 29 | 38.890 | 49 | 65.710 | 69 | 92.530 | 89 | 119.351 |
| 10 | 13.410 | 30 | 40.231 | 50 | 67.051 | 70 | 93.871 | 90 | 120.692 |
| 11 | 14.751 | 31 | 41.572 | 51 | 68.392 | 71 | 95.212 | 91 | 122.033 |
| 12 | 16.092 | 32 | 42.913 | 52 | 69.733 | 72 | 96.553 | 92 | 123.374 |
| 13 | 17.433 | 33 | 44.254 | 53 | 71.074 | 73 | 97.894 | 93 | 124.715 |
| 14 | 18.774 | 34 | 45.595 | 54 | 72.415 | 74 | 99.235 | 94 | 126.056 |
| 15 | 20.115 | 35 | 46.936 | 55 | 73.756 | 75 | 100.577 | 95 | 127.397 |
| 16 | 21.456 | 36 | 48.277 | 56 | 75.097 | 76 | 101.918 | 96 | 128.738 |
| 17 | 22.797 | 37 | 49.618 | 57 | 76.438 | 77 | 103.259 | 97 | 130.079 |
| 18 | 24.138 | 38 | 50.959 | 58 | 77.779 | 78 | 104.600 | 98 | 131.420 |
| 19 | 25.479 | 39 | 52.300 | 59 | 79.120 | 79 | 105.941 | 99 | 132.761 |
| 20 | 26.820 | 40 | 53.641 | 60 | 80.461 | 80 | 107.282 | 100 | 134.102 |

**POUNDS FORCE FEET (lbf ft) TO NEWTON METERS (Nm)**
**(1 lbf ft = 1.35582 Nm)**

| lbf ft | Nm | lbf ft | Nm | lbf ft | Nm | lbf ft | Nm | lbf ft | Nm |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.356 | 21 | 28.472 | 41 | 55.589 | 61 | 82.705 | 81 | 109.821 |
| 2 | 2.712 | 22 | 29.828 | 42 | 56.944 | 62 | 84.061 | 82 | 111.177 |
| 3 | 4.067 | 23 | 31.184 | 43 | 58.300 | 63 | 85.417 | 83 | 112.533 |
| 4 | 5.423 | 24 | 32.540 | 44 | 59.656 | 64 | 86.772 | 84 | 113.889 |
| 5 | 6.779 | 25 | 33.896 | 45 | 61.012 | 65 | 88.128 | 85 | 115.245 |
| 6 | 8.135 | 26 | 35.251 | 46 | 62.368 | 66 | 89.484 | 86 | 116.601 |
| 7 | 9.491 | 27 | 36.607 | 47 | 63.724 | 67 | 90.840 | 87 | 117.956 |
| 8 | 10.847 | 28 | 37.963 | 48 | 65.079 | 68 | 92.196 | 88 | 119.312 |
| 9 | 12.202 | 29 | 39.319 | 49 | 66.435 | 69 | 93.552 | 89 | 120.668 |
| 10 | 13.558 | 30 | 40.675 | 50 | 67.791 | 70 | 94.907 | 90 | 122.024 |
| 11 | 14.914 | 31 | 42.030 | 51 | 69.147 | 71 | 96.263 | 91 | 123.380 |
| 12 | 16.270 | 32 | 43.386 | 52 | 70.503 | 72 | 97.619 | 92 | 124.715 |
| 13 | 17.626 | 33 | 44.742 | 53 | 71.808 | 73 | 98.975 | 93 | 126.001 |
| 14 | 18.981 | 34 | 46.098 | 54 | 73.214 | 74 | 100.331 | 94 | 127.447 |
| 15 | 20.337 | 35 | 47.454 | 55 | 74.570 | 75 | 101.687 | 95 | 128.803 |
| 16 | 21.693 | 36 | 48.810 | 56 | 75.926 | 76 | 103.042 | 96 | 130.159 |
| 17 | 23.049 | 37 | 50.165 | 57 | 77.282 | 77 | 104.398 | 97 | 131.515 |
| 18 | 24.405 | 38 | 51.521 | 58 | 78.638 | 78 | 105.754 | 98 | 132.870 |
| 19 | 25.761 | 39 | 52.877 | 59 | 79.993 | 79 | 107.110 | 99 | 134.226 |
| 20 | 27.116 | 40 | 54.233 | 60 | 81.349 | 80 | 108.466 | 100 | 135.582 |

These tables are reproduced from the booklet "Vehicle Metrics" published by Transport and Distribution Press Ltd., 118 Ewell Road, Surbiton, Surry, KT6 6HA England.

type of fuel used in the gas turbine. The turbine manufacturer's performance brochure should be consulted for necessary corrections.

The following example shows the method of calculating performance for a gas turbine at site conditions using data typically supplied in the manufacturer's performance brochure.

**Example 15-3** — Calculate maximum available site power and heat rate for the example gas turbine at the following conditions:

| | | |
|---|---|---|
| Turbine ISO Horsepower | = | 27,500 |
| Turbine ISO Heat Rate | = | 7,090 Btu/(hp · hr) |
| Ambient Temperature | = | 80°F |
| Altitude | = | 1000 ft (above sea level) |
| Inlet Pressure Drop | = | 4 in. $H_2O$ |
| Exhaust Pressure Drop | = | 2 in. $H_2O$ |
| Relative Humidity | = | 60% |
| Fuel | = | Natural Gas |

**Solution Steps**

Find the power altitude correction factor from Fig. 15-28. For 1000 ft altitude, the correction factor is 0.965.

Find power inlet loss correction factor from Fig. 15-29. For 4 inches of water, the correction factor is 0.984.

Find power exhaust loss correction factor from Fig. 15-30. For 2 inches of water, the correction factor is 0.9965.

Find the power ambient temperature correction factor from Fig. 15-31. For 80°F the correction factor is 0.915.

Since relative humidity is 60% and fuel is natural gas, no corrections are required.

Calculate the maximum available site power by multiplying maximum-no-loss power by each of the correction factors.

Power (site) = power (0.965) (0.984) (0.9965) (0.915)
Power (site) = 27,500 (0.965) (0.984) (0.9965) (0.915)
Power (site) = 23,800 hp

For the heat rate find the inlet loss correction factor, exhaust loss correction factor, and ambient temperature correction factor from Figs. 15-29, 15-30, and 15-31, respectively. (Note: Heat rate is not affected by altitude.)

Inlet loss factor = 1.0065
Exhaust loss factor = 1.003
Temperature factor = 1.03

Calculate site heat rate by multiplying no-loss heat rate by the correction factors.

Heat rate (site) = (Heat rate) (1.0065) (1.003) (1.03)
Heat rate (site) = [7090 Btu/(hp · hr)](1.0065)(1.003)(1.03)
Heat rate (site) = 7370 Btu/(hp · hr)

The above calculation procedures may vary slightly with different manufacturers but will follow the same principles.

Basic specifications for some of the commonly used gas turbine engines are shown in Fig. 15-32.

**FIG. 15-28**
**Altitude Correction Factor**

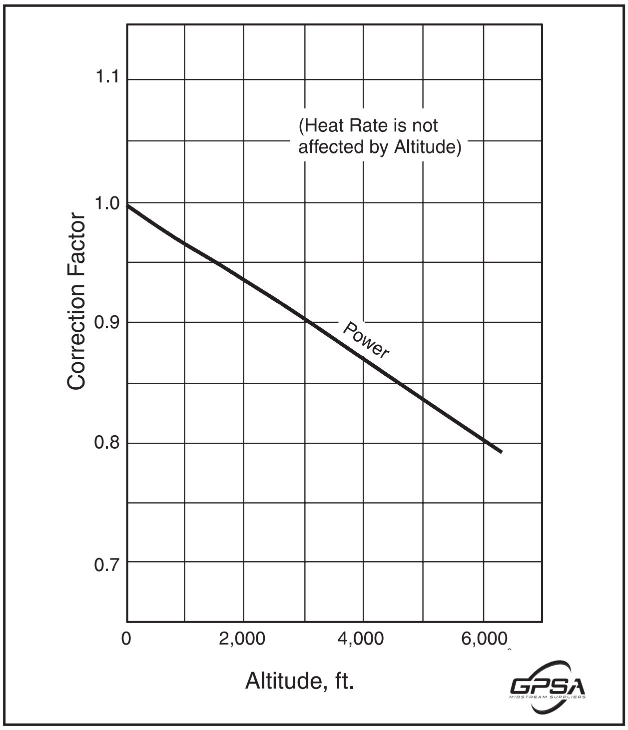



**FIG. 15-29**
**Inlet Loss Correction Factor**

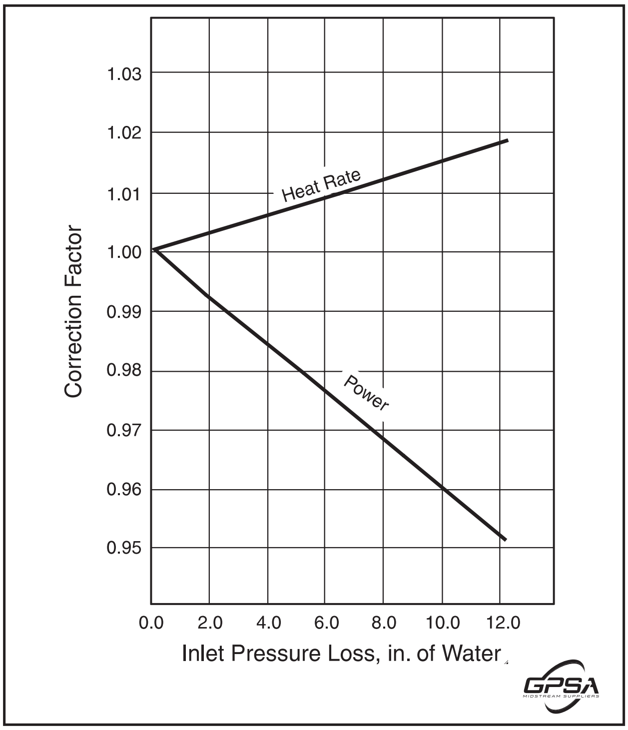


FIG. 15-30
Exhaust Loss Correction Factor

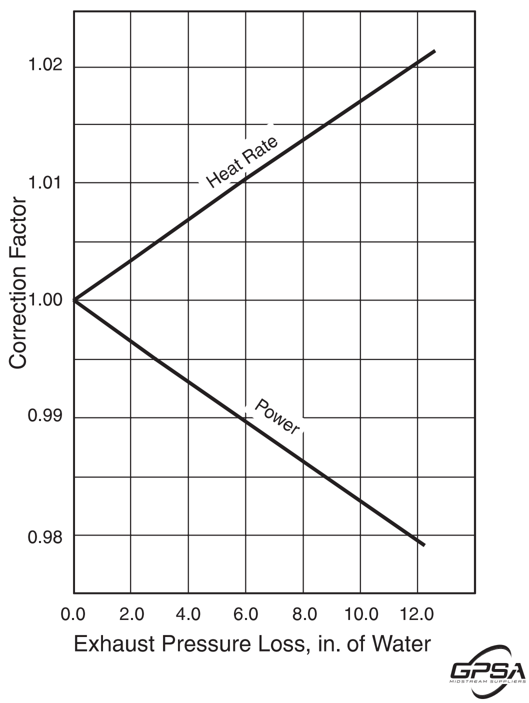
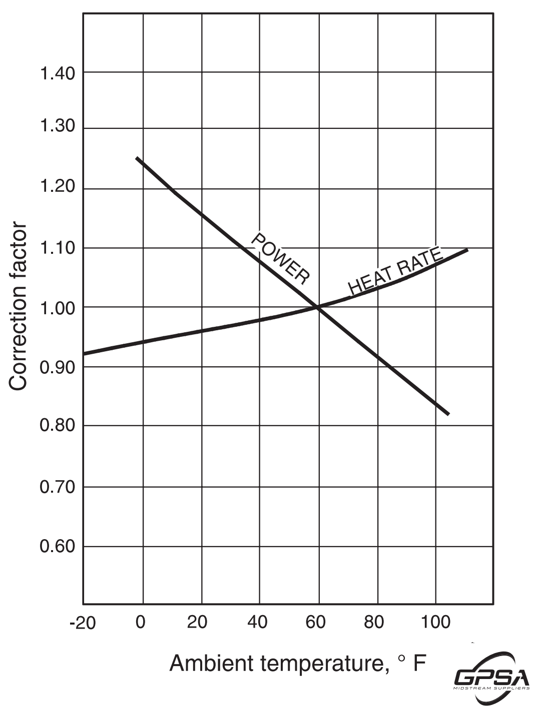



FIG. 15-31
Ambient Temperature Correction Factor




## Gas Turbine Emissions

The gas turbine, in general, is a low emitter of exhaust gas pollutants relative to other heat engines in similar service. This is because the fuel is burned with ample excess air to ensure complete combustion at all but minimum load conditions. It is unique in its ability to burn a wide variety of fuels making each application unique in terms of exhaust emissions. However, gas turbine engine emissions recently have become a major factor in the design, selection, and operation of the unit. Various federal, state, and local authorities have issued standards and codes to control pollution of the atmosphere.

Carbon monoxide (CO) emissions occur because of incomplete combustion of fuel carbon. CO emissions for distillate and other liquid fuels are generally higher than for natural gas.

Unburned hydrocarbons (UHC) are formed by the incomplete combustion of fuel. Like CO emissions, they are directly related to combustion efficiency. However, because most gas turbine units on the market today have good combustor designs, the CO and UHC emissions are of secondary importance to $NO_x$ emissions.

Sulfur oxides ($SO_x$) exhausted from gas turbines are a direct function of sulfur content in the fuel.

The high temperature and oxygen content during combustion tends to favor the formation of $SO_3$ and $SO_2$ at equilibrium. Sulfur oxide emissions from pipeline natural gas are virtually zero while wellhead gases, process gases, coal gases, and other fuels may contain significant quantities of sulfur in the form of $H_2S$.

Gas turbine particulate emissions are influenced by the fuel properties and combustion conditions. Particulates generally refer to visible smoke, ash, ambient non-combustibles, and products of erosion and corrosion in the hot gas path. Particulate and smoke emissions are usually small when burning natural gas, but are a significant consideration when operating on liquid fuels.

Of the exhaust components the most significant are the oxides of nitrogen ($NO_x$). The amount of $NO_x$ produced is a function of the fuel burned, firing temperature, compressor discharge temperature, and residence time in the combustion zone. Since the trend towards high turbine efficiencies leads to higher pressure ratios and firing temperatures, the emission rates of $NO_x$ are higher for these units.

Nitrogen oxides are categorized into two areas according to the mechanism of formation. $NO_x$ formed by oxidation of free nitrogen in the combustion air or fuel is called "thermal $NO_x$," while that due to oxidation of organically bound nitrogen in the fuel is referred to as "organic $NO_x$." As implied by the name, thermal $NO_x$ are mainly a function of the stoichiometric flame temperature. The formation of thermal $NO_x$ is on the order of parts per million (by volume) or ppmv; however, the conversion of organic $NO_x$ is virtually 100%. Efforts to reduce thermal $NO_x$ by reducing flame temperatures have little effect on, and actually may increase, organic $NO_x$.

UHC emissions can be reduced by proper combustor design for maximum efficiency. Sulfur oxides can be eliminated by removing sulfur compounds from the fuel. Similarly, particulates can be minimized by appropriate fuel treatment. However, reduction of $NO_x$ formation also produces increased inefficiency.

Two general approaches are used for $NO_x$ reduction:

- The use of an inert heat sink such as water or steam injection.

**FIG. 15-32**

**2011 Basic Specifications — Gas Turbine Engines (Mechanical Drive)**

| Model | Power Rating (ISO Rating) hp | Heat Rate (LVH) Btu/hp-hr | Pressure Ratio | Power Shaft RPM | At ISO RATING CONDITIONS | | |
|---|---|---|---|---|---|---|---|
| | | | | | Turbine Inlet Temp. °F | Exhaust Flow lb/s | Exhaust Temp °F |
| **Dresser- Rand** | | | | | | | |
| VECTRA 30G | 31,469 | 6816 | 17.9 | 6510 | 1530 | 149.7 | 1017 |
| VECTRA 40G | 42,102 | 6347 | 22.4 | 6510 | 1521 | 190.2 | 979 |
| VECTRA 40G4 | 45,902 | 6316 | 23.6 | 6510 | 1571 | 198.4 | 1006 |
| DR-63G PC | 59,436 | 6042 | 27.9 | 3780 | 1578 | 280.0 | 855 |
| DR-63G PG | 66,822 | 6054 | 29.7 | 3930 | 1666 | 259.3 | 907 |
| **GE Oil & Gas** | | | | | | | |
| GE10-2 DLE | 15907.2 | 7762.2 | 15.8 | 7900 | | 103.6 | 912 |
| GE10-2 | 16288.1 | 7620.9 | 15.6 | 7900 | | 103.6 | 901 |
| PGT16 | 19143.1 | 7042.7 | 20.1 | 7900 | | 103.8 | 928 |
| PGT20 SAC | 24300.4 | 6974.1 | 19.7 | 6500 | | 138.0 | 895 |
| PGT20 DLE | 24926.9 | 6984.7 | 19.8 | 6500 | | 137.3 | 915 |
| PGT25 DLE | 31194.9 | 6793.2 | 17.9 | 6500 | | 151.0 | 983 |
| PGT25 SAC | 31205.6 | 6756.4 | 17.9 | 6500 | | 151.9 | 971 |
| MS5002C | 37950.9 | 8700.9 | 8.8 | 4670 | | 274.0 | 963 |
| MS5002C POWER CRYSTAL | 39520 | 8714 | 9.1 | 4670 | | 270.0 | 1004 |
| PGT25+DLE | 41673.6 | 6207.2 | 21.5 | 6100 | | 184.7 | 934 |
| PGT25+SAC | 42070.7 | 6187.4 | 21.5 | 6100 | | 185.8 | 932 |
| MS5002E | 42912.7 | 7052.6 | 17 | 5714 | | 225.5 | 947 |
| MS5002D | 43717.3 | 8411.1 | 10.8 | 4670 | | 311.7 | 948 |
| PGT25+G4 DLE | 45164.3 | 6207.2 | 23 | 6100 | | 197.3 | 955 |
| PGT25+G4 SAC | 45492.8 | 6208.7 | 23 | 6100 | | 198.4 | 954 |
| MS5002D POWER CRYSTAL | 45553 | 8413 | 10.4 | 4670 | | 308.0 | 993 |
| LM6000 PD | 58809.2 | 5985.3 | 28.3 | 3600 | | 274.9 | 851 |
| LM6000 PF | 58809.2 | 5985.3 | 28.3 | 3600 | | 274.9 | 851 |
| MS6001B | 58955.4 | 8140.4 | 12.3 | 5160 | | 322.3 | 1016 |
| LM6000 PC SAC FIXED IGV | 59384.5 | 5971.9 | 27.9 | 3600 | | 276.9 | 850 |
| LM6000 PC SAC OPEN IGV | 59558.8 | 5976.1 | 28.2 | 3600 | | 278.9 | 846 |
| LM6000 PC SAC VARIABLE IGV | 59663.4 | 5967.6 | 28.1 | 3600 | | 278.2 | 849 |
| MS7001EA | 121362 | 7584.1 | 12.9 | 3600 | | 662.0 | 1011 |
| LMS100 | 134370 | 5767.6 | 40 | 3600 | | 456.1 | 783 |
| MS9001E | 175272 | 7357.9 | 12.8 | 3000 | | 926.6 | 1001 |
| **MAN Diesel & Turbo SE** | | | | | | | |
| THM 1203A | 8046 | 10870 | 7.8 | 7800 | 1724 | 78.0 | 959 |
| THM 1304-10R | 12606 | 7011 | 10 | 9030 | 1787 | 100.0 | |
| THM 1304-10 | 13008 | 8715 | 10 | 9030 | 1787 | 100.0 | 932 |
| THM 1304-11 | 15019 | 8206 | 10.8 | 9030 | 1823 | 108.0 | 941 |

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**FIG. 15-32 (Cont'd)**

**2011 Basic Specifications — Gas Turbine Engines (Mechanical Drive)**

| Model | Power Rating (ISO Rating) hp | Heat Rate (LVH) Btu/hp-hr | Pressure Ratio | Power Shaft RPM | At ISO RATING CONDITIONS | | |
|---|---|---|---|---|---|---|---|
| | | | | | Turbine Inlet Temp. °F | Exhaust Flow lb/s | Exhaust Temp °F |
| THM 1304-12 | 16226 | 8001 | 11 | 9030 | | 108.0 | 959 |
| THM 1304-14 | 17701 | 7881 | 11 | 9030 | | 108.0 | 1013 |
| FT8 | 34690 | 6615 | 19.5 | 5775 | | 188.5 | 856 |
| **MTU Friedrichshafen GmbH** | | | | | | | |
| LM2500-PE | 30180 | 6784 | 17.9 | 3600 | | 152.0 | 975 |
| LM2500-PH | 36210 | 5986 | 19.3 | 3600 | | 167.0 | 932 |
| LM2500+(PK) | 41840 | 6440 | 22 | 3600 | | 192.0 | 959 |
| LM6000 | 60346 | | | | | | |
| **Rolls-Royce** | | | | | | | |
| 501-KC5 | 5500 | 8495 | 9.4 | | | 34.2 | 1060 |
| 501-KC7 | 7400 | 7902 | 13.5 | | | 46.2 | 968 |
| Avon2648 | 21923 | 8323 | 9.6 | | | 179.0 | 799 |
| Avon2656 | 22807 | 8022 | 9.6 | | | 179.0 | 788 |
| RB211 - G62 | 39600 | 6705 | 20.8 | | | 209.0 | 916 |
| RB211 - GT62 | 41450 | 6585 | 21.7 | | | 214.0 | 918 |
| RB211 - GT61 | 44650 | 6285 | 21.7 | | | 210.0 | |
| RB211 - H63 | 50848 | 6134 | 23 | | | 235.0 | |
| RB211 - H63 | 59005 | 6247 | 25.1 | | | 254.6 | |
| Trent 60 DLE | 70418 | 5939 | 34 | | | 337.8 | 824 |
| Trent 60 WLE | 79120 | 6074 | 35.3 | | | 358.6 | |
| **Siemens AG Energy Sector** | | | | | | | |
| SGT-100 | 7640 | 7738 | 14.9 | 13650 | | 43.4 | 1009 |
| SGT-200 | 10300 | 7616 | 12.6 | 11525 | | 64.9 | 919 |
| SGT-300 | 11000 | 7738 | 13.3 | | | 63.9 | 928 |
| SGT-400 | 18000 | 7028 | 16.8 | 10000 | | 86.8 | 1031 |
| SGT-500 | 26177 | 7373 | 13 | | | 215.9 | 696 |
| SGT-600 | 34100 | 7250 | 14 | 8085 | | 177.0 | 1009 |
| SGT-700 | 42960 | 6805 | 18 | 6930 | | 208.0 | 982 |
| SGT-750 | 49765 | 6362 | 23.8 | 6405 | | 249.8 | 864 |
| **Solar Turbines Incorporated** | | | | | | | |
| Saturn 20 | 1590 | 10370 | 6.7 | 22300 | | 14.3 | 968 |
| Centaur 40 | 4700 | 9125 | 10.3 | 15500 | | 41.8 | 833 |
| Centaur 50 | 6130 | 8500 | 10.3 | 16500 | | 41.5 | 959 |
| Taurus 60 | 7700 | 7965 | 11.5 | 13950 | | 47.7 | 950 |
| Taurus 70 | 10310 | 7310 | 16.5 | 11400 | | 58.6 | 923 |
| Mars 90 | 13220 | 7655 | 16.3 | 9400 | | 88.5 | 869 |
| Mars 100 | 16000 | 7370 | 17.7 | 9500 | | 93.1 | 905 |
| Titan 130 | 20500 | 7025 | 16.1 | 8500 | | 110.3 | 941 |
| Titan 250 | 30000 | 6360 | 24.1 | 7000 | | 150.4 | 869 |

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# COMPRESSOR HORSEPOWER SELECTION CHART

(Brake Horsepower Per Million Cu. Ft.)

| SUCTION PRESSURE | DISCHARGE PRESSURE (PSIG) 25 | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 550 | 600 | 650 | 700 | 750 | 800 | 850 | 900 | 950 | 1000 | 1050 | 1100 | 1150 | 1200 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 65 | 99 | 128 | 144 | 156 | 168 | 178 | 187 | 203 | 218 | 233 | 233 | 241 | 248 | 254 | 260 | 266 | 272 | 277 | 282 | 286 | 291 | 295 | 299 | 303 | 307 | 311 | 315 | 3 STAGE |
| 10 | 35 | 63 | 85 | 104 | 121 | 131 | 140 | 149 | 163 | 175 | 186 | 196 | 205 | 214 | 223 | 231 | 228 | 233 | 237 | 242 | 245 | 250 | 253 | 257 | 260 | 264 | 267 | 270 | |
| 20 | | 43 | 62 | 78 | 92 | 106 | 118 | 126 | 139 | 151 | 160 | 170 | 178 | 186 | 193 | 199 | 206 | 212 | 218 | 225 | 231 | 226 | 229 | 232 | 236 | 239 | 242 | 245 | |
| 30 | | 29 | 47 | 62 | 74 | 85 | 96 | 107 | 123 | 133 | 143 | 152 | 159 | 167 | 173 | 179 | 185 | 191 | 196 | 201 | 206 | 211 | 216 | 221 | 226 | 230 | 224 | 227 | |
| 40 | | | 36 | 50 | 61 | 72 | 81 | 90 | 107 | 121 | 130 | 138 | 145 | 152 | 158 | 164 | 170 | 175 | 180 | 185 | 190 | 194 | 198 | 202 | 206 | 210 | 214 | 218 | 2 STAGE |
| 50 | | | 26 | 41 | 52 | 61 | 70 | 78 | 93 | 106 | 119 | 127 | 134 | 141 | 147 | 153 | 158 | 163 | 168 | 173 | 177 | 181 | 185 | 189 | 193 | 196 | 200 | 203 | |
| 60 | | | | 32 | 44 | 53 | 61 | 69 | 83 | 95 | 108 | 118 | 125 | 131 | 137 | 143 | 148 | 153 | 158 | 162 | 166 | 170 | 174 | 178 | 182 | 185 | 188 | 192 | |
| 70 | | | | 25 | 37 | 46 | 54 | 61 | 74 | 86 | 97 | 109 | 117 | 123 | 129 | 135 | 140 | 145 | 149 | 153 | 157 | 161 | 165 | 169 | 172 | 176 | 179 | 182 | |
| 80 | | | | | 30 | 40 | 47 | 54 | 67 | 78 | 89 | 98 | 109 | 117 | 122 | 127 | 132 | 137 | 142 | 146 | 150 | 153 | 157 | 161 | 164 | 167 | 171 | 174 | |
| 90 | | | | | 24 | 34 | 42 | 49 | 61 | 72 | 81 | 91 | 100 | 109 | 116 | 121 | 126 | 131 | 135 | 139 | 143 | 147 | 150 | 154 | 157 | 160 | 163 | 166 | |
| 100 | | | | | | 28 | 37 | 44 | 55 | 66 | 75 | 84 | 92 | 100 | 109 | 116 | 120 | 125 | 129 | 133 | 137 | 141 | 144 | 148 | 151 | 154 | 157 | 160 | |
| 125 | | | | | | | 25 | 32 | 44 | 54 | 63 | 71 | 78 | 85 | 92 | 99 | 106 | 113 | 117 | 121 | 124 | 128 | 131 | 134 | 137 | 140 | 143 | 146 | |
| 150 | | | | | | | | 22 | 35 | 45 | 53 | 60 | 67 | 74 | 80 | 86 | 92 | 98 | 103 | 110 | 114 | 118 | 121 | 124 | 127 | 130 | 133 | 135 | |
| 175 | | | | | | | | | 27 | 37 | 45 | 52 | 57 | 60 | 71 | 76 | 82 | 87 | 92 | 97 | 102 | 107 | 112 | 115 | 118 | 121 | 123 | 126 | |
| 200 | | | | | | | | | | 30 | 38 | 45 | 52 | 58 | 63 | 68 | 73 | 78 | 83 | 88 | 92 | 96 | 101 | 105 | 110 | 113 | 116 | 119 | |
| 250 | | | | | | | | | | | 26 | 33 | 40 | 46 | 51 | 56 | 60 | 65 | 69 | 73 | 77 | 81 | 85 | 88 | 92 | 95 | 99 | 102 | 1 STAGE |
| 300 | | | | | | | | | | | | 23 | 30 | 36 | 41 | 46 | 50 | 54 | 58 | 62 | 66 | 69 | 73 | 76 | 79 | 83 | 86 | 89 | |
| 350 | | | | | | | | | | | | | 21 | 27 | 33 | 38 | 42 | 46 | 50 | 53 | 57 | 60 | 63 | 67 | 70 | 73 | 75 | 78 | |
| 400 | | | | | | | | | | | | | | | 25 | 30 | 35 | 39 | 43 | 46 | 50 | 53 | 56 | 59 | 60 | 64 | 67 | 70 | |
| 450 | | | | | | | | | | | | | | | | 23 | 28 | 32 | 36 | 40 | 43 | 46 | 49 | 52 | 55 | 58 | 60 | 63 | |
| 500 | | | | | | | | | | | | | | | | | 22 | 26 | 30 | 34 | 38 | 41 | 44 | 46 | 49 | 52 | 54 | 57 | |
| 550 | | | | | | | | | | | | | | | | | | 20 | 25 | 20 | 32 | 36 | 39 | 41 | 44 | 46 | 49 | 51 | |
| 600 | | | | | | | | | | | | | | | | | | | | 23 | 27 | 30 | 34 | 37 | 39 | 42 | 44 | 46 | |
| 650 | | | | | | | | | | | | | | | | | | | | | 22 | 26 | 29 | 32 | 35 | 38 | 40 | 42 | |
| 700 | | | | | | | | | | | | | | | | | | | | | | 22 | 25 | 28 | 30 | 33 | 36 | 38 | |
| 750 | | | | | | | | | | | | | | | | | | | | | | | 20 | 24 | 27 | 29 | 32 | 34 | |

**NOTE: 1** MMSCFD MEASURED 14.7 AND 60°F NOT CORRECTED FOR COMPRESSIBILITY **2** "N"=1.26 **3** SUCTION TEMPERATURE 100°F **4** NATURAL GAS

For 900 rpm loads, synchronous motors should be considered at 1000 hp and above.

Motor requirements below 500 hp in the 900 to 1800 rpm speed range are normally handled by standard induction motors.

**514 to 720 rpm** — Synchronous motors are often selected above 1 hp per rpm, such as 800 hp at 720 rpm, 700 hp at 600 rpm, and 600 hp at 514 rpm.

**Below 514 rpm** — The synchronous motor should be considered for sizes down to 200 hp because of higher efficiency, improved power factor, and possible lower cost. At high voltages (4 kV and above), the synchronous motor becomes more economical at even lower horsepowers.

## Motor Voltage

The proper selection of voltage for a given motor drive can vary from a routine procedure to a complex study requiring a complete electrical system analysis. In many instances the in-plant distribution system is well established at a particular voltage, say 2300 (2.3 kV). The new machine may be small compared to available system capacity on the 2.3 kV bus so no problem is involved in purchasing a standard motor of that voltage. In more complicated cases additional substation capacity may be necessary to accommodate the new machine. However, when very large units are to be added, many factors must be considered. A new distribution voltage level, a new transmission line, or a higher voltage transmission from the electric utility might be necessary.

# MOTOR ENCLOSURES

Motor enclosure selection should be predicated upon the environmental conditions under which the motor must operate. Directly related to this is the amount of maintenance required to provide long-term reliability and motor life. In general, the more open the enclosure is to the atmosphere, the lower the first cost of the machine but the higher the maintenance costs that may be necessary. Enclosures frequently used in a-c motors are listed below.

## Drip-Proof

These are generally used only indoors or in enclosed spaces not exposed to severe environmental conditions. Maintenance requirements will depend upon general cleanliness of the location and any chemical contaminants in the area.

## Weather-Protected Type I

This is the least costly outdoor machine. It is essentially a drip-proof guarded motor with heaters and outdoor bearing seals and is very susceptible to weather and atmospheric contamination. Considerable maintenance may be required to ensure satisfactory winding and bearing life.

## Weather-Protected Type II

This is the more commonly used outdoor enclosure. It is more expensive than the WP-I but minimizes the entrance of water and dirt. Maintenance is less than for WP-I types. Chemical contaminants in gaseous form may be carried into a WP-II machine with the ventilating air and attack parts that are vulnerable to them.

**FIG. 15-33**

**Energy Evaluation Chart**
**NEMA Frame Size Motors, Induction**

| HP | Approx. Full Load RPM | Amperes Based on 460V | | Efficiency in Percentage at Full Load | |
|---|---|---|---|---|---|
| | | Standard Efficiency | High Efficiency | Standard Efficiency | High Efficiency |
| 1 | 1,800 | 1.9 | 1.5 | 72.0 | 84.0 |
| | 1,200 | 2.0 | 2.0 | 68.0 | 78.5 |
| 1-1/2 | 1,800 | 2.5 | 2.2 | 75.5 | 84.0 |
| | 1,200 | 2.8 | 2.6 | 72.0 | 84.0 |
| 2 | 1,800 | 2.9 | 3.0 | 75.5 | 84.0 |
| | 1,200 | 3.5 | 3.2 | 75.5 | 84.0 |
| 3 | 1,800 | 4.7 | 3.9 | 75.5 | 87.5 |
| | 1,200 | 5.1 | 4.8 | 75.5 | 86.5 |
| 5 | 1,800 | 7.1 | 6.3 | 78.5 | 89.5 |
| | 1,200 | 7.6 | 7.4 | 78.5 | 87.5 |
| 7-1/2 | 1,800 | 9.7 | 9.4 | 84.0 | 90.2 |
| | 1,200 | 10.5 | 9.9 | 81.5 | 89.5 |
| 10 | 1,800 | 12.7 | 12.4 | 86.5 | 91.0 |
| | 1,200 | 13.4 | 13.9 | 84.0 | 89.5 |
| 15 | 1,800 | 18.8 | 18.6 | 86.5 | 91.0 |
| | 1,200 | 19.7 | 19.0 | 84.0 | 89.5 |
| 20 | 1,800 | 24.4 | 25.0 | 86.5 | 91.0 |
| | 1,200 | 25.0 | 24.9 | 86.5 | 90.2 |
| 25 | 1,800 | 31.2 | 29.5 | 88.5 | 91.7 |
| | 1,200 | 29.2 | 29.1 | 88.5 | 91.0 |
| 30 | 1,800 | 36.2 | 35.9 | 88.5 | 93.0 |
| | 1,200 | 34.8 | 34.5 | 88.5 | 91.0 |
| 40 | 1,800 | 48.9 | 47.8 | 88.5 | 93.0 |
| | 1,200 | 46.0 | 46.2 | 90.2 | 92.4 |
| 50 | 1,800 | 59.3 | 57.7 | 90.2 | 93.6 |
| | 1,200 | 58.1 | 58.0 | 90.2 | 91.7 |
| 60 | 1,800 | 71.6 | 68.8 | 90.2 | 93.6 |
| | 1,200 | 68.5 | 69.6 | 90.2 | 93.0 |
| 75 | 1,800 | 92.5 | 85.3 | 90.2 | 93.6 |
| | 1,200 | 86.0 | 86.5 | 90.2 | 93.0 |
| 100 | 1,800 | 112.0 | 109.0 | 91.7 | 94.5 |
| | 1,200 | 114.0 | 115.0 | 91.7 | 93.6 |
| 125 | 1,800 | 139.0 | 136.0 | 91.7 | 94.1 |
| | 1,200 | 142.0 | 144.0 | 91.7 | 93.6 |
| 150 | 1,800 | 167.0 | 164.0 | 91.7 | 95.0 |
| | 1,200 | 168.0 | 174.0 | 91.7 | 94.1 |
| 200 | 1,800 | 217.0 | 214.0 | 93.0 | 94.1 |
| | 1,200 | 222.0 | 214.0 | 93.0 | 95.0 |

For 900 rpm loads, synchronous motors should be considered at 1000 hp and above.

Motor requirements below 500 hp in the 900 to 1800 rpm speed range are normally handled by standard induction motors.

**514 to 720 rpm** — Synchronous motors are often selected above 1 hp per rpm, such as 800 hp at 720 rpm, 700 hp at 600 rpm, and 600 hp at 514 rpm.

**Below 514 rpm** — The synchronous motor should be considered for sizes down to 200 hp because of higher efficiency, improved power factor, and possible lower cost. At high voltages (4 kV and above), the synchronous motor becomes more economical at even lower horsepowers.

## Motor Voltage

The proper selection of voltage for a given motor drive can vary from a routine procedure to a complex study requiring a complete electrical system analysis. In many instances the in-plant distribution system is well established at a particular voltage, say 2300 (2.3 kV). The new machine may be small compared to available system capacity on the 2.3 kV bus so no problem is involved in purchasing a standard motor of that voltage. In more complicated cases additional substation capacity may be necessary to accommodate the new machine. However, when very large units are to be added, many factors must be considered. A new distribution voltage level, a new transmission line, or a higher voltage transmission from the electric utility might be necessary.

# MOTOR ENCLOSURES

Motor enclosure selection should be predicated upon the environmental conditions under which the motor must operate. Directly related to this is the amount of maintenance required to provide long-term reliability and motor life. In general, the more open the enclosure is to the atmosphere, the lower the first cost of the machine but the higher the maintenance costs that may be necessary. Enclosures frequently used in a-c motors are listed below.

## Drip-Proof

These are generally used only indoors or in enclosed spaces not exposed to severe environmental conditions. Maintenance requirements will depend upon general cleanliness of the location and any chemical contaminants in the area.

## Weather-Protected Type I

This is the least costly outdoor machine. It is essentially a drip-proof guarded motor with heaters and outdoor bearing seals and is very susceptible to weather and atmospheric contamination. Considerable maintenance may be required to ensure satisfactory winding and bearing life.

## Weather-Protected Type II

This is the more commonly used outdoor enclosure. It is more expensive than the WP-I but minimizes the entrance of water and dirt. Maintenance is less than for WP-I types. Chemical contaminants in gaseous form may be carried into a WP-II machine with the ventilating air and attack parts that are vulnerable to them.

**FIG. 15-33**

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**NEMA Frame Size Motors, Induction**

| HP | Approx. Full Load RPM | Amperes Based on 460V | | Efficiency in Percentage at Full Load | |
|---|---|---|---|---|---|
| | | Standard Efficiency | High Efficiency | Standard Efficiency | High Efficiency |
| 1 | 1,800 | 1.9 | 1.5 | 72.0 | 84.0 |
| | 1,200 | 2.0 | 2.0 | 68.0 | 78.5 |
| 1-1/2 | 1,800 | 2.5 | 2.2 | 75.5 | 84.0 |
| | 1,200 | 2.8 | 2.6 | 72.0 | 84.0 |
| 2 | 1,800 | 2.9 | 3.0 | 75.5 | 84.0 |
| | 1,200 | 3.5 | 3.2 | 75.5 | 84.0 |
| 3 | 1,800 | 4.7 | 3.9 | 75.5 | 87.5 |
| | 1,200 | 5.1 | 4.8 | 75.5 | 86.5 |
| 5 | 1,800 | 7.1 | 6.3 | 78.5 | 89.5 |
| | 1,200 | 7.6 | 7.4 | 78.5 | 87.5 |
| 7-1/2 | 1,800 | 9.7 | 9.4 | 84.0 | 90.2 |
| | 1,200 | 10.5 | 9.9 | 81.5 | 89.5 |
| 10 | 1,800 | 12.7 | 12.4 | 86.5 | 91.0 |
| | 1,200 | 13.4 | 13.9 | 84.0 | 89.5 |
| 15 | 1,800 | 18.8 | 18.6 | 86.5 | 91.0 |
| | 1,200 | 19.7 | 19.0 | 84.0 | 89.5 |
| 20 | 1,800 | 24.4 | 25.0 | 86.5 | 91.0 |
| | 1,200 | 25.0 | 24.9 | 86.5 | 90.2 |
| 25 | 1,800 | 31.2 | 29.5 | 88.5 | 91.7 |
| | 1,200 | 29.2 | 29.1 | 88.5 | 91.0 |
| 30 | 1,800 | 36.2 | 35.9 | 88.5 | 93.0 |
| | 1,200 | 34.8 | 34.5 | 88.5 | 91.0 |
| 40 | 1,800 | 48.9 | 47.8 | 88.5 | 93.0 |
| | 1,200 | 46.0 | 46.2 | 90.2 | 92.4 |
| 50 | 1,800 | 59.3 | 57.7 | 90.2 | 93.6 |
| | 1,200 | 58.1 | 58.0 | 90.2 | 91.7 |
| 60 | 1,800 | 71.6 | 68.8 | 90.2 | 93.6 |
| | 1,200 | 68.5 | 69.6 | 90.2 | 93.0 |
| 75 | 1,800 | 92.5 | 85.3 | 90.2 | 93.6 |
| | 1,200 | 86.0 | 86.5 | 90.2 | 93.0 |
| 100 | 1,800 | 112.0 | 109.0 | 91.7 | 94.5 |
| | 1,200 | 114.0 | 115.0 | 91.7 | 93.6 |
| 125 | 1,800 | 139.0 | 136.0 | 91.7 | 94.1 |
| | 1,200 | 142.0 | 144.0 | 91.7 | 93.6 |
| 150 | 1,800 | 167.0 | 164.0 | 91.7 | 95.0 |
| | 1,200 | 168.0 | 174.0 | 91.7 | 94.1 |
| 200 | 1,800 | 217.0 | 214.0 | 93.0 | 94.1 |
| | 1,200 | 222.0 | 214.0 | 93.0 | 95.0 |

### Totally Enclosed Forced Ventilated (TEFV)

TEFV enclosures can be used indoors or outdoors in dirty or hazardous environments. Since the motor cooling air is piped in from a separate source the influx of dirt and gaseous contaminants is minimized. Maintenance is minimal depending upon the cleanliness of the cooling air.

### Totally Enclosed Water-to-Air Cooled (TEWAC)

The totally enclosed water-to-air cooled machine uses an air to water heat exchanger to remove heat generated by motor losses. It is the quietest enclosure available and will usually result in the lowest maintenance costs. It will breathe during shutdown but often a breather filter is used to remove particulate contaminants. It is more efficient than a TEFC motor because it does not have the external fan to drive. Its first cost is greater than WP-II but less than TEFC, excluding any additional capital cost for a cooling water system. Operating costs are higher because of the necessity to continuously supply it with cooling water.

### Totally Enclosed Fan Cooled (TEFC)

This is the highest degree of enclosure for an air cooled machine. In large sizes, the TEFC motor has an air-to-air heat exchanger. Internal motor air is recirculated around the outside of the tubes while outside air is driven through the tubes by a shaft driven fan. These motors are quite expensive especially in large sizes because of the high volume of cooling air required relative to motor size. These motors are indicated for use in very dirty or hazardous locations.

The TEFC enclosure minimizes the maintenance required for these very dirty applications. However, the machines will breathe when shut down and vapor and gaseous contaminants can be drawn into them. TEFC motors are usually noisy because of the large external fan.

### Explosion-Proof

An explosion-proof machine is a totally enclosed machine whose enclosure is designed and constructed to withstand an internal explosion. It is also designed to prevent the ignition of combustibles surrounding the machine by sparks, flashes, or explosions which may occur within the machine casing.

## THE INDUCTION GENERATOR

The induction generator can be used as a convenient means of recovering industrial process energy that would otherwise be wasted. Excess steam or compressed gas can often drive such a generator to convert useless energy to valuable kilowatts.

An induction generator is simply an induction motor driven above its synchronous speed by a suitable prime mover. This results in production rather than consumption of electric energy. Normally the induction generator does not differ in any aspect of electrical or mechanical construction from an induction motor. Only the operating speed range separates one mode of behavior from the other.

Important differences exist between the induction generator and the more widely used synchronous generator. These are basically the same as the differences between induction and synchronous motors. Besides low cost and simplicity of control an important benefit is that the induction machine is instantly convertible from generator to motor operation or vice versa.

The synchronous generator needs precise prime-mover speed control to maintain its output at correct frequency. When connected to a public utility system such a machine cannot be allowed to deviate more than a fraction of a cycle from rated frequency without being tripped off the line. However, speed changes do not affect the voltage or power output of the generator — only the frequency.

For the induction machine, voltage and frequency remain constant, set by the connected power system, whatever the driven speed. The speed change does directly affect the power output of the generator and therefore the temperature of its windings. Unless other machines are coupled into the same drive to dampen speed swings, close control of rpm is almost as necessary to the induction generator as to the alternator.

Smaller generators (down to 300 kW) are finding many uses. Among them:

- Recovering energy of compression on the downhill side of a natural gas pipeline.
- Producing electric power from the expansion of geothermal steam.
- Generating power through expansion of compressed gases in cryogenic production.
- Recovering energy from single-stage waste steam turbines in the 5-175 psig inlet pressure range.

## SPEED VARIATION

Because of the continuing increase in the cost of electric energy, variable speed drives offer an economical means of reducing energy requirements in many areas of operation.

### Variable Frequency Electric Motors

For many years variable speed applications relied on either d-c motors or a constant speed a-c motor coupled to various mechanical systems to provide the range of speeds required. Solid-state electronics provide an effective means of speed control for a-c motors by changing the frequency of the electrical signal. They can be used with both induction and synchronous motors.

A standard a-c motor operating at 60 hertz will operate at a constant speed, depending upon the number of magnetic poles it has in accordance with the formula:

$$\text{rpm} = \frac{120f}{P} \qquad \textbf{Eq 15-2}$$

However, if the input frequency can be varied in accordance with the speed requirements, then a wide range of speeds can be obtained. For example, with a frequency range from 50 to 120 hertz, a 4-pole motor has a speed range from 1500 through 3600 rpm.

### Fixed Speed Electric Motors With Fluid Couplings

The speed of an equipment item driven by a fixed-speed electric motor can be varied with a fluid coupling. This is essentially a pump discharging to a power-recovery turbine, both in the same casing. The pump is connected to the driver shaft and the turbine to the driven shaft. The turbine speed is varied by varying the amount of fluid in the casing. Increasing the fluid increases the circulation between the pump and turbine, thereby increasing the speed of the turbine. A fluid coupling the fly-ball governor replaced by speed pick-ups and the hydraulic relays with electronic circuit boards. A third generation control system was developed and replaced the electronic circuitry with digital logic. A microprocessor is used and the control logic is programmed into the governor. The major advantage of this system is the ability to utilize two governors simultaneously, each capable of governing the turbine alone. If the primary governor incurs a fault, the back-up governor assumes control of the turbine and provides diagnostic information to the operator.

## STEAM TURBINE EFFICIENCY

### Factors Affecting Efficiency

The objective of the steam turbine is to maximize the use of the available steam energy where the available steam energy is defined as the difference between the inlet and exhaust energies (enthalpies) for a 100% efficient constant entropy (i.e., isentropic) process. There are numerous loss mechanisms which reduce the efficiency from the isentropic such as throttling losses, steam leakage, friction between the steam and the nozzles/buckets, bearing losses, etc. Efficiency can range from a low of 40% for a low horsepower single-stage turbine to a high approaching 90% for a large multistage, multi-valve turbine.

### Techniques to Improve Efficiency

Various techniques are employed to maximize turbine efficiency, each designed to attack a specific loss mechanism. For example, the number of stages utilized can range from the fewest possible to develop the load reliably to the thermodynamically optimum selection. Spill bands can be utilized to minimize throttling losses. High efficiency nozzle/bucket profiles are available to reduce friction losses. Exhaust flow guides are available to reduce the pressure within the exhaust casing.

The specific features employed on a given application are usually based on the trade-off between capital investment and the cost to produce steam over the life of the turbine.

### Operation at Part Load

Most equipment driven by steam turbines are centrifugal machines where horsepower varies as the cube of speed. Part load efficiency varies as a function of speed, flow, and the number of stages. By assuming horsepower to vary as the cube of speed the turbine part load efficiency can be approximated as a percentage of the design efficiency (Fig. 15-11).

## EXAMPLES

Figs. 15-11 through 15-19 and 24-30 and 24-31 allow estimates to be made of steam rate, turbine efficiency, number of stages, and the inlet and exhaust nozzle diameters. The following examples illustrate the use of these figures:

**Example 15-1** — Given a steam turbine application with the following characteristics:

| | |
|---|---|
| Inlet Pressure | 600 psia |
| Inlet Temperature | 750°F |
| Exhaust Pressure | 2 psia |
| Required Horsepower | 6000 hp |
| Speed | 7000 rpm |

Determine:

- The actual steam rate (ASR).
- The inlet and exhaust nozzle diameters.
- The approximate number of stages.
- The steam rate at a partial load of 4000 hp and 6100 rpm.

**Solution Steps**

Using Figs. 24-30 and 31, the theoretical steam rate (TSR) may be determined from the difference in the inlet enthalpy and the theoretical exhaust enthalpy (i.e. isentropic exhaust

**FIG. 15-11**

**Part Load Efficiency Correction Factor vs Percent Power Multi-Valve Steam Turbines**

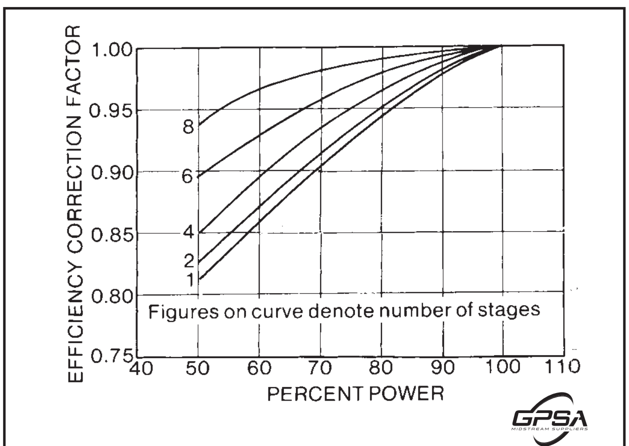
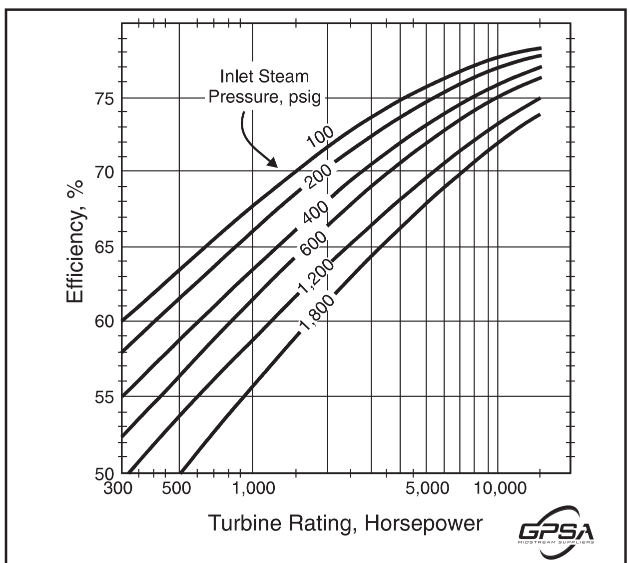



**FIG. 15-12**

**Basic Efficiency of Multi-Valve, Multi-Stage Condensing Turbines**



## ABBREVIATIONS

| | |
|---|---|
| abs | absolute |
| ata | atmosphere absolute |
| Btu | British thermal unit |
| Btu/hr | British thermal unit/hour |
| °C | Celsius |
| cfm | cubic foot/minute |
| cm | centimeter |
| $cm^2$ | square centimeter |
| $cm^3$ | cubic centimeter |
| cu.ft. | cubic foot |
| °F | Fahrenheit |
| ft/sec | foot/second |
| ft-lb | foot-pound |
| gal | gallon |
| hp | horsepower |
| in | inch |
| in. Hg | inch mercury |
| in. $H_2O$ | inch water |
| kcal | kilocalorie |
| kg | kilogram |
| kJ | kilojoule |
| kPa | kilopascal |
| kW | kilowatt |
| L | liter |
| m | meter |
| mm | millimeter |
| $m^2$ | square meter |
| $m^3$ | cubic meter |
| $m^3$/min | cubic meter/minute |
| mph | mile per hour |
| N | Newton |
| N/$m^2$ | Pascal |
| N$m^3$/hr | normal* cubic meter/hour |
| psi | pound/square inch |
| psia | pound/square inch absolute |
| psig | pound/square inch gage |
| scf | standard* cubic foot |
| scfm | standard* cubic foot/minute |
| sq | square |

* "Normal" = 0°C and 1.01325 x $10^5$ Pascals
* "Standard" = 59°F and 14.73 psia

## MILLIMETERS (mm) TO INCHES (in)
(1 millimeter = 0.03937 inch)

| mm | in | mm | in | mm | in | mm | in | mm | in |
|---|---|---|---|---|---|---|---|---|---|
| **1** | 0.039 | **21** | 0.827 | **41** | 1.614 | **61** | 2.402 | **81** | 3.189 |
| **2** | 0.079 | **22** | 0.866 | **42** | 1.654 | **62** | 2.441 | **82** | 3.228 |
| **3** | 0.118 | **23** | 0.906 | **43** | 1.693 | **63** | 2.480 | **83** | 3.268 |
| **4** | 0.157 | **24** | 0.945 | **44** | 1.732 | **64** | 2.520 | **84** | 3.307 |
| **5** | 0.197 | **25** | 0.984 | **45** | 1.772 | **65** | 2.559 | **85** | 3.346 |
| **6** | 0.236 | **26** | 1.024 | **46** | 1.811 | **66** | 2.598 | **86** | 3.386 |
| **7** | 0.276 | **27** | 1.063 | **47** | 1.850 | **67** | 2.638 | **87** | 3.425 |
| **8** | 0.315 | **28** | 1.102 | **48** | 1.890 | **68** | 2.677 | **88** | 3.465 |
| **9** | 0.354 | **29** | 1.142 | **49** | 1.929 | **69** | 2.717 | **89** | 3.504 |
| **10** | 0.394 | **30** | 1.181 | **50** | 1.968 | **70** | 2.756 | **90** | 3.543 |
| **11** | 0.433 | **31** | 1.220 | **51** | 2.008 | **71** | 2.795 | **91** | 3.583 |
| **12** | 0.472 | **32** | 1.260 | **52** | 2.047 | **72** | 2.835 | **92** | 3.622 |
| **13** | 0.512 | **33** | 1.299 | **53** | 2.087 | **73** | 2.874 | **93** | 3.661 |
| **14** | 0.551 | **34** | 1.339 | **54** | 2.126 | **74** | 2.913 | **94** | 3.701 |
| **15** | 0.591 | **35** | 1.378 | **55** | 2.165 | **75** | 2.953 | **95** | 3.740 |
| **16** | 0.630 | **36** | 1.417 | **56** | 2.205 | **76** | 2.992 | **96** | 3.779 |
| **17** | 0.669 | **37** | 1.457 | **57** | 2.244 | **77** | 3.032 | **97** | 3.819 |
| **18** | 0.709 | **38** | 1.496 | **58** | 2.283 | **78** | 3.071 | **98** | 3.858 |
| **19** | 0.748 | **39** | 1.535 | **59** | 2.323 | **79** | 3.110 | **99** | 3.898 |
| **20** | 0.787 | **40** | 1.575 | **60** | 2.362 | **80** | 3.150 | **100** | 3.937 |

## KILOGRAMS (kg) TO POUNDS (lb)
(1 kilogram = 2.20462 pounds)

| kg | lb | kg | lb | kg | lb | kg | lb | kg | lb |
|---|---|---|---|---|---|---|---|---|---|
| **1** | 2.204 | **21** | 46.297 | **41** | 90.390 | **61** | 134.482 | **81** | 178.574 |
| **2** | 4.409 | **22** | 48.502 | **42** | 92.594 | **62** | 136.687 | **82** | 180.779 |
| **3** | 6.614 | **23** | 50.706 | **43** | 94.799 | **63** | 138.891 | **83** | 182.984 |
| **4** | 8.819 | **24** | 52.911 | **44** | 97.003 | **64** | 141.096 | **84** | 185.188 |
| **5** | 11.023 | **25** | 55.116 | **45** | 99.208 | **65** | 143.300 | **85** | 187.393 |
| **6** | 13.228 | **26** | 57.320 | **46** | 101.413 | **66** | 145.505 | **86** | 189.598 |
| **7** | 15.432 | **27** | 59.525 | **47** | 103.617 | **67** | 147.710 | **87** | 191.802 |
| **8** | 17.637 | **28** | 61.729 | **48** | 105.822 | **68** | 149.914 | **88** | 194.007 |
| **9** | 19.843 | **29** | 63.934 | **49** | 108.026 | **69** | 152.119 | **89** | 196.211 |
| **10** | 22.046 | **30** | 66.139 | **50** | 110.231 | **70** | 154.324 | **90** | 198.416 |
| **11** | 24.251 | **31** | 66.343 | **51** | 112.436 | **71** | 156.528 | **91** | 200.621 |
| **12** | 26.455 | **32** | 70.548 | **52** | 114.640 | **72** | 158.733 | **92** | 202.825 |
| **13** | 28.660 | **33** | 72.753 | **53** | 116.845 | **73** | 160.937 | **93** | 205.030 |
| **14** | 30.865 | **34** | 74.957 | **54** | 119.050 | **74** | 163.142 | **94** | 207.235 |
| **15** | 33.069 | **35** | 77.162 | **55** | 121.254 | **75** | 165.347 | **95** | 209.439 |
| **16** | 35.274 | **36** | 79.366 | **56** | 123.459 | **76** | 167.551 | **96** | 211.644 |
| **17** | 37.479 | **37** | 81.571 | **57** | 125.663 | **77** | 169.756 | **97** | 213.848 |
| **18** | 39.683 | **38** | 83.776 | **58** | 127.868 | **78** | 171.961 | **98** | 216.053 |
| **19** | 41.888 | **39** | 85.980 | **59** | 130.073 | **79** | 174.165 | **99** | 218.258 |
| **20** | 44.093 | **40** | 88.185 | **60** | 132.277 | **80** | 176.370 | **100** | 220.462 |

## CONVERSION FACTORS

| TO CONVERT FROM ENGLISH | TO S.I. METRIC | MULTIPLY BY | TO OLD METRIC | MULTIPLY BY |
|---|---|---|---|---|
| sq. in. | $mm^2$ | **645.16** | $cm^2$ | **6.4516** |
| sq. ft. | $m^2$ | **0.0929** | $m^2$ | **0.0929** |
| lb/cu.ft. | kg/$m^3$ | **16.0185** | kg/$m^3$ | **16.0185** |
| $lb_f$ | N | **4.4482** | N | **4.4482** |
| $lb_f$/ft | N/m | **14.5939** | N/m | **14.5939** |
| Btu | kJ | **1.0551** | kcal | **0.252** |
| Btu/hr | W | **0.2931** | kcal/hr | **0.252** |
| Btu/scf | kJ/$mm^3$ | **37.2590** | kcal/n$m^3$ | **0.1565** |
| in | mm | **25.400** | cm | **2.540** |
| ft | m | **0.3048** | m | **0.3048** |
| yd | m | **0.914** | m | **0.914** |
| lb | kg | **0.4536** | kg | **0.4536** |
| hp | kW | **0.7457** | kW | **0.7457** |
| psi | kPa | **6.8948** | kg/$cm^2$ | **0.070** |
| psia | kPa abs | **6.8948** | bars abs | **0.0716** |
| psig | kPa gage | **6.8948** | ata | **0.070** |
| in. Hg | kPa | **3.3769** | cm Hg | **2.540** |
| in. $H_2O$ | kPa | **0.2488** | cm $H_2O$ | **2.540** |
| °F | °C = | **(°F -32) 5/9** | °C = | **(°F -32) 5/9** |
| °F (Interval) | °C (Interval) | **5/9** | °C (Interval) | **5/9** |
| ft-lb | N • m | **1.3558** | N • m | **1.3558** |
| mph | km/hr | **1.6093** | km/hr | **1.6093** |
| ft/sec | m/sec | **0.3048** | m/sec | **0.3048** |
| cu. ft. | $m^3$ | **0.0283** | $m^3$ | **0.0283** |
| gas (US) | L | **3.7854** | L | **3.7854** |
| cfm | $m^3$/min | **0.0283** | $m^3$/min | **0.0283** |
| scfm | n$m^3$/min | **0.0268** | n$m^3$/hr | **1.61** |

| TO CONVERT FROM OLD METRIC | TO S.I. METRIC | MULTIPLY BY |
|---|---|---|
| $cm^2$ | $mm^2$ | **100.** |
| kcal | kJ | **4.1868** |
| kcal/hr | W | **1.16279** |
| cm | mm | **10.** |
| kg/$cm^2$ | kPa | **98.0665** |
| bars | kPa | **100.** |
| atm | kPa | **101.325** |
| cm Hg | kPa | **1.3332** |
| cm $H_2O$ | kPa | **9.8064** |
| n$m^3$/hr | n$m^3$/min | **0.0176** |

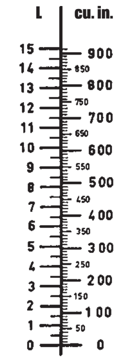
## TEMPERATURE CONVERSION TABLES*

| | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 TO 100 | | | 2.78 | **37** | 98.6 | 23.9 | **75** | 167.0 | 93 | **200** | 392 | 299 | **570** | 1058 | 510 | **950** | 1742 |
| -17.8 | **0** | 32 | 3.33 | **38** | 100.4 | 24.4 | **76** | 168.8 | 99 | **210** | 410 | 304 | **580** | 1076 | 516 | **960** | 1760 |
| -17.2 | **1** | 33.8 | 3.89 | **39** | 102.2 | 25.0 | **77** | 170.6 | 100 | **212** | 413 | 310 | **590** | 1094 | 521 | **970** | 1778 |
| -16.7 | **2** | 35.6 | 4.44 | **40** | 104.0 | 25.6 | **78** | 172.4 | 104 | **220** | 428 | 316 | **600** | 1112 | 527 | **980** | 1796 |
| -16.1 | **3** | 37.4 | 5.00 | **41** | 105.8 | 26.1 | **79** | 174.2 | 110 | **230** | 446 | 321 | **610** | 1130 | 532 | **990** | 1814 |
| -15.6 | **4** | 39.2 | 5.56 | **42** | 107.6 | 26.7 | **80** | 176.0 | 116 | **240** | 464 | 327 | **620** | 1148 | 538 | **1000** | 1832 |
| -15.0 | **5** | 41.0 | 6.11 | **43** | 109.4 | 27.2 | **81** | 177.8 | 121 | **250** | 482 | 332 | **630** | 1166 | | | |
| -14.4 | **6** | 42.8 | 6.67 | **44** | 111.2 | 27.8 | **82** | 179.6 | 127 | **260** | 500 | 338 | **640** | 1184 | 1000 TO 1630 | | |
| -13.9 | **7** | 44.9 | 7.22 | **45** | 113.0 | 28.3 | **83** | 181.4 | 132 | **270** | 518 | 343 | **650** | 1202 | 538 | **1000** | 1832 |
| -13.3 | **8** | 46.4 | 7.78 | **46** | 114.8 | 28.9 | **84** | 183.2 | 138 | **280** | 536 | 349 | **660** | 1220 | 543 | **1010** | 1850 |
| -12.8 | **9** | 48.2 | 8.33 | **47** | 116.6 | 29.4 | **85** | 185.0 | 143 | **290** | 554 | 354 | **670** | 1238 | 549 | **1020** | 1868 |
| -12.1 | **10** | 50.0 | 8.89 | **48** | 118.4 | 30.0 | **86** | 186.8 | 149 | **300** | 572 | 360 | **680** | 1256 | 554 | **1030** | 1886 |
| -11.7 | **11** | 51.8 | 9.44 | **49** | 120.0 | 30.6 | **87** | 188.6 | 154 | **310** | 590 | 366 | **690** | 1274 | 560 | **1040** | 1904 |
| -11.1 | **12** | 53.6 | 10.0 | **50** | 122.0 | 31.1 | **88** | 190.4 | 160 | **320** | 608 | 371 | **700** | 1292 | 566 | **1050** | 1922 |
| -10.6 | **13** | 55.4 | 10.6 | **51** | 123.8 | 31.7 | **89** | 192.2 | 166 | **330** | 626 | 377 | **710** | 1310 | 571 | **1060** | 1940 |
| -10.0 | **14** | 57.2 | 11.1 | **52** | 125.6 | 32.2 | **90** | 194.0 | 171 | **340** | 644 | 382 | **720** | 1328 | 577 | **1070** | 1958 |
| -9.44 | **15** | 59.0 | 11.7 | **53** | 127.4 | 32.8 | **91** | 195.8 | 177 | **350** | 662 | 388 | **730** | 1346 | 582 | **1080** | 1976 |
| -8.89 | **16** | 60.8 | 12.2 | **54** | 129.2 | 33.3 | **92** | 197.6 | 182 | **360** | 680 | 393 | **740** | 1364 | 588 | **1090** | 1994 |
| -8.33 | **17** | 62.6 | 12.8 | **55** | 131.0 | 33.9 | **93** | 199.4 | 188 | **370** | 698 | 399 | **750** | 1382 | 593 | **1100** | 2012 |
| -7.78 | **18** | 64.4 | 13.3 | **56** | 132.8 | 34.4 | **94** | 201.2 | 193 | **380** | 716 | 404 | **760** | 1400 | 599 | **1110** | 2030 |
| -7.22 | **19** | 66.2 | 13.9 | **57** | 134.6 | 35.0 | **95** | 203.0 | 199 | **390** | 734 | 410 | **770** | 1418 | 604 | **1120** | 2048 |
| -6.67 | **20** | 68.0 | 14.4 | **58** | 136.4 | 35.6 | **96** | 204.8 | 204 | **400** | 752 | 416 | **780** | 1436 | 610 | **1130** | 2066 |
| -6.11 | **21** | 69.8 | 15.0 | **59** | 138.2 | 36.1 | **97** | 206.6 | 210 | **410** | 770 | 421 | **790** | 1454 | 816 | **1500** | 2732 |
| -5.56 | **22** | 71.6 | 15.6 | **60** | 140.0 | 36.7 | **98** | 208.4 | 216 | **420** | 788 | 427 | **800** | 1472 | 821 | **1510** | 2750 |
| -5.00 | **23** | 73.4 | 16.1 | **61** | 141.8 | 37.2 | **99** | 210.2 | 221 | **430** | 806 | 432 | **810** | 1490 | 827 | **1520** | 2768 |
| -4.44 | **24** | 75.2 | 16.7 | **62** | 143.6 | 37.8 | **100** | 212.0 | 227 | **440** | 824 | 438 | **820** | 1508 | 832 | **1530** | 2786 |
| -3.89 | **25** | 77.0 | 17.2 | **63** | 145.4 | | | | 232 | **450** | 842 | 443 | **830** | 1526 | 838 | **1540** | 2804 |
| -3.33 | **26** | 78.8 | 17.8 | **64** | 147.2 | 100 TO 1000 | | | 238 | **460** | 860 | 449 | **840** | 1544 | 843 | **1550** | 2822 |
| -2.78 | **27** | 80.6 | 18.3 | **65** | 149.0 | 38 | **100** | 212 | 243 | **470** | 878 | 454 | **850** | 1562 | 849 | **1560** | 2840 |
| -2.22 | **28** | 82.4 | 18.9 | **66** | 150.8 | 43 | **110** | 230 | 249 | **480** | 896 | 460 | **860** | 1580 | 854 | **1570** | 2858 |
| -1.67 | **29** | 84.2 | 19.4 | **67** | 152.6 | 49 | **120** | 248 | 254 | **490** | 914 | 466 | **870** | 1598 | 860 | **1580** | 2876 |
| -1.11 | **30** | 86.0 | 20.0 | **68** | 154.4 | 54 | **130** | 266 | 260 | **500** | 932 | 471 | **880** | 1616 | 866 | **1590** | 2894 |
| -0.56 | **31** | 87.8 | 20.6 | **69** | 156.2 | 60 | **140** | 284 | 266 | **510** | 950 | 477 | **890** | 1634 | 871 | **1600** | 2912 |
| 0 | **32** | 89.6 | 21.1 | **70** | 158.0 | 66 | **150** | 302 | 271 | **520** | 968 | 482 | **900** | 1652 | 877 | **1610** | 2930 |
| 0.56 | **33** | 91.4 | 21.7 | **71** | 159.8 | 71 | **160** | 320 | 277 | **530** | 986 | 488 | **910** | 1670 | 882 | **1620** | 2948 |
| 1.11 | **34** | 93.2 | 22.2 | **72** | 161.6 | 77 | **170** | 338 | 282 | **540** | 1004 | 493 | **920** | 1688 | 888 | **1630** | 2966 |
| 1.67 | **35** | 95.0 | 22.8 | **73** | 163.4 | 82 | **180** | 356 | 288 | **550** | 1022 | 499 | **930** | 1706 | | | |
| 2.22 | **36** | 96.8 | 23.3 | **74** | 165.2 | 88 | **190** | 374 | 293 | **560** | 1040 | 504 | **940** | 1724 | | | |

Note: The numbers in **bold** face type refer to the temperature either in degrees Centigrade or Fahrenheit which is desired to convert into the other scale. If converting from Fahrenheit degrees to Centigrade degrees, the equivalent temperatures will be found in the left column; while if converting from degrees Centigrade to degrees Fahrenheit, the answer will be found in the column on the right.

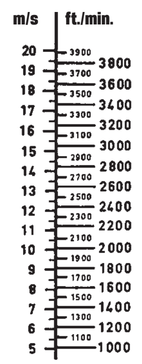
## VOLUME CONVERSION FACTORS

**1 L = 61.02 cu. in.**
**10 cu. in. = 0,164 L**



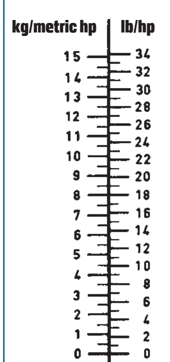

## PISTON SPEED CONVERSION FACTORS

**1 m/s = 196.9 ft./min.**
**100 ft./min. = 0,51 m/s**




## WEIGHT/ HORSEPOWER CONVERSION FACTORS

**1 kg/metric hp = 2.235 lb./hp**
**1 lb/hp = .4474 kg/metric hp**



with a particular service. Internal combustion engines are classified according to speed in the following broad categories:

High speed — above 1500 rpm
Medium speed — 700 to 1500 rpm
Low speed — below 700 rpm

High speed engines can offer weight and space advantages but will usually require more maintenance than a medium or low speed engine. High speed engines are often selected for standby or intermittent applications. As a general rule the lower the speed the longer the service life. Although internal combustion engines are usually selected to run over a limited speed range, they will operate well over large ranges of speed just as an automobile engine does.

## PERFORMANCE RATING

Several measurements of performance can be used to compare engines. Four commonly used measurements are:

1. Specific fuel consumption, (lb or Btu)/(bhp<$E• >hr)
2. BMEP, psi
3. Specific weight, lb/bhp
4. Output per unit of displacement, bhp/cu in.

The relationship between brake mean effective pressure (BMEP) and brake horsepower (bhp) is given below.

$$BMEP = \frac{(bhp)\ (33{,}000)}{(S)\ (A)\ (N)} \qquad \textbf{Eq 15-3}$$

The value of N is equivalent to RPM for two-stroke-cycle engines, and RPM divided by two for four-stroke cycle. BMEP indicates how much turbocharging increases the brake horsepower which is the power delivered to the driven equipment by the engine output shaft.

The intended use of the engine will determine the most important measure of performance. For an aircraft engine the first and third items may be the most important; while for a stationary engine in continuous service with no space or weight limitations, the first item would be of primary importance.

The power delivered is directly related to atmospheric conditions. Operation in areas of low atmospheric pressure (high altitudes) will reduce the power output. High inlet air temperature will also reduce the power output. Engines are rated for various altitudes above sea level (i.e. barometric pressures) and ambient temperatures (e.g. 1500/3000 feet and 90°F according to DEMA; 1500 feet and 85°F; and so forth). A rule of thumb for derating naturally aspirated engines is 3.5% reduction in power for each 1000 ft above the rating altitude, and 1% reduction for every 10°F above the rating temperature. For exact deration of naturally aspirated engines, or for turbocharged engines, the manufacturers must be consulted.

Following are gas-engine design parameters. The values vary considerably depending on the engine type, make and model, and on the site conditions, but ranges of typical values are given.

- Fuel-gas requirements (i.e. heat rate) [6500 to 8500 Btu/(bhp · hr), LHV].
- Heat rejection at the power-end exhaust manifold [1500 to 3000 Btu/(bhp · hr) with jacket water cooling, or 800 to 1500 without].
- Heat rejection at a turbo aftercooler if applicable [100 to 500 Btu/(bhp · hr)].
- Heat rejection at the lube-oil cooler [300 to 900 Btu/(bhp · hr)].

New technologies have reduced specific weights (i.e. lb/bhp), increased fuel efficiencies, lengthened the periods between overhauls, and reduced emissions. Precisely programmed electronically controlled fuel injection incorporates ambient and other important operating conditions to minimize fuel consumption and emissions over full operating ranges. Many engine designs include pre-combustion chambers that jet flames into the main combustion chambers effectively igniting leaner air/fuel mixtures (i.e. "lean burn") resulting in higher efficiencies and lower emissions. New thermal-barrier coatings (TBCs) insulate many engine components from thermal shock and reduce heat losses that would otherwise decrease thermal efficiencies.

### Engine Energy Balance

A gas engine converts the combustion energy in the fuel to mechanical power and heat. The combustion energy is usually distributed as follows:

| | % Range |
|---|---|
| Mechanical power | 30–40 |
| Heat rejected to cylinder cooling | 25–40 |
| Heat rejected to oil cooler | 3–5 |
| Heat rejected to turbo aftercooler | 4–9 |
| Heat rejected to exhaust | 25–30 |
| Heat rejected to atmosphere (i.e. surface heat loss) | 3–6 |

The mechanical power is the sum of the brake horsepower (bhp) (i.e. available shaft power), and the power to drive such engine auxiliaries as a lube-oil pump, cooling-water pump, radiator fan, and alternator (for a spark ignition engine).

Fig. 15-35 includes engine power ratings, specific fuel requirements (i.e., "heat rates"), heat rejections and exhaust con-

**FIG. 15-34**
**Grades of Diesel Fuel, ASTM D-975 (1995) Classification**

| | 1-D | 2-D | 4-D |
|---|---|---|---|
| Flash point, °F Min | 100 | 125 | 130 |
| Carbon Residue, % Max | 0.15 | 0.35 | – |
| Water and Sediment, % by Vol Max | 0.05 | 0.05 | 0.50 |
| Ash, % by Wt Max | 0.01 | 0.01 | 0.10 |
| Distillation °F 90% Pt Max | 550 | 640 | – |
| Distillation °F 90% Pt Min | – | 540 | – |
| Viscosity at 104°F Centistokes Min | 1.3 | 1.9 | 5.5 |
| Viscosity at 104°F Centistokes Max | 2.4 | 4.1 | 24.0 |
| Sulfur, % by Wt | 0.05 | 0.05 | 2.0 |
| Cetane No. Min | 40 | 40 | 30 |
| Aromaticity, % by Vol Max | 35 | 35 | – |

ditions for a variety of gas engines. The values are based on full design operating power at the speeds noted for various altitudes above sea level and ambient temperatures.

An engine's power efficiency, typically called "thermal efficiency," is calculated from the following equation:

$$\text{Thermal Efficiency} = \frac{100 \times 2544}{\text{Heat Rate (Btu/(bhp} \cdot \text{hr), LHV)}}$$

The total heat rejected is calculated from the following equation:

Heat Rejected = (Heat Rate – 2544) Btu/(bhp · hr)

The heat rejected to the engine exhaust gas is calculated from the following equation:

Exhaust Heat Btu/(bhp · hr) = Total heat rejected minus the sum of the heat rejected to cylinder cooling, oil cooling, turbo aftercooling, and engine-surface heat loss to the atmosphere

It is technically feasible to recover part of the heat. Low temperature heat at about 180°F, for such as space heating, can be recovered from the cooling circuits for cylinder jackets, lube oil and turbo charged air. Higher level heat at above 300°F can be recovered by heat exchange with engine exhaust. Below 300°F water vapor will condense with $CO_2$ absorption, acid formation, and resulting corrosion. A heat recovery arrangement is illustrated in Fig. 15-36. Technical feasibility depends upon the economic criteria and improves as the engine size increases. Heat recovery can increase the overall thermal efficiency to as high as 75%. For example, an engine's thermal efficiency can be increased from a typical regular value of 33% to 75% by recovering about 60% of the heat normally rejected to the coolant and exhaust.

**Example 15-4** —Calculate the thermal efficiency, total heat rejected, and total exhaust heat for a Waukesha L7042GL at 1200 RPM, 77°F and sea level, and its full power rating.

**Solutions Steps**

From Fig. 15-35

Full Power = 1480 bhp

Heat rate = 7284 Btu/(bhp · hr), LHV

Heat rejected to water cooling, oil cooling, turbo intercooling, and radiation = 1953 + 298 + 427 + 189 = 2867 Btu/(bhp · hr)

Therefore:

$$\text{Thermal efficiency} = \frac{100 \times 2544}{7284} = 34.9\%$$

Heat rejected per bhp = 7284 – 2544 = 4740[Btu/(bhp · hr)]

Total heat rejected = 4740 [Btu/(bhp · hr)] 1480 bhp = 7.0 MMBtu/hr

Exhaust heat per bhp = 4740 – 2867 = 1873 Btu/(bhp · hr)

Total exhaust heat = 1873 [Btu/(bhp · hr)] 1480 bhp = 2.77 MMBtu/hr

# AUXILIARIES

## Bearings

Hydrodynamic journal bearings are found in all types of industrial turbomachinery, which include pumps, electric motors, steam turbines, electric generators, and gas compressors. The hydrodynamic bearing types most commonly found in turbomachinery are:

- Plane cylindrical
- Pressure dam
- Tilting pad

For all bearing types, the fundamental geometric parameters are journal diameter, pad arc angle, length-to-diameter ratio, and running clearance. Some bearing types, such as tilting pad bearings, have additional geometric variations including number of pads, preload, pad pivot offset angle, and orientation of the bearing (on or between pads). The key operating conditions are oil viscosity, oil density, rotating speed, gravity load at the bearing, and applied external loads (such as gear mesh or pump volute loadings). A machinery expert should be consulted for further details concerning types of bearings and their applications and designs.

## Gears

There are many different types of open gears such as spur, helical, spiral bevel, and worm. This section will focus on enclosed high speed helical gear reducers or increasers commonly used in the natural gas, refinery, and petrochemical industries.

**Speed Increasers and Reducers** — Speed increasers are usually used on centrifugal compressors, axial compressors, blowers, and centrifugal pumps driven by motors, turbines, and industrial combustion engines. Speed reducers are used on reciprocating compressors, rotary positive displacement compressors, centrifugal pumps, generators, and fans driven by turbines and motors.

**High Speed Gears** — High-speed gears are generally defined as having either or both of the following:

1. Pinion speed of at least 2,900 rpm.
2. Pitch line velocities above 5,000 ft/min.

There are units operating with pitch line velocities in excess of 35,000 ft/min and transmitting 30,000 hp.

**Gearing** — High speed gears can be selected with either single helical gearing (used extensively in Europe) or opposed double helical (i.e., "herringbone") gearing (predominant in the United States). Pros and cons of each type of gear design are numerous with double helical gearing being more efficient because there is only one thrust bearing required. The thrust bearing is usually on the low speed shaft.

**Surface Finish** — High speed gears are classified as precision quality gears. Fig. 15-37 shows a minimum surface finish and quality required for various pitch line velocities as recommended in Figure 1, page 14 of AGMA 2001-C95, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.

**FIG. 15-35**
**Engine Ratings and Operating Parameters**

Note: Figures may be approximate due to variations in engine services and are representative of new engines only. Refer to manufacturer for exact information.

| ENGINE | Full Power at Full Speed (bhp) | Full Speed (rpm) | Strokes Per Cycle | BMEP (psi) | Fuel Reqmt [Btu/(bhp·hr)] LHV | Heat Rejection Btu / (bhp · hr): Jacket Water Cooler | Heat Rejection: Cylinder Cooling: Oil Cooler | Heat Rejection: Turbo Intercooler/ Aftercooler | Heat Rejection: Atmosphere i.e. Surface Heat Loss | Exhaust rate [lb/(bhp·hr)] | Exhaust temp °F |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **Caterpillar** | | | | | | | | | | | |
| G3304 NA | 95 | 1800 | 4 | Not Avail. | 7875 | 2574 | 421 | N/A | 316 | 6.85 | 1089 |
| G3304B NA | 95 | 1800 | 4 | Not Avail. | 7875 | 2679 | 404 | N/A | 316 | 7.00 | 1047 |
| G3306 NA | 145 | 1800 | 4 | Not Avail. | 7775 | 2503 | 409 | N/A | 311 | 6.74 | 1101 |
| G3306B NA | 145 | 1800 | 4 | Not Avail. | 7775 | 2486 | 371 | N/A | 311 | 6.78 | 1160 |
| G3306 TA | 203 | 1800 | 4 | Not Avail. | 8098 | 2673 | 423 | 152 | 324 | 7.06 | 1064 |
| G3306B TA | 205 | 1800 | 4 | Not Avail. | 8066 | 2651 | 395 | 139 | 323 | 7.00 | 1094 |
| G3406 NA | 215 | 1800 | 4 | Not Avail. | 7845 | 2532 | 414 | N/A | 313 | 7.43 | 1039 |
| G3406 TA | 276 | 1800 | 4 | Not Avail. | 7418 | 2763 | Note (1) | 81 | 297 | 6.53 | 1004 |
| G3408 NA | 255 | 1800 | 4 | Not Avail. | 7643 | 2392 | 378 | N/A | 305 | 7.07 | 1069 |
| G3408 TA | 332 | 1500 | 4 | Not Avail. | 7507 | 2890 | Note (1) | 45 | 301 | 6.50 | 957 |
| G3408C LE | 425 | 1800 | 4 | Not Avail. | 7595 | 2108 | 333 | 402 | 310 | 9.96 | 806 |
| G3412 TA | 500 | 1500 | 4 | Not Avail. | 7800 | 2725 | 431 | 54 | 312 | 6.61 | 974 |
| G3412C LE | 637 | 1800 | 4 | Not Avail. | 7635 | 2179 | 344 | 435 | 311 | 9.84 | 788 |
| G3508 TA | 524 | 1200 | 4 | Not Avail. | 7712 | 2694 | 402 | 157 | 313 | 6.62 | 914 |
| G3508 LE | 670 | 1400 | 4 | Not Avail. | 7510 | 1630 | 258 | 408 | 285 | 9.62 | 985 |
| G3508B LE | 690 | 1400 | 4 | Not Avail. | 7254 | 938 | 230 | 698 | 304 | 10.62 | 931 |
| G3512 TA | 790 | 1200 | 4 | Not Avail. | 7824 | 2812 | 445 | 187 | 277 | 6.58 | 892 |
| G3512 LE | 860 | 1200 | 4 | Not Avail. | 7402 | 1933 | 288 | 457 | 254 | 9.92 | 823 |
| G3512 LE | 1005 | 1400 | 4 | Not Avail. | 7368 | 1838 | 274 | 493 | 254 | 9.96 | 834 |
| G3512B LE | 1035 | 1400 | 4 | Not Avail. | 7237 | 1008 | 230 | 617 | 270 | 10.39 | 975 |
| G3516 TA | 1050 | 1200 | 4 | Not Avail. | 7700 | 2789 | 441 | 170 | 260 | 6.16 | 912 |
| G3516 LE | 1150 | 1200 | 4 | Not Avail. | 7324 | 1896 | 283 | 392 | 238 | 9.83 | 846 |
| G3516 LE | 1340 | 1400 | 4 | Not Avail. | 7405 | 1886 | 281 | 427 | 238 | 9.80 | 873 |
| G3516B LE | 1380 | 1400 | 4 | Not Avail. | 7301 | 1018 | 195 | 670 | 266 | 10.42 | 992 |
| G3520B LE | 1480 | 1200 | 4 | Not Avail. | 7455 | 1259 | 195 | 520 | 255 | 10.90 | 985 |
| G3520B LE | 1725 | 1400 | 4 | Not Avail. | 7205 | 1056 | 194 | 607 | 255 | 10.28 | 989 |
| G3606 LE | 1775 | 1000 | 4 | Not Avail. | 6649 | 604 | 306 | 470 | 238 | 12.05 | 847 |
| G3608 LE | 2370 | 1000 | 4 | Not Avail. | 6629 | 605 | 306 | 446 | 238 | 11.90 | 857 |
| G3612 LE | 3550 | 1000 | 4 | Not Avail. | 6629 | 614 | 306 | 469 | 238 | 12.03 | 838 |
| G3616 LE | 4735 | 1000 | 4 | Not Avail. | 6605 | 607 | 304 | 435 | 237 | 11.85 | 856 |
| G12CM34 | 6135 | 750 | 4 | Not Avail. | 5839 | 361 | 383 | 757 | 120 | 6.09 | 653 |
| G16CM34 | 8180 | 750 | 4 | Not Avail. | 5839 | 361 | 414 | 757 | 120 | 6.00 | 653 |
| **Cummins** | | | | | | | | | | | |
| G5.9 | 99 | 2200 | 4 | 99 | 8454 | 2051 | Not Avail. | na | 651 | Not Avail. | 1327 |
| G8.3 | 118 | 1800 | 4 | 103 | 8455 | 2335 | Not Avail. | na | 388 | Not Avail. | 1342 |
| GTA8.3 | 175 | 1800 | 4 | 152 | 7369 | 1642 | Not Avail. | 347 | 388 | Not Avail. | 1341 |
| QSL9G | 175 | 1800 | 4 | 142 | 8088 | 2634 | Not Avail. | 304 | 700 | Not Avail. | 1077 |
| G855 | 188 | 1800 | 4 | 97 | 8528 | 2692 | Not Avail. | na | 488 | Not Avail. | 1179 |
| GTA855 | 256 | 1800 | 4 | 132 | 8439 | 2863 | Not Avail. | 286 | 636 | Not Avail. | 1347 |
| KTA19GC | 380 | 1800 | 4 | 144 | 8091 | 2317 | Not Avail. | 208 | 791 | Not Avail. | 1341 |
| KTA38GC | 760 | 1800 | 4 | 144 | 7942 | 3012 | Not Avail. | 149 | 404 | Not Avail. | 1197 |

**FIG. 15-35 (Cont'd.)**

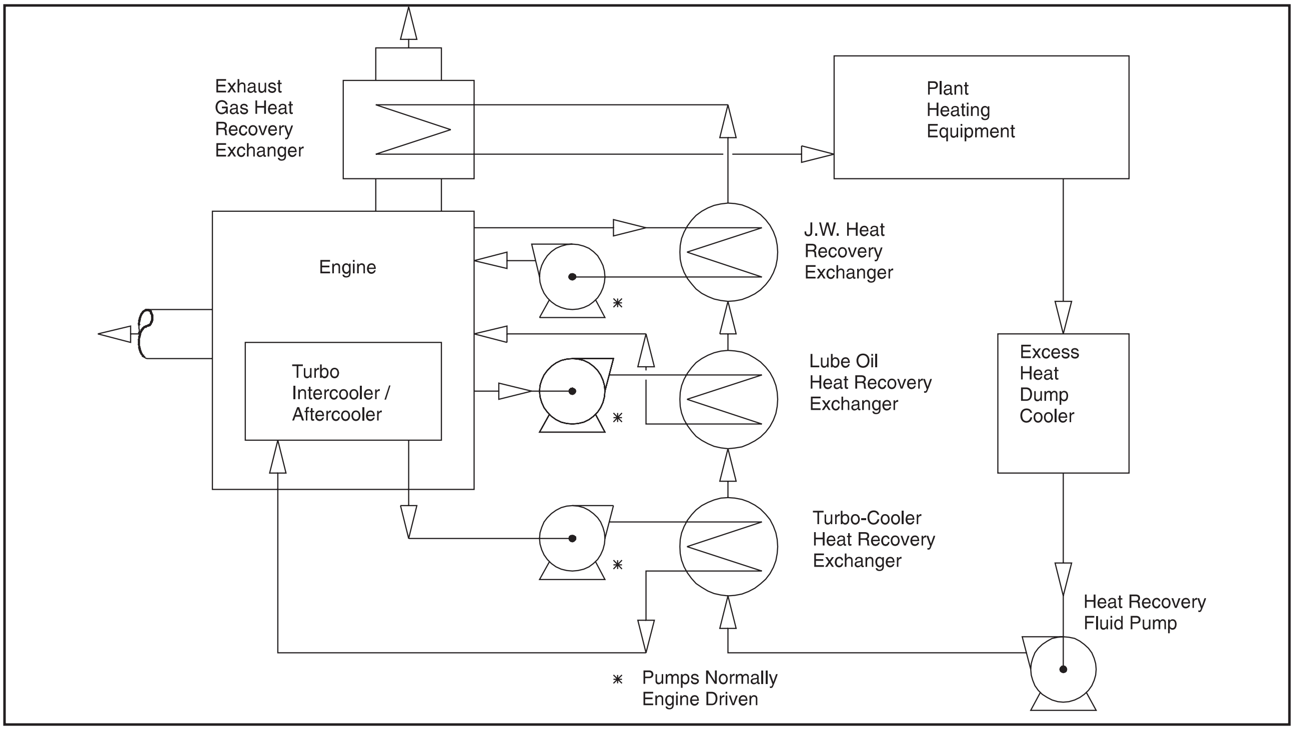
**Engine Ratings and Operating Parameters**

Note: Figures may be approximate due to variations in engine services and are representative of new engines only. Refer to manufacturer for exact information.

| ENGINE | Full Power at Full Speed (bhp) | Full Speed (rpm) | Strokes Per Cycle | BMEP (psi) | Fuel Reqmt [Btu/(bhp·hr)] LHV | Heat Rejection Btu / (bhp · hr): Jacket Water Cooler | Heat Rejection Btu / (bhp · hr): Cylinder Cooling: Oil Cooler | Heat Rejection Btu / (bhp · hr): Turbo Intercooler/ Aftercooler | Heat Rejection Btu / (bhp · hr): Atmosphere i.e. Surface Heat Loss | Exhaust rate [lb/(bhp·hr)] | Exhaust temp °F |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **Wartsila (4)** | | | | | | | | | | | |
| 6L34SG | 3,621 | 750 | 4 | 287 | 5,435 | | | | | | |
| 9L34SG | 5,431 | 750 | 4 | 287 | 5,435 | 725 (5) (6) | 295 (5) | 233 (5) (7) | 94 (5) | 9.82 | 779 |
| 12V34SG | 7,241 | 750 | 4 | 287 | 5,435 | 744 (5) (6) | 295 (5) | 214 (5) (7) | 94 (5) | 9.82 | 779 |
| 16V34SG | 9,655 | 750 | 4 | 287 | 5,435 | | | | | | |
| 20V34SG | 12,069 | 750 | 4 | 287 | 5,435 | 700 (5) (6) | 295 (5) | 240 (5) (7) | 92 (5) | 9.84 | 797 |
| **Waukesha** | | | | | | | | | | | |
| F18G | 240 | 1800 | 4 | 96 | 7570 | 2788 | 225 | — | 204 | 6.45 | 1064 |
| F18GL | 400 | 1800 | 4 | 160 | 7123 | 1875 | 243 | 473 | 155 | 9.37 | 836 |
| F18GSI | 400 | 1800 | 4 | 160 | 7523 | 2285 | 423 | 195 | 248 | 6.57 | 1116 |
| H24G | 320 | 1800 | 4 | 96 | 7897 | 2984 | 234 | — | 184 | 6.73 | 1098 |
| H24GL | 530 | 1800 | 4 | 160 | 7120 | 1879 | 242 | 475 | 138 | 9.37 | 838 |
| H24GSI | 530 | 1800 | 4 | 160 | 7497 | 2294 | 423 | 194 | 213 | 6.55 | 1114 |
| L36GL | 800 | 1800 | 4 | 160 | 7114 | 1874 | 241 | 473 | 120 | 9.36 | 838 |
| L36GSI | 800 | 1800 | 4 | 160 | 7389 | 2335 | 371 | 188 | 178 | 6.45 | 1116 |
| P48GL | 1065 | 1800 | 4 | 160 | 7092 | 1924 | 237 | 472 | 110 | 9.33 | 836 |
| P48GSI | 1065 | 1800 | 4 | 160 | 7373 | 2318 | 366 | 188 | 157 | 6.44 | 1113 |
| F3521G | 515 | 1200 | 4 | 96 | 7336 | 2470 | 383 | — | 336 | 6.41 | 1059 |
| F3521GL | 738 | 1200 | 4 | 138 | 7383 | 2054 | 314 | 432 | 199 | 10.99 | 703 |
| F3514GSI | 740 | 1200 | 4 | 138 | 8180 | 2577 | 409 | 162 | 439 | 6.97 | 1169 |
| F3524GSI | 840 | 1200 | 4 | 158 | 8037 | 2489 | 376 | 165 | 402 | 6.85 | 1192 |
| L5790G | 845 | 1200 | 4 | 96 | 7446 | 2550 | 378 | — | 375 | 6.50 | 1044 |
| L5774LT | 1280 | 1200 | 4 | 146 | 6961 | 1670 | 360 | 303 | 277 | 9.07 | 842 |
| L5794LT | 1450 | 1200 | 4 | 165 | 6995 | 1687 | 337 | 358 | 247 | 9.11 | 849 |
| L5794GSI | 1380 | 1200 | 4 | 158 | 7665 | 2249 | 348 | 129 | 438 | 6.51 | 1136 |
| L7042G | 1025 | 1200 | 4 | 96 | 7351 | 2469 | 382 | — | 316 | 6.42 | 1058 |
| L7042GL | 1480 | 1200 | 4 | 138 | 7284 | 1953 | 298 | 427 | 189 | 10.84 | 710 |
| L7042GSI | 1480 | 1200 | 4 | 138 | 7833 | 2430 | 243 | 190 | 401 | 6.84 | 1126 |
| L7044GSI | 1680 | 1200 | 4 | 158 | 7919 | 2350 | 343 | 150 | 389 | 6.74 | 1179 |
| P9390GL | 1980 | 1200 | 4 | 138 | 7198 | 1784 | 321 | 446 | 165 | 10.72 | 762 |
| P9390GSI | 1980 | 1200 | 4 | 138 | 7930 | 2518 | 262 | 194 | 320 | 6.93 | 1177 |
| 12V275GL+ | 3625 | 1000 | 4 | 220 | 6550 | 596 | 262 | 708 | 83 | 11.77 | 820 |
| 16V275GL+ | 4835 | 1000 | 4 | 220 | 6579 | 620 | 219 | 734 | 81 | 11.82 | 812 |

Notes
(1) The heat rejected to the oil cooler is included with that to the jacket-water cooler.
(2): G3508B LE, G3512B LE, G3516B LE, G3520B LE, G3600, GCM34 and G3300B engine information based on 0.5 gram NOx rating
(3): All G3300, G3400NA, G3400TA and G3500 engine information based on catalyst setting
(4) Performance data is based upon the A2 version at High Efficiency setting and reference conditions in accordance with ISO 3046/1-6
(5) tolerance 10%
(6) Jacket water circuit (HT circuit) includes jacket and HT charge air cooler heat
(7) LT charge air cooler portion only

**FIG. 15-36**
**Example Engine Heat Recovery Arrangement**




## Gear Ratings

Various parameters affecting the durability, strength ratings, and scoring temperatures include:

**Horsepower** — The horsepower rating of high speed gears is determined from the durability rating and strength rating on the gear or pinion as specified in AGMA 6011-G92, Specifications for High Speed Helical Gear Units. In addition, the rating is limited by the scuffing temperature as determined in accordance with AGMA 217.01, Information Sheet, Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears, and Annex A of AGMA 2001-C95.

**Durability** — The durability hp rating for a specific gear set is primarily dependent on the speed and allowable contact stress of the gear and does not vary significantly with tooth size. The allowable contact stress is dependent on the surface hardness of the gear or pinion tooth and varies with material composition and mechanical properties.

**Strength** — The strength hp rating for a specific gear set varies mainly with the speed, allowable fatigue stress of material, and with tooth thickness. The tooth form, pressure angle, filet radius, number of teeth, helix angle, and pitch line velocity also affect the strength horsepower rating.

Allowable fatigue stress is dependent on the tensile strength of the material and varies with heat treatment and chemical composition.

**Scuffing Temperatures** — The scuffing or flash temperature index is the calculated temperature of the oil in the gear mesh. This temperature is arrived at by calculating the temperature rise of the lubricant in the mesh and adding it to the inlet oil temperature. The temperature rise for a given set of gears increases with the tooth loading, speed, and surface finish (i.e. increasing roughness).

The temperature of the gears will increase as the pitch is decreased, pressure angle is decreased, or helix angle is increased.

## Design Factors

The following design factors must be considered for high speed drives.

**Housings** — Must be of rugged design for strength and rigidity to maintain precise alignment of gears and bearings.

**Bearings** — Should be split-sleeve, babbitt lined, steel-backed precision journal bearings with thrust faces for axial loads. Fixed pad or tilting pad (Kingsbury type) should be used where required. Tilt pad radial bearings may also be required for high rpm, high load applications.

**Shafts** — Precision machined from heat treated, high quality alloy (4140 is common) steel. Adequately sized to rigidly maintain gear alignment and protect from overload.

**FIG. 15-37**
**Gear Quality**

| Pitch Line Velocity (ft/min) | Surface Quality (RMS) (micro inches) | Minimum Gear Quality Number |
|---|---|---|
| Under 8,000 | 45 | 10 |
| 8,001 – 10,000 | 32 | 11 |
| 10,001 – 20,000 | 32 | 12 |
| 20,001 – 30,000 | 20 | 12-13 |
| Over 30,000 | 16 | 12-14 |

**Pinions** — Normally cut integral with shaft from a high-quality forging that is through hardened or surface hardened by carburizing or nitriding. Grinding is the most common finishing method but precision hobbing, shaving, or lapping are also used.

**Gears** — Usually made from a high-quality forging that is through hardened or surface hardened by carburizing or nitriding and is separate from the low speed shaft. Gear may be integral with the shaft when operating conditions require. Grinding is the most common finish method but precision hobbing, shaving, or lapping are also used.

**Dynamic Balance** — Balance all rotating elements to assure smooth operation at high rpm.

**Seals** — Shaft seal should be of the labyrinth type, with clearance between shaft and seal of 0.020 to 0.030 inch. To prevent oil leakage through the clearance, the labyrinth is made interlocking with grooves machined in the cap to create air back pressure during rotation to retain the lubricant.

High speed gears are usually used on critical process trains where down time is quite costly and catastrophic failure must be avoided at all costs. Therefore, gear drives are becoming more and more instrumented. Optional monitoring equipment often specified by users include:

- Vibration probes and proximitors (to measure shaft vibration).
- Keyphasors (provide timing and phase reference).
- Accelerometers (measure casing acceleration).
- Direct reading dial type thermometers in stainless thermowells (measure bearing temperature).
- Resistance temperature detectors (RTDs) and thermocouples (measure bearing temperature).
- Temperature and pressure switches (alarm and shutdown functions).

## Lubrication

The majority of high horsepower, high speed gears are lubricated from a common sump which also lubricates the driving and the driven equipment. These systems are normally designed to operate with a high-grade turbine oil with a minimum viscosity of 150 SSU at 100°F. A good operating pressure range for the oil is 25 to 50 psi, with 25 micron filtration.

## Couplings

A coupling is required to connect a prime mover to a piece of driven machinery. The purpose of a coupling is to transmit rotary motion and torque from one piece of machinery to another. A coupling may also serve a secondary purpose such as accommodating misalignment of the two pieces of equipment. There are two general categories of couplings: rigid and flexible.

**Rigid Couplings** — Rigid couplings are used when the two machines must be kept in exact alignment or when the rotor of one machine is used to support the rotor of another machine. Very precise alignment of machine bearings is necessary when using this type of coupling. Manufacturing tolerances are also extremely important. One common application for rigid couplings is in the pump industry where the prime mover, generally an electric motor, is positioned vertically above the pump.

**Flexible Couplings** — Flexible couplings, in addition to transmitting torque, accommodate unavoidable misalignment between shafts. Mechanically flexible couplings provide for misalignment by clearances in the design of the coupling. The most common type of mechanically flexible coupling is the gear type. Material flexible couplings use the natural flexing of the coupling element to compensate for shaft misalignment. Metal, elastomer, or plastic having sufficient resistance to fatigue failure may be used for the flexing element of the coupling. Many types of flexible couplings are in common use and selection for a particular application depends on many factors including cost, horsepower, shaft speed, and reliability. A specialist should always be consulted for proper selection on any critical piece of equipment.

## Vibration Monitoring

The oldest and most basic type of vibration measurement involved the use of the human senses to feel and listen to a machine. The basic approach has not changed, just the method. It was always difficult to justify enough time for one individual, or a group of individuals, to acquire periodic measurements on a large number of machines. Also, with the advent of high speed, high performance machines, failures can occur faster than personnel can react. In addition, very subtle changes can occur over a long period of time, making it difficult to realize by the human senses, but still affecting the machine's mechanical stability and safety. Vibration monitoring is simply the full-time electronic measurement and monitoring of vibration levels from a given machine. Typically, the monitoring responds to the overall signal input from the transducer regardless of the source of vibration (in-balance, bearing wear, coupling problems, misalignment, etc.).

A typical vibration monitor provides two levels of alarm: alert and danger, that can be adjusted to fit the characteristics of a given machine. These set points have associated relays which can be connected to external audible or visual annunciators on the control panel. If the alert or danger set point is exceeded, the monitor and annunciator will alert operations and maintenance personnel of this event. Ideally, the alert alarm will indicate that the machine condition has changed significantly, but allow some discrete time before the machine is in a dangerous condition. For most applications, if the machine does reach the danger level of vibration and continued operation would probably result in machine failure, automatic shutdown is mandatory regardless of the time lag that has occurred between alert and danger signal.

There are three types of vibration sensors: (1) accelerometers, (2) velocity transducers, and (3) proximity probes. For most large critical machinery, and certainly for machinery with fluid film-type bearings, the important measurement to be made is rotor motion relative to the machine bearing or bearing support. For this application, the proximity probe transducer has proven to be the most reliable indicator of machinery malfunctions.

The proximity probe is a noncontacting transducer, typically installed on the bearing or bearing housing, and observes the rotor radial dynamic motion and position with respect to the bearing clearance. This same type of proximity probe can be used to measure axial position and vibration as well.

For machines which exhibit significant amounts of casing motion, it may be necessary to add to this system a seismic transducer measuring machine casing vibration. Some unique applications dictate that measurements are necessary in the high frequency region, where accelerometers are typically employed.

The American Petroleum Institute (API) has published a specification describing vibration monitoring systems, API 670, "Vibration, Axial Position, and Bearing Temperature Monitoring Systems."

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# Practical advice for analyzing, preventing cylinder valve failures

Key opportunities to improve the performance, longevity of the most frequent failing compressor component revolve around timing, failure patterns. By **Paul Modern**

Cylinder valves are the single most common cause of unscheduled reciprocating compressor shutdowns. As equipment operators try to extend maintenance intervals, the question is, "Why do valves fail more often than other compressor components?" The necessary follow-up question is, "How do we improve valve reliability?" This article intends to summarize the main causes of valve failure, recognizable failure patterns and key opportunities to improve performance.

## Failure susceptibility

First, though, why are valves so susceptible to failure? Valve design is often a series of conflicting requirements: reliability vs efficiency, multiple gas duties, multiple pressure duties, varying amounts of lubrication and so forth. Despite design requirements, however, the valve can only be optimized for one set of running conditions. A compressor valve is pressure and flow operated. Flow forces keep the valve seal element open against springs. Spring loads close the valve as the flow forces decay at the end of the compressor stroke. That closure timing is critical for valve life.

Now take a valve designed to run in hydrogen gas at 435 psi (30 bar) discharge in a refinery hydrocracking application. While the main valve duty is hydrogen, the compressor will also need to run on nitrogen – a much heavier gas – for catalyst regeneration. The mass flow and drag forces will be completely different with the nitrogen and delay the closing timing significantly, which will result in closing impacts at higher speeds.

AUTHORS CORNER

**PAUL MODERN is the chief engineer for valves and flow control at Cook Compression. He has more than 20 years of experience relating to valve products in reciprocating compressors. He has a degree in Aerospace Engineering from Bristol University and has also worked on turbocharger flow-based analysis.**

## Incomplete data

Further, the valve must perform in often-difficult regimes over many millions of opening and closing cycles. Even a slow-speed compressor running at 400 rpm would see 210 million valve events in one year. That's 24,000 events every hour, with two impacts per event, bringing component fatigue to the forefront of concerns.

Finally, data from the field is often incomplete, inaccurate or difficult to quantify. Valve design is an exact science, often performed with inexact information.

Given the difficulties of valve design and operating conditions, what can a valve failure tell us? Often, there are clues in the timing or patterns of failure.

FIGURE 1
**A broken outer edge on a valve plate is often an indicator of impact failure.**

## Valve failure timing

Accurate run logs and maintenance records are the first data points needed in valve failure analysis. The runtime for the valve that failed and how that compares to historical runtimes is critical knowledge. Start-up or very-low-run-hour failures should be investigated with a different thought process than failures that occur after many operating hours.

Compressor turnarounds have a habit of creating difficult environments for the valves. Pooled lubrication oil in discharge pockets, condensate liquid forming in cylinders, the introduction of inlet pipe scale and debris, nitrogen start-up runs, lack of initial run lubrication – all of these can cause failure

FIGURE 2
**A mottled witness pattern on the back face of a valve plate might indicate excessive opening impacts.**

FIGURE 5
**Valve guard breakage at the center stud and polishing on the clamp face due to clamping issues.**

even before the valve has a chance to run in its correct operating environment.

If a valve has been running well between maintenance intervals for many years and then failures start to occur, what changed? Has the duty or gas changed significantly? Has the quality of the valve supply or reconditioning changed? Has the lubrication rate been altered? Has the maintenance period been extended? Did something upstream or downstream of the compressor change? Accurate information about the operating history can help pinpoint a cause.

One particularly vexing suction valve problem on an air compressor in India turned out to be tree blossoms blocking an inlet filter at a certain time of the year. By tracking the timing of the failures, the root cause was discovered.

Pressure and temperature logs are useful data for determining whether a failure was instantaneous or the result of performance degradation over time. Inter-stage pressures and valve cap temperatures over time could indicate whether valves are leaking and accurate logs can be invaluable in determining a root cause.

The other key part of valve failure analysis is to examine the valves themselves for failure patterns. These will typically point to specific mechanical and/or environmental failure mechanisms that can then be addressed.

FIGURE 4
**Spring fatigue failure may result from excessive valve flutter cycles.**

FIGURE 3
**Circular wear patterns on the back of valve rings might indicate spinning caused by valve flutter.**

## Mechanical valve failure mechanisms

Chief among mechanical valve failure mechanisms is failure from closing and opening impacts. As noted earlier, valves will open and close millions of times in their lifespan. Each time they do, the sealing element impacts the seat and the guard at either end of the motion. Limiting the speed of these impacts is critical to valve life and is a function of the valve springs. Valves are designed dynamically in order to optimize spring loads for a particular duty. Poor spring choice can lead to late closure and high impact speeds, which almost always leads to failure.

Impact-related failures often manifest

FIGURE 6
**Fretting on the valve guide due to movement in the valve pocket.**

FIGURE 7
**Unloader top cap showing wear marks from the actuator.**

as broken outer segments on plate type valves or broken (often radially cracked) rings on ring-type valves. Impact issues may also be seen as a mottled witness pattern on the contact faces of the seal elements. If opening speeds are too high, there is potential for spring damage due to the over-travel of the spring in the pocket.

Another mechanical failure mechanism is valve flutter, caused by excessive spring forces for the specific duty. In this instance, pressure opens the valve as required, but the flow and drag forces are insufficient to keep the valve open against the springs. The spring loads close the valve early, which in turn reduces the flow area, increasing the forces and momentarily re-opening the valve. This flutter repeats, possibly many times for each normal valve event. If a valve normally opens and closes millions of times, a fluttering valve will multiply that figure and create an extreme wear and fatigue regime.

Valve flutter can often be seen as excessive wear on the valve guide or sealing elements. Springs may also fail due to fatigue from the extreme cycles. On ring-type valves, the rings may spin within the valve and cause further wear in a circular pattern on the back face.

FIGURE 9
**Piston ring debris caught in a compressor valve.**

## Clamps

Valve clamping is a mechanical failure mechanism that is simple to spot but often overlooked. Valves are held in place in the valve pocket by a cage and cover arrangement. If the clamping forces are too low or the clamp height of the valve is too far reduced due to reconditioning, then the valve can potentially move in the pocket. The hammering action of the valve would likely be noisy, and failure is usually quick due to the forces involved. Physical evidence includes fretting or polishing on the valve guide or clamp faces. Hanging guards may fail at the center stud, and in extreme cases, the seat or guard might fail at the clamping location.

On valves with finger unloaders, actuator adjustment introduces another potential failure mechanism. Actuators depress a valve unloader finger to hold the valve open during start-up or capacity control. Normally a specific gap must be maintained between the actuator spindle end and the

FIGURE 8
**Finger unloader wear on a valve plate.**

unloader top cap. The consequences of poor adjustment can be heavy wear to the unloader cap and, potentially, wear from the unloader fingers on the seal elements. This wear can cause localized fatigue points and eventual failure.

## Environmental causes of valve failure

The cause of valve failures may also be related to the operating environment: lubrication, gas type, liquid build-up and the presence of dirt or debris.

Any particulate or debris that passes through a valve is likely to become trapped between the moving seal element and the seat. Over the course of the repeated valve opening and closing impacts, this trapped debris can cause damage to the seat and sealing elements. With plastic seal elements, particles can become embedded in the seal, creating small leak paths that lead to increased temperatures and seal degradation. In extreme cases, the embedded particle can weaken the seal element. During visual inspection of a valve, debris may still be evident in the valve; otherwise, impact marks from the trapped debris may be visible on seats and seal elements. Dirt or abrasive media can also lead to other problems, such as blocked spring pockets. Dirt that mixes with lubrication oil can create a lapping paste that causes erosion of the valve internals.

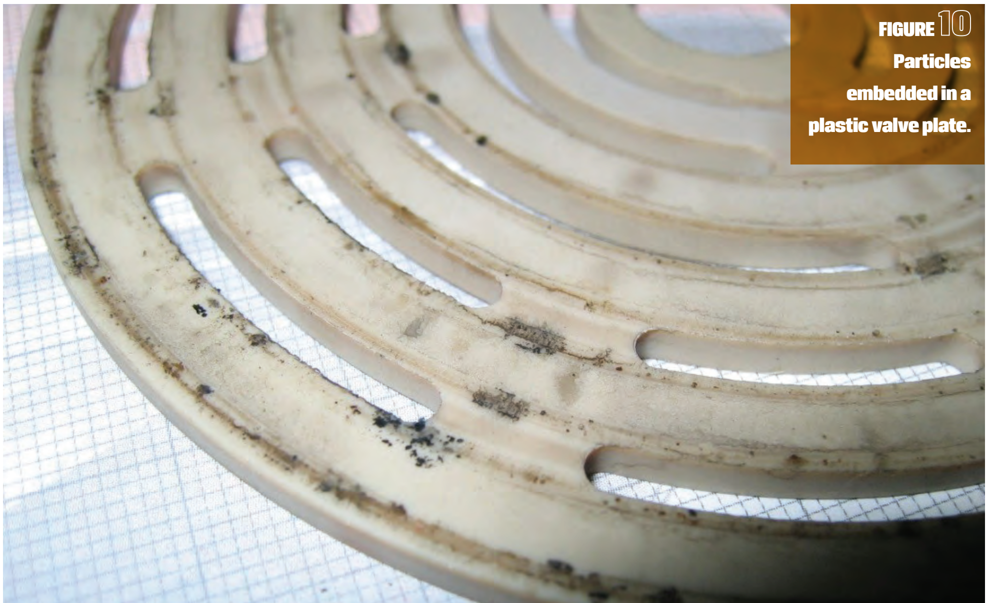

FIGURE 10
**Particles embedded in a plastic valve plate.**

FIGURE 11A
**Valve seat destroyed by a liquid slug; the top half of the seat was broken away, exposing the seal elements and springs.**

FIGURE 11B
**Simulated overpressure peak (circled) at the discharge valve opening on a PV curve.**

## Liquids

Liquids are an even more damaging contaminant in compressor cylinders, even in small quantities. When liquid forms from condensate or upstream process and gets trapped in pipework, pulsation bottles, inlet

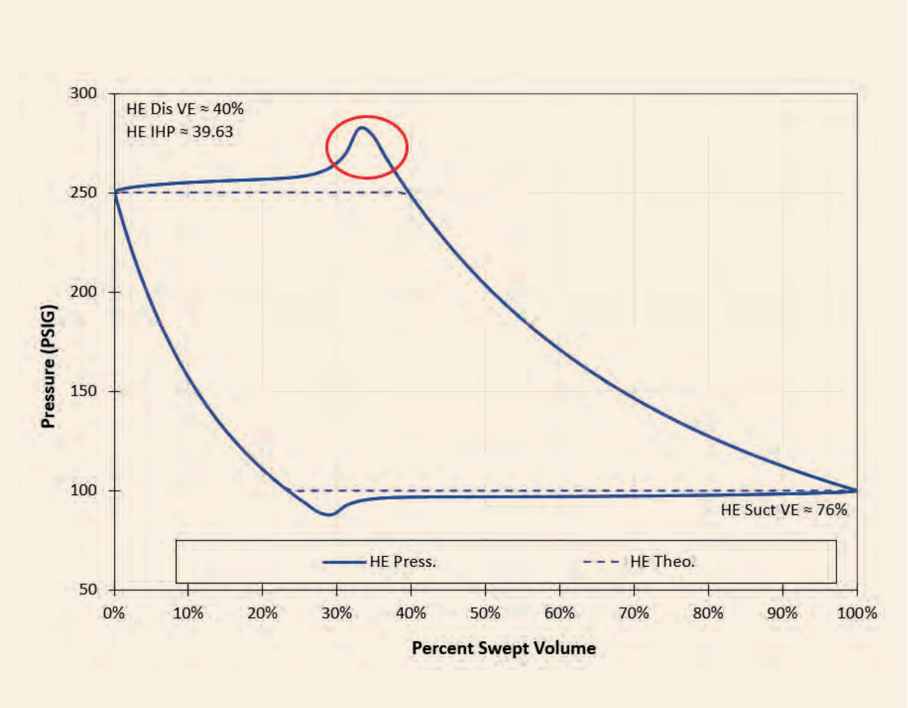


FIGURE 12

**Simulated overpressure peak (circled) at the discharge valve opening on a PV curve.**

separators, etc., it can then get sucked through the cylinder. This process is called liquid slugging. Excessive lubrication can also lead to oil pooling in the cylinder. In either case, the consequences for the valve are usually catastrophic. Liquid damage often manifests as failure in the center of the valve components, sometimes failing with a radiating pattern from the center. In less extreme cases, valve springs may collapse in the valve due to shock loading, which may lead to other impact failure modes. Despite the severity of valve damage, however, it is likely better that the valve fails instead of more expensive items within the compressor. Liquid events can go far beyond valve failures; they can cause failures of piston rods, connecting rods, or even cylinders.

## Excess oil

Valve failures may also result from more localized oil excesses within the valve itself. Excess oil in a valve has a suction effect, sticking the seal elements to the valve seat or guard and changing the opening and closing timing. The seal elements are held in place until pressure forces build up sufficiently to overcome the suction and move the elements. The effect is often impact-related failures from the increased opening and/or closing velocity, as valves either open with high overpressure or close late as spring forces build up. If PV analysis curves are available, the overpressure is often evident in the valve events. During valve inspection, oil will likely be visible on the valve surfaces, and excessive, burnt-on or sticky oil residues will be evident when the valves are disassembled. Anything greater than a light coating is problematic.

FIGURE 14

**Corrosion in metal plate valve due to extended compressor shutdown period in a PET air compressor.**

## Harsh gas

Harsh gas environments can further accelerate valve failures by attacking valve materials. Sour gases containing hydrogen sulfide can cause stress corrosion cracking in certain non-resistant metal grades. Specialty gases can react badly with poorly specified materials. Chlorine, for example, is extremely difficult when not bone dry, and it requires specific nickel alloys. Propylene may contain trace tetraethyl aluminum, which attacks PEEK. Knowledge of impurities

FIGURE 13

**Oil residue (sometimes in a burnt-on carbon form) is often evidence of excessive lubrication oil in a compressor valve.**

The system outlined here is the International System of Units (Systeme International d' Unites), for which the abbreviation SI is being used in all languages.

The SI system, which is becoming universally used, is founded on seven base units, these being:

| | | |
|---|---|---|
| Length | meter | m |
| Mass | kilogram | kg |
| Time | second | s |
| Electric current | ampere | A |
| Thermodynamic temperature | Kelvin | K |
| Luminous intensity | candela | cd |
| Amount of substance | mole | mol |

## POWER

The derived SI unit for power is the Watt (W), this being based on the SI unit of work, energy and quantity of heat – the Joule (J). One Watt (1 W) is equal to one Joule per second (1 J/s). One Watt is a very small unit of power, being equivalent to just 0.00134102 horsepower, so for engine ratings the kilowatt (kW) is used, 1 kW being equal to 1.341 hp and 1 hp being the equivalent of 0.7457 kW. The British unit of horsepower is equal to 1.014 metric horsepower (CV, PS, PK, etc.).

1 kW = 1.341 hp = 1.360 metric hp
1 hp = 0.746 kW = 1.014 metric hp
1 metric hp = 0.735 kW = 0.986 hp

## TORQUE

The derived SI unit for torque (or moment of force) is the Newton meter (Nm), this being based on the SI unit of force — the Newton (N) – and the SI unit of length – the meter (m). One Newton (1 N) is equivalent to 0.2248 pound-force (lbf) or 0.10197 kilogram-force (kgf), and one meter is equal to kilogram force (kgf) and one member is equal to 3.28084 feet (ft), so one Newton meter (1 N m) is equal to 0.737562 pound-force (lbf ft). or 0.101972 kilogram-force meter (kgf m).

1 Nm = 0.738 lbf ft = 0.102 kgf m
1 lbf ft = 1.356 Nm = 0.138 kgf m
1 kgf m = 9.807 Nm = 7.233 lbf ft

## PRESSURE AND STRESS

Although it has been decided that the SI derived unit for pressure and stress should be the Pascal (Pa), this is a very small unit, being the same as one Newton per square meter (1 N/m$^2$), which is only 0.000145 lbf/in$^2$ or 0.0000102 kgf/cm$^2$. So many European engine designers favor the bar as the unit of pressure, one bar being 100,000 Pascal (100 kPa), which is the equivalent of 14,504 lbf/in$^2$ or 1.020 kgf/cm$^2$, so being virtually the same as the currently accepted metric equivalent. On the other hand, for engine performance purposes, the millibar seems to be favored to indicate barometric pressure, this unit being one thousandth of a bar. Then again, there is a school that favors the kiloNewton per square meter (kN/m$^2$), this being the same as a kilopascal, and equal to 0.145 lbf/in$^2$ or 0.0102 kgf/cm$^2$.

1 bar = 14.5 lbf/in$^2$ = 1.0197 kgf/cm$^2$
1 lbf/in$^2$ = 0.069 bar
1 kgf/cm$^2$ = 0.98 bar

The American Society of Mechanical Engineers in 1973 published its Performance Test Codes for Reciprocating Internal Combustion engines. Known as PTC 17, this code is intended for tests of all types of reciprocating internal combustion engines for determining power output and fuel consumption. In its Section 2, Description and Definition of Terms, both the FPS and corresponding SI units of meas-urements are given.

## SPECIFIC CONSUMPTION

Fuel consumption measurements will be based on the currently accepted unit, the gram (g), and the Kilowatt Hour (kWh). Also adopted is heat units/power units so that energy consumption of an internal combustion engine referred to net power output, mechanical, is based on low unsaturated heat value of the fuel whether liquid or gaseous type. Thus the SI unit of measurement for net specific energy consumption is expressed: g/kWh.

1 g/kWh = 0.001644 lb/hph =
0.746 g/hph = 0.736 g/metric hph
1 lb/hph = 608.3 g/kWh
1 g/hph = 1.341 k/kWh
1 g/metric hph = 1.36 g/kWh

## HEAT RATE

Heat Rate is a product of Lower Heating Value (LHV) of Fuel (measured in Btu/lb or kJ/g for liquid fuel and Btu/ft$^3$ or kJ/m$^3$ for gas fuel) multiplied times (sfc) specific fuel consumption (measured in lb/hph or g/kWh).

**For Liquid Fuel**

Heat Rate (Btu/hph) = LVH (Btu/lb) X sfc (lb/hph)

**For Gaseous Fuel**

Heat Rate (Btu/hph) = LVH (Btu/ft$^3$) X sfc (ft$^3$/hph)

**To convert these units to SI units:**

Btu/hph X 1.414 = kJ/kWh
Or
Btu/kWh X 1.055 = kJ/kWh

## LUBRICATING-OIL CONSUMPTION

Although the metric liter is not officially an SI unit, its use will continue to be permitted, so measurement of lube-oil consumption will be quoted in liters per hour (liters/h).

1 liter/h = 0.22 Imp gal/h
1 Imp gal/h = 4.546 liters/h

## TEMPERATURES

The SI unit of temperature is Kelvin (K), and the character is used without the degree symbol (°) normally employed with other scales of temperature. A temperature of zero degree Kelvin is equivalent to a temperature of -273.15°C on the Celsius (centigrade) scale. The Kelvin unit is identical in interval to the Celsius unit, so direct conversions can be made by adding or subtracting 273. Use of Celsius is still permitted.

0 K = 273°C; absolute zero K
1°C = 273 K

## WEIGHTS AND LINEAR DIMENSIONS

For indications of "weight" the original metric kilogram (kg) will continue to be used as the unit of mass, but it is important to note that the kilogram will no longer apply for force, for which the SI unit is the Newton (N), which is a kilogram meter per second squared. The Newton is that force which, when applied to a body having a mass of one kilogram, gives it an acceleration of one meter per second squared.

"Weight" in itself will no longer apply, since this is an ambiguous term, so the kilogram in effect should only be used as the unit of mass. Undoubtedly, though, it will continue to be common parlance to use the word "weight" when referring to the mass of an object.

The base SI unit for linear dimensions will be the meter, with a wide range of multiples and sub-multiples ranging from exa ($10^{18}$) to atto ($10^{-18}$): A kilometer is a meter x $10^3$, for example, while a millimeter is a meter x $10^{-3}$.

To give an idea of how currently used units convert to SI units, the tables below give examples.

**KILOWATTS (kW) TO HORSEPOWER (hp) (1 Kw = 1.34102 hp)**

| kW | hp | kW | hp | kW | hp | kW | hp | kW | hp |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.341 | 21 | 28.161 | 41 | 54.982 | 61 | 81.802 | 81 | 108.623 |
| 2 | 2.682 | 22 | 29.502 | 42 | 56.323 | 62 | 83.143 | 82 | 109.964 |
| 3 | 4.023 | 23 | 30.843 | 43 | 57.664 | 63 | 84.484 | 83 | 111.305 |
| 4 | 5.364 | 24 | 32.184 | 44 | 59.005 | 64 | 85.825 | 84 | 112.646 |
| 5 | 6.705 | 25 | 33.526 | 45 | 60.346 | 65 | 87.166 | 85 | 113.987 |
| 6 | 8.046 | 26 | 34.867 | 46 | 61.687 | 66 | 88.507 | 86 | 115.328 |
| 7 | 9.387 | 27 | 36.208 | 47 | 63.028 | 67 | 89.848 | 87 | 116.669 |
| 8 | 10.728 | 28 | 37.549 | 48 | 64.369 | 68 | 91.189 | 88 | 118.010 |
| 9 | 12.069 | 29 | 38.890 | 49 | 65.710 | 69 | 92.530 | 89 | 119.351 |
| 10 | 13.410 | 30 | 40.231 | 50 | 67.051 | 70 | 93.871 | 90 | 120.692 |
| 11 | 14.751 | 31 | 41.572 | 51 | 68.392 | 71 | 95.212 | 91 | 122.033 |
| 12 | 16.092 | 32 | 42.913 | 52 | 69.733 | 72 | 96.553 | 92 | 123.374 |
| 13 | 17.433 | 33 | 44.254 | 53 | 71.074 | 73 | 97.894 | 93 | 124.715 |
| 14 | 18.774 | 34 | 45.595 | 54 | 72.415 | 74 | 99.235 | 94 | 126.056 |
| 15 | 20.115 | 35 | 46.936 | 55 | 73.756 | 75 | 100.577 | 95 | 127.397 |
| 16 | 21.456 | 36 | 48.277 | 56 | 75.097 | 76 | 101.918 | 96 | 128.738 |
| 17 | 22.797 | 37 | 49.618 | 57 | 76.438 | 77 | 103.259 | 97 | 130.079 |
| 18 | 24.138 | 38 | 50.959 | 58 | 77.779 | 78 | 104.600 | 98 | 131.420 |
| 19 | 25.479 | 39 | 52.300 | 59 | 79.120 | 79 | 105.941 | 99 | 132.761 |
| 20 | 26.820 | 40 | 53.641 | 60 | 80.461 | 80 | 107.282 | 100 | 134.102 |

**POUNDS FORCE FEET (lbf ft) TO NEWTON METERS (Nm) (1 lbf ft = 1.35582 Nm)**

| lbf ft | Nm | lbf ft | Nm | lbf ft | Nm | lbf ft | Nm | lbf ft | Nm |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.356 | 21 | 28.472 | 41 | 55.589 | 61 | 82.705 | 81 | 109.821 |
| 2 | 2.712 | 22 | 29.828 | 42 | 56.944 | 62 | 84.061 | 82 | 111.177 |
| 3 | 4.067 | 23 | 31.184 | 43 | 58.300 | 63 | 85.417 | 83 | 112.533 |
| 4 | 5.423 | 24 | 32.540 | 44 | 59.656 | 64 | 86.772 | 84 | 113.889 |
| 5 | 6.779 | 25 | 33.896 | 45 | 61.012 | 65 | 88.128 | 85 | 115.245 |
| 6 | 8.135 | 26 | 35.251 | 46 | 62.368 | 66 | 89.484 | 86 | 116.601 |
| 7 | 9.491 | 27 | 36.607 | 47 | 63.724 | 67 | 90.840 | 87 | 117.956 |
| 8 | 10.847 | 28 | 37.963 | 48 | 65.079 | 68 | 92.196 | 88 | 119.312 |
| 9 | 12.202 | 29 | 39.319 | 49 | 66.435 | 69 | 93.552 | 89 | 120.668 |
| 10 | 13.558 | 30 | 40.675 | 50 | 67.791 | 70 | 94.907 | 90 | 122.024 |
| 11 | 14.914 | 31 | 42.030 | 51 | 69.147 | 71 | 96.263 | 91 | 123.380 |
| 12 | 16.270 | 32 | 43.386 | 52 | 70.503 | 72 | 97.619 | 92 | 124.715 |
| 13 | 17.626 | 33 | 44.742 | 53 | 71.808 | 73 | 98.975 | 93 | 126.001 |
| 14 | 18.981 | 34 | 46.098 | 54 | 73.214 | 74 | 100.331 | 94 | 127.447 |
| 15 | 20.337 | 35 | 47.454 | 55 | 74.570 | 75 | 101.687 | 95 | 128.803 |
| 16 | 21.693 | 36 | 48.810 | 56 | 75.926 | 76 | 103.042 | 96 | 130.159 |
| 17 | 23.049 | 37 | 50.165 | 57 | 77.282 | 77 | 104.398 | 97 | 131.515 |
| 18 | 24.405 | 38 | 51.521 | 58 | 78.638 | 78 | 105.754 | 98 | 132.870 |
| 19 | 25.761 | 39 | 52.877 | 59 | 79.993 | 79 | 107.110 | 99 | 134.226 |
| 20 | 27.116 | 40 | 54.233 | 60 | 81.349 | 80 | 108.466 | 100 | 135.582 |

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in the gas stream can also significantly change material selection. Even air duty can be corrosive due to moisture content within the gas. Sometimes the effects of chemical incompatibility are easy to spot; other times, a microscopic investigation of the materials may be necessary to see the corrosion. Simple changes to material grades can have highly beneficial effects on valve life.

Finally, an often overlooked issue for valves is flow-induced pressure pulsations, which can affect valve timing and lead to impact-related failures. API 618 sets guidelines for maximum peak-to-peak pulsation levels. If impact-type valve problems persist, it may be worthwhile to perform a pulsation study of the compressor to check for abnormal levels and implement corrections.

FIGURE 15
Valve body corrosion due to moisture combining with corrosive elements in the gas.

## Giving valves a better chance

Valves have a hard-enough life when well specified and operating in a clean, low-oil, low-dirt environment. Add in corrosive gas, dirt, debris, sticky oil and pulsations and you have a recipe for failures in short order.

The first step in improving valve performance is to minimize environmental challenges for compressor valves. Minimize liquid carryover. Ensure inter-stage coolers and inlet pipework are designed correctly with no low spots where liquid can accumulate; ensure drain traps are checked regularly, even if it is an automated process; and ensure cylinder jacket temperatures are maintained 10 to 15 degrees above inlet gas temperatures to avoid condensation in cylinders.

Minimize sticky or dirty gas by ensuring that appropriate lubrication rates are maintained. The specified oil must also be compatible with the gas to avoid sludge or carbon build-up. Filtration or separation of water or dirt must be adequate at the compressor inlet.

Using the correct valve type for the application is also critical to compressor performance. Certain valve types perform better in harsh or dirty environments. High-speed applications require different valve characteristics. Material choices must suit the chemical and temperature environment. One size does not fit all.

Also, ensure that the valve supplier is aware of the full range of operating conditions and the amount of time spent operating at those set-points. Without that information, there is no way a designer can appropriately balance the valve characteristics for multiple duties or for varying pressures, gas mixes or rpm. Even small details such as the runtime with a clearance pocket open could potentially make a difference to the valve design.

Of course, the above list of failures is not exhaustive and multiple factors may combine to shorten valve life. In valve design and valve failure analysis, it is best to consult an expert who brings years of application experience. CT2

FIGURE 16
Process dropout in vinyl chloride gas.

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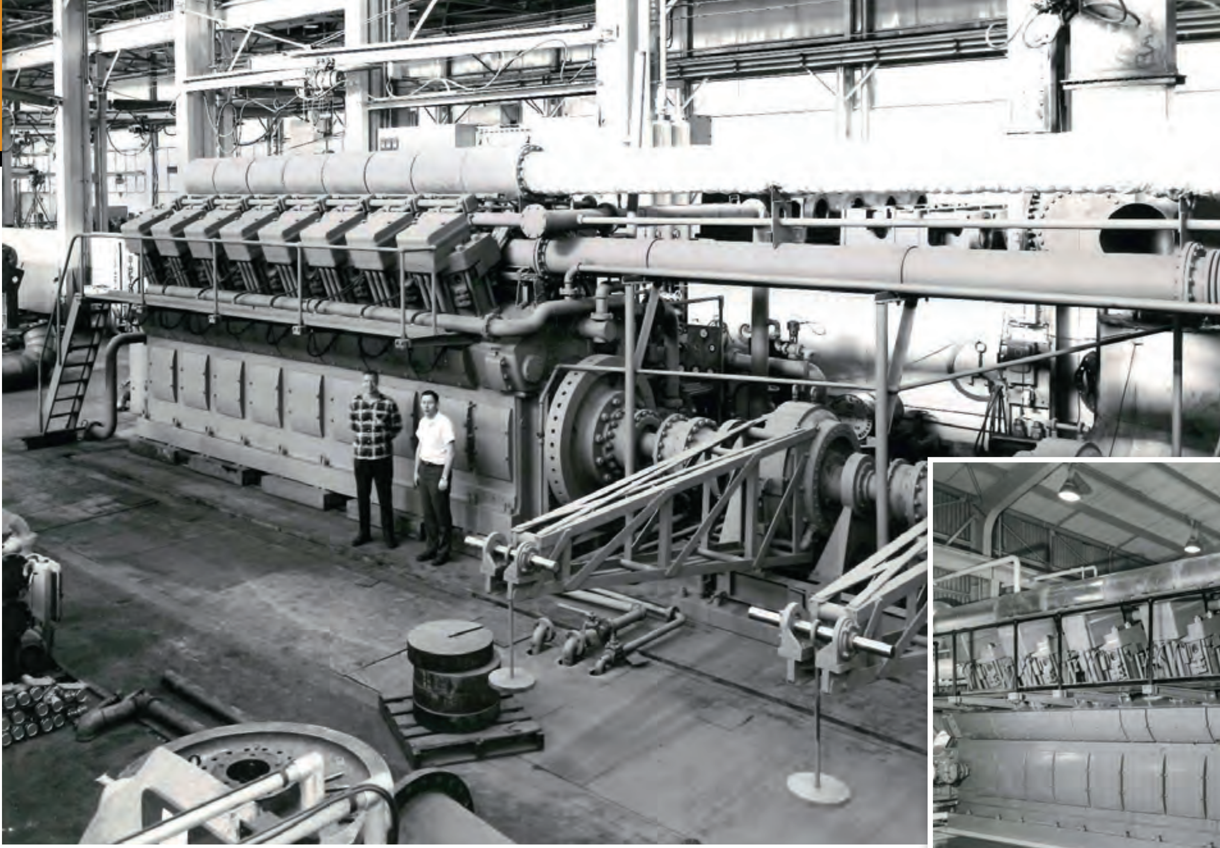

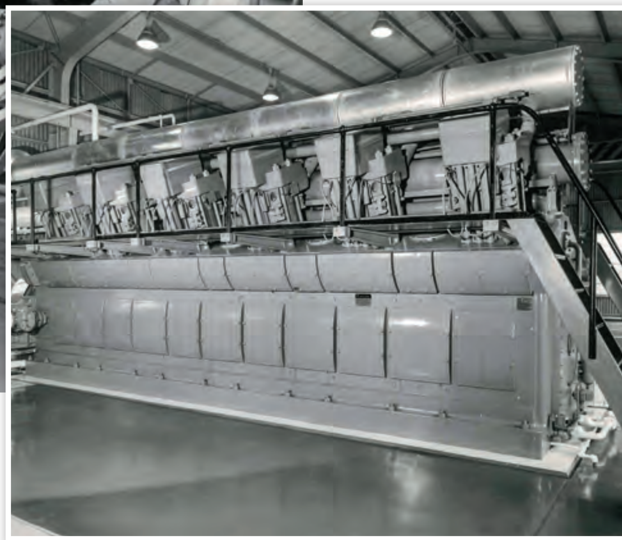

**Left: Shop test of the 16PKVT, January 1961. Bottom: The 512KVT, which operated at 3000 bhp at 330 rpm, March 1961.**

# KVT to KVTR revamp conversion

Work provided modern structure, less emissions.

By **T.M. (Mac) Sine** and **Jesse Burgey** of Siemens

The Ingersoll-Rand KVT integral gas engine-compressor was developed in the late-1950s in response to industry demands for higher horsepower and improved fuel economy. Ingersoll-Rand had introduced its first turbocharged engine series, the pulse turbocharged KVS engine, in the early 1950s. The 15.25 X 18.0 in. KVS engines are rated at 121 psi. BMEP at 330 rpm, utilizing the Buchi-system turbocharger configuration with Otto Cycle valve timing including a substantial scavenge period.

Production of the KVT engines began in 1959, and continued through 1967 when the KVT was replaced by the 17 X 22 in. KVR engine.

The KVT engine is rated at 135.7 psi. BMEP at 330 rpm (at 115o F. air manifold temperature) and was advertised as achieving a specific fuel consumption (SFC) of 6300 btu/bhp-hr.

To keep the KVT fleet in service for years to come, Dresser-Rand's gas engine engineers considered various strategies that could be applied on a KVT engine to improve engine operation plus reduce NOx emissions and ultimately concluded that for the KVT, what was needed was to improve its operating stability via improved air/fuel

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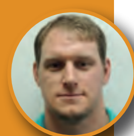
**MAC SINE** is a gas engine engineer at Siemens. He joined Dresser-Rand in January 1988 as a project supervisor and became a test and development engineer in 1990, whereby he supported the production of both two-and four-stroke-cycle, large-bore, slow-speed integral gas engine compressors. He also has experience with engine application engineering, engine revamp engineering, designing engine control systems, torsional critical speed analysis, conducting in-plant and site-specific customer engine schools and field support.



**JESSE J. BURGEY** is a senior mechanical project engineer at Siemens. Jesse began his career with Dresser-Rand in 2006 where he focused on leading gas engine emissions reduction retrofit projects as well as torsional analysis for both two-stroke-cycle and four-stroke cycle, large bore, slow-speed integral gas engine-compressor performance. In 2009, Jesse moved to Fort Collins, Colorado to work at the company's Gas Engine Technology Center. Here, he concentrates on emissions reduction retrofits for gas engines, automation and controls, application engineering and product development activities.

mixing. To do so, key combustion related design elements of the KVT engine were re-engineered; leveraged from the highly reliable and fuel efficient KVR engine, which also uses the Miller Cycle with constant pressure turbocharging. These elements include the cams, piston crowns, cylinder heads, and manifolds.

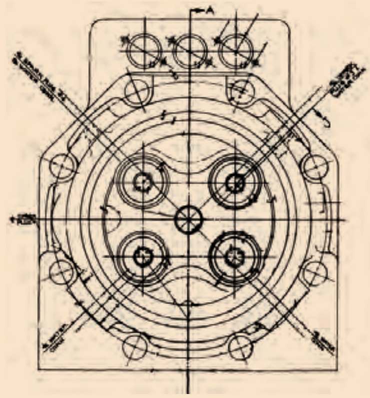
The extensive changes from the base KVT engine configuration lead to the adoption of the new engine model designator: KVTR. The KVTR engine design incorporates electronic port fuel injection, which provides advanced firing pressure balancing plus engine speed governing. Pre-combustion chamber (PCC) ignition is another standard KVTR feature, using either a bolt-in or screw-in PCC, depending on client preference. Making the change from a KVT engine into a KVTR engine constitutes a substantial revamp order that includes the assigning of a new engine serial number specific to the KVTR revamp engine series.

The base rating on the KVTR engine is 135.7 psi. BMEP at 330 rpm at 110 degrees F. air manifold temperature, at emissions levels of 1.5 g/bhp-hr NOx, 3.5 g/bhp-hr carbon monoxide (CO), and 1.35 g/bhp-hr non-methane hydrocarbon (N-MHC), at SFC of 6820 btu/bhp-hr (+/- 3%). When the KVTR revamp is applied on 512 and 616 KVT engines there is a potential for an increase in brake horsepower up to a 154 psi BMEP level. Additionally, with an appropriate turbocharger match, NOx reduction to below 1.5 g/bhp-hr can be achieved.

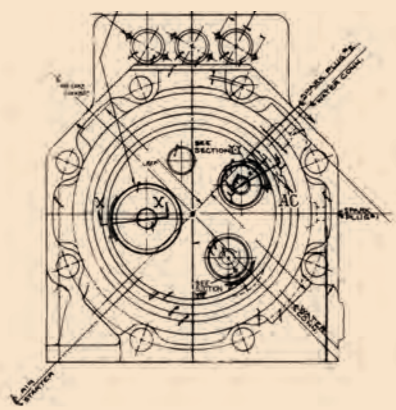
The KVTR revamp configuration began to be "roughed-out" in the 2003-4 period, during discussions with a Canadian client concerning what D-R would propose to enable their KVT engines located in the Province of Ontario to meet impending NOx emissions regulations.

## Cylinder heads

The KVT cylinder head departed from the "conventional" I-R design in that it is constructed with two intake valves and two exhaust valves. The dual-valve arrangement is not conducive to locating a tangent-converging inlet passage so, instead, the inlet passage provides a "down flow"



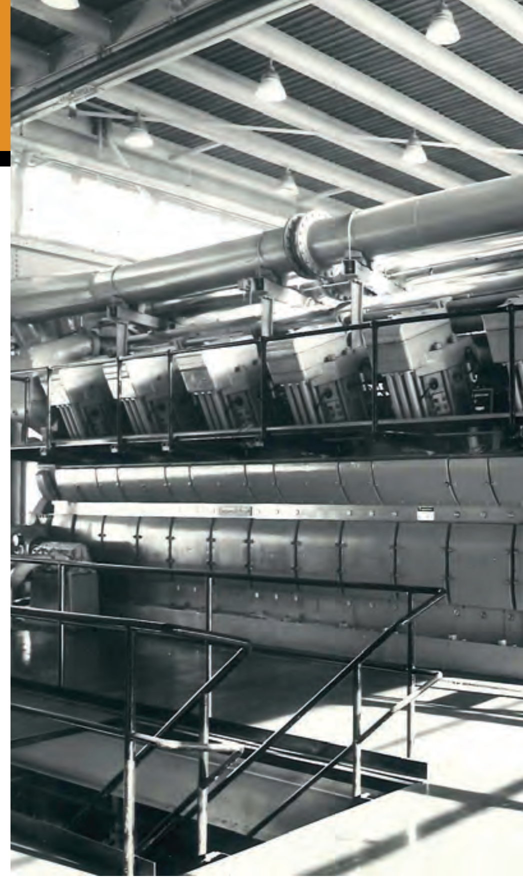
**Top: The KVT cylinder head.**
**Bottom: The KVTR cylinder head.**



arrangement at the valve seat counterbores that can impart a degree of tumble to the air fuel mixture within the cylinder. Tumble refers to charge motion about a horizontal axis, in contrast to swirl, which takes place about a vertical axis.

The combustion chamber in the KVT cylinder head is configured with two "squish bips" located opposite one another on the centerline of the combustion chamber. These can be seen in the accompanying picture of the combustion chamber side of the KVT cylinder head. The squish bips are used to remove clearance volume from the combustion chamber, in support of the design value of expansion ratio.

The original designer(s) of the KVT engine are to be credited with having the forethought of future expansion of the KVT frame into a higher-horsepower engine that ultimately became the KVR. For this reason, the KVT, KVH and KVR engines utilize the



same bolt circle diameter on the cylinder/cylinder head stud pattern. This became a key point in the definition of components that would be used in the KVTR configuration.

The cylinder head designs employed in the KVH and KVR engines are similar, but with a 1-in. difference in the combustion chamber diameter. Both heads utilize a single large intake valve in conjunction with the "fire nozzle inlet passage." One aspect that the KVT and KVH cylinder heads do have in common is the shape of the exhaust passage downstream of the two valve seat counterbores. A review of the exhaust passage shape revealed that it is necked-down at the juncture with the seat counterbore chamber and then opens into a relatively constant inside diameter extending to the manifold port face on the exterior of the cylinder head. Like the KVT and KVH, the KVR cylinder head design utilizes dual exhaust valves but, during the development of the KVR cylinder head, the exhaust passage was re-shaped into a relatively smooth diverging interior with no neck-down at the juncture with the seat counterbore chamber.

Two variations of the KVR cylinder head are in production: one utilizing a bolt-in PCC and the other a non-PCC design. The non-PCC design can be machined to accept

The pulse turbocharged 512KVH engine (3400 bhp at 330 rpm).

a screw-in PCC. These two variations were carried into the KVTR cylinder head design, therefore, the choice of a bolt-in PCC or a screw-in PCC is left to client preference.

## Pistons

The original-design KVT piston was evaluated for its suitability to be used in conjunction with the KVR-style cylinder head, with no change being made to the depth of the combustion chamber within the head and a determination was made that a change in the design of the piston would have to be made to achieve the KVTR target expansion ratio.


The KVTR two-piece piston assembly.

The majority of KVT engines were built with "single-piece" power pistons; that is, the piston crown and the piston skirt are not individual components that can be replaced separately. However, in the late-1960's, a 2-piece piston design was developed for the KVT engines and two KVT engines were built-new using this 2-piece piston assembly. Additionally, the three KVT engines at the Ontario, Canada, site had been retrofitted with the 2-piece pistons. Therefore, the decision was made to use the 2-piece piston design in the KVTR package, with a new piston crown being developed, specific to the KVTR engine.

Ultimately, the original-design KVTR piston crown came to be designated as the "A" crown. The "A" crown design has been superseded by a "B" crown design.

The piston skirt in the original KVT 2-piece design is made with a tin-plate anti-friction coating. The current KVTR piston skirt is made with a Molykote D-10 anti-friction coating. However, the older KVT tin-plated skirt can be mated with a new KVTR crown to make a piston assembly for a KVTR revamp engine.

The installation of 2-piece pistons during a KVTR revamp into an engine previously equipped with 1-piece pistons requires that a drilling modification be made to the small end of the connecting rods, to provide an oil spray hole directed at the interior cavity of the piston crown. This modification is detailed on a drawing that is provided to the client along with other revamp drawings and information.

## Cams, intake valve seats and air/fuel mixing

By the time that the KVTR engine concept was under discussion, Woodward Corp. was manufacturing their solenoid operated gas admission valve (SOGAV); an electronic-actuated FGIV intended to be used for fuel injection outside of an engine cylinder. Dresser-Rand had gained experience with the SOGAVs via the installation of "port-type" fuel injection systems on multiple KVG engines.

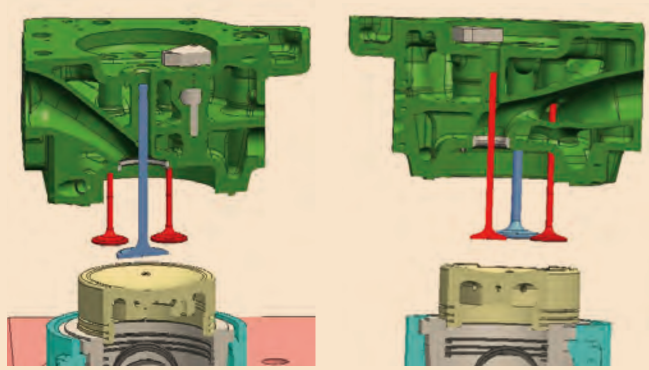
Building on the passage fuel injection testing experience from the 1980's, a decision was made to utilize "port-type" fuel injection on the KVTR engine, using the SOGAV hardware, in conjunction with the bias dam intake seat and the secondary dwell intake cams. This configuration of hardware had been previously tested on the K4X Laboratory Engine, in KVR-PCC configuration, in 2007. In this configuration, the SOGAV is installed on the air inlet elbow, between the air manifold and the cylinder head.

Ultimately, the design of delivery adaptor that yielded the lowest SFC during this K4X-KVR Port-4 testing was a design that came to be called the "false valve" adaptor. The "false valve" adaptor is based upon the geometry of a conventional poppet-type FGIV at full lift, providing a conical dispersion of fuel into the air stream within the air inlet elbow.

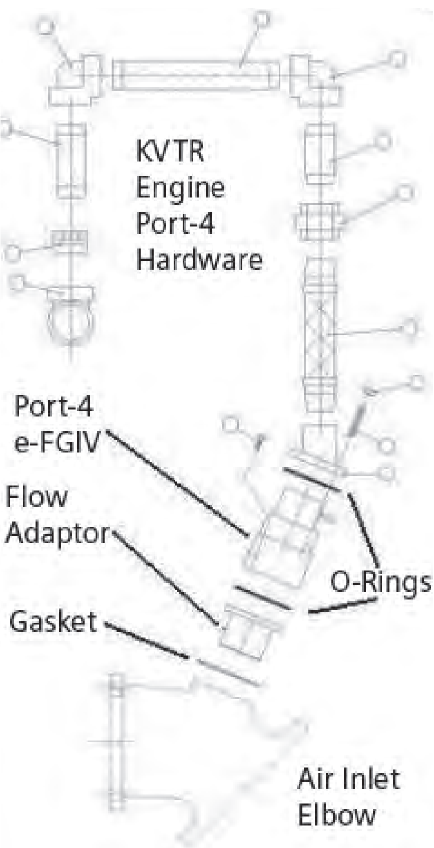
The "false valve" delivery adaptor that was designed for K4X Port-4 testing with the engine in the KVR-PCC configuration was sized in relation to the fuel flow needed to run the engine at the 80-deg. F rated KVR operating condition. During the design phase of the KVTR project, a new "false valve" delivery adaptor was designed specifically for the KVTR engine operating conditions.

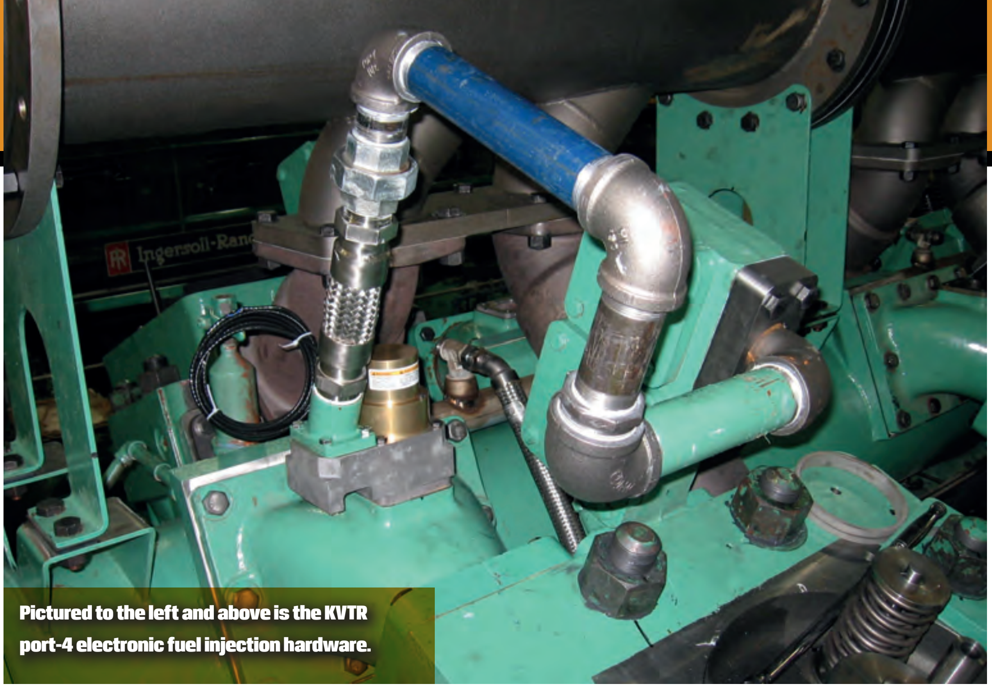
## Turbocharger and air aftercooler

The key to a successful low-NOx engine design is to apply the appropriate



Left: The KVTR cylinder head intake passage. Right: The KVTR cylinder head exhaust passage.



Pictured to the left and above is the KVTR port-4 electronic fuel injection hardware.

turbocharger to deliver the air required to serve as the diluent during combustion, in conjunction with the ignition energy needed to consistently ignite the lean main-chamber air/fuel charge. Accordingly, for the turbocharger, the decision was made to utilize a new high-ratio turbocharger within the KVTR package, rather than attempt to upgrade an existing KVT turbocharger. The turbocharger is supplied with a turbine wastegate valve driven by the engine control panel to achieve air/fuel ratio control via manipulation of turbocharger speed that, in turn, determines the engine air manifold pressure. The turbocharger is also equipped with jet assist acting on the blower wheel. The application of a low level of jet assist will help the engine to start faster and idle better, although the KVTR engines can be started and idled without the application of jet assist. Jet assist is also applied during loading of the engine, to mitigate "turbo lag" that is a characteristic of a constant pressure turbocharger system. The turbocharger is lubricated with oil routed from the engine oil pump, flowing through a set of filters that are dedicated to the turbocharger.

One aspect of the KVT combustion air system that can be eliminated during a KVTR revamp is the air pre-heater that is installed upstream of the blower inlet on the KVT turbocharger. The pre-heater was used as a means of establishing air/fuel ratio control by maintaining a constant blower inlet temperature and also to mitigate the tendency of the blower to reach a surge condition in low ambient temperatures. Elimination of the blower inlet pre-heater also means that the associated control hardware used for control of the coolant temperature serving the pre-heater can be removed.

## Fuel system

The size and location of the fuel manifolds on the KVTR engine are unchanged from the KVT engine, therefore, the original fuel manifolds remain in place. However, on a 410KVTR engine the original-size bank orifices within the fuel manifold assembly must be enlarged by drilling to a 1" diameter. On a 512KVTR, the fuel manifold bank orifice size remains unchanged from the KVT-original diameter.

The original jumper piping between the fuel manifold and the cylinder heads is replaced with a new arrangement, supplying the SOGAV on each air inlet elbow. Individual cylinder orifice plates are not used on the KVTR engines, with power cylinder peak firing pressure balancing being accomplished by electronically "trimming" the opening of the SOGAV at each cylinder.

Upstream of the fuel manifold, the KVT governor-operated gas control valve is removed and replaced by a simple spool piece. The Port-4 fuel system requires that the fuel gas manifold pressure be biased against the prevailing engine air manifold pressure, therefore, a bias-capable fuel gas supply regulator must be installed in the fuel gas supply piping upstream of the fuel manifold. On the KVTR revamp engines presently in operation, a 25 psig. bias above the engine air manifold pressure has proven to be an optimum setting. The supply point for the PCC fuel gas regulator (that is also a bias-type regulator) must be located upstream of the main bias regulator, in order to provide a constant supply pressure at the inlet of the PCC fuel gas regulator.

## Fuel consumption

The 410KVTR-102EP and later -103EP engines were equipped with the same "A" piston crowns as supplied for the 512KVTR-101EP engine. Following the conclusion of the K4X – KVTR piston crown development work, new "B" piston crowns were provided to replace the "A" crowns in both of these engines. The -103EP engine was the first to have the "B" piston crowns installed and when this change was made the engine readily achieved the guaranteed fuel consumption at the guaranteed exhaust emissions levels, as is shown by the following performance data:

## Problem with "B" piston crowns

Following the commissioning of the 410KVTR-104EP engine that was revamped with

# COMPRESSOR HORSEPOWER SELECTION CHART

(Brake Horsepower Per Million Cu. Ft.)

| SUCTION PRESSURE | DISCHARGE PRESSURE (PSIG) | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 25 | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 550 | 600 | 650 | 700 | 750 | 800 | 850 | 900 | 950 | 1000 | 1050 | 1100 | 1150 | 1200 | |
| 0 | 65 | 99 | 128 | 144 | 156 | 168 | 178 | 187 | 203 | 218 | 233 | 233 | 241 | 248 | 254 | 260 | 266 | 272 | 277 | 282 | 286 | 291 | 295 | 299 | 303 | 307 | 311 | 315 | 3 STAGE |
| 10 | 35 | 63 | 85 | 104 | 121 | 131 | 140 | 149 | 163 | 175 | 186 | 196 | 205 | 214 | 223 | 231 | 228 | 233 | 237 | 242 | 245 | 250 | 253 | 257 | 260 | 264 | 267 | 270 | |
| 20 | | 43 | 62 | 78 | 92 | 106 | 118 | 126 | 139 | 151 | 160 | 170 | 178 | 186 | 193 | 199 | 206 | 212 | 218 | 225 | 231 | 226 | 229 | 232 | 236 | 239 | 242 | 245 | |
| 30 | | 29 | 47 | 62 | 74 | 85 | 96 | 107 | 123 | 133 | 143 | 152 | 159 | 167 | 173 | 179 | 185 | 191 | 196 | 201 | 206 | 211 | 216 | 221 | 226 | 230 | 224 | 227 | |
| 40 | | | 36 | 50 | 61 | 72 | 81 | 90 | 107 | 121 | 130 | 138 | 145 | 152 | 158 | 164 | 170 | 175 | 180 | 185 | 190 | 194 | 198 | 202 | 206 | 210 | 214 | 218 | 2 STAGE |
| 50 | | | 26 | 41 | 52 | 61 | 70 | 78 | 93 | 106 | 119 | 127 | 134 | 141 | 147 | 153 | 158 | 163 | 168 | 173 | 177 | 181 | 185 | 189 | 193 | 196 | 200 | 203 | |
| 60 | | | | 32 | 44 | 53 | 61 | 69 | 83 | 95 | 108 | 118 | 125 | 131 | 137 | 143 | 148 | 153 | 158 | 162 | 166 | 170 | 174 | 178 | 182 | 185 | 188 | 192 | |
| 70 | | | | 25 | 37 | 46 | 54 | 61 | 74 | 86 | 97 | 109 | 117 | 123 | 129 | 135 | 140 | 145 | 149 | 153 | 157 | 161 | 165 | 169 | 172 | 176 | 179 | 182 | |
| 80 | | | | | 30 | 40 | 47 | 54 | 67 | 78 | 89 | 98 | 109 | 117 | 122 | 127 | 132 | 137 | 142 | 146 | 150 | 153 | 157 | 161 | 164 | 167 | 171 | 174 | |
| 90 | | | | | 24 | 34 | 42 | 49 | 61 | 72 | 81 | 91 | 100 | 109 | 116 | 121 | 126 | 131 | 135 | 139 | 143 | 147 | 150 | 154 | 157 | 160 | 163 | 166 | |
| 100 | | | | | | 28 | 37 | 44 | 55 | 66 | 75 | 84 | 92 | 100 | 109 | 116 | 120 | 125 | 129 | 133 | 137 | 141 | 144 | 148 | 151 | 154 | 157 | 160 | |
| 125 | | | | | | | 25 | 32 | 44 | 54 | 63 | 71 | 78 | 85 | 92 | 99 | 106 | 113 | 117 | 121 | 124 | 128 | 131 | 134 | 137 | 140 | 143 | 146 | |
| 150 | | | | | | | | 22 | 35 | 45 | 53 | 60 | 67 | 74 | 80 | 86 | 92 | 98 | 103 | 110 | 114 | 118 | 121 | 124 | 127 | 130 | 133 | 135 | |
| 175 | | | | | | | | | 27 | 37 | 45 | 52 | 57 | 60 | 71 | 76 | 82 | 87 | 92 | 97 | 102 | 107 | 112 | 115 | 118 | 121 | 123 | 126 | |
| 200 | | | | | | | | | | 30 | 38 | 45 | 52 | 58 | 63 | 68 | 73 | 78 | 83 | 88 | 92 | 96 | 101 | 105 | 110 | 113 | 116 | 119 | |
| 250 | | | | | | | | | | | 26 | 33 | 40 | 46 | 51 | 56 | 60 | 65 | 69 | 73 | 77 | 81 | 85 | 88 | 92 | 95 | 99 | 102 | 1 STAGE |
| 300 | | | | | | | | | | | | 23 | 30 | 36 | 41 | 46 | 50 | 54 | 58 | 62 | 66 | 69 | 73 | 76 | 79 | 83 | 86 | 89 | |
| 350 | | | | | | | | | | | | | 21 | 27 | 33 | 38 | 42 | 46 | 50 | 53 | 57 | 60 | 63 | 67 | 70 | 73 | 75 | 78 | |
| 400 | | | | | | | | | | | | | | | 25 | 30 | 35 | 39 | 43 | 46 | 50 | 53 | 56 | 59 | 60 | 64 | 67 | 70 | |
| 450 | | | | | | | | | | | | | | | | 23 | 28 | 32 | 36 | 40 | 43 | 46 | 49 | 52 | 55 | 58 | 60 | 63 | |
| 500 | | | | | | | | | | | | | | | | | 22 | 26 | 30 | 34 | 38 | 41 | 44 | 46 | 49 | 52 | 54 | 57 | |
| 550 | | | | | | | | | | | | | | | | | | 20 | 25 | 20 | 32 | 36 | 39 | 41 | 44 | 46 | 49 | 51 | |
| 600 | | | | | | | | | | | | | | | | | | | | 23 | 27 | 30 | 34 | 37 | 39 | 42 | 44 | 46 | |
| 650 | | | | | | | | | | | | | | | | | | | | | 22 | 26 | 29 | 32 | 35 | 38 | 40 | 42 | |
| 700 | | | | | | | | | | | | | | | | | | | | | | 22 | 25 | 28 | 30 | 33 | 36 | 38 | |
| 750 | | | | | | | | | | | | | | | | | | | | | | | 20 | 24 | 27 | 29 | 32 | 34 | |

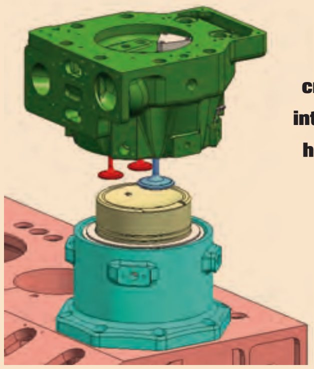
**NOTE: 1** MMSCFD MEASURED 14.7 AND 60°F NOT CORRECTED FOR COMPRESSIBILITY **2** "N"=1.26 **3** SUCTION TEMPERATURE 100°F **4** NATURAL GAS

the "B" piston crowns installed; the client reported that during a borescope inspection of the engine's combustion chambers, "nick marks" could be observed at one side of the valve head relief that is machined into the "B" piston crowns. The "nick marks" seemed to correspond with a small crescent of the outside diameter of the intake valve head. With the purpose of the valve head relief machined into the piston crown being to prevent contact between the intake valve head and the piston crown, receiving this report provoked considerable concern among the KVTR project team members. The "nick marks" were not observed in all cylinders and among the crowns showing "nick marks" the depth of each mark was not consistent from cylinder to cylinder; in what could be described as a random nick in the affected crowns.

Immediately following receipt of the client's report, the K4X – KVTR combustion chambers were examined again via borescope with special attention being focused on all surfaces of the piston crown notch in each of the two crowns and these borescope checks within the K4X combustion chambers revealed no evidence of contact between the intake valve head and the edge of the relief notch in the piston crown.

Next, one cylinder of the K4X engine was positioned with the piston at TDC and the intake valve was jacked-open to bring the valve head down into the piston crown relief. In this state, again viewed through the borescope, clearance could be observed between the O.D. of the valve head (at the margin on the valve head) and the wall of the piston crown relief, just as the original "B" crown design had intended.

Relative to the inconsistent degree of "coining-in" of the valve head on each piston crown that had been contacted by an intake



The KVTR "B" piston crown with intake valve head relief.

valve, the question was raised: Could the cylinder heads and/or the frame and/or frame top on the 410KVTR-104EP engine have a machining anomaly that has resulted in cylinder-to-cylinder differences in the height of each piston crown, at TDC, relative to the combustion chamber valve deck in the cylinder heads?

A checking tool was fabricated to measure the installed height at the center of the piston crown relative to a fixed reference point on the cylinder head and baseline measurements taken on the "B" piston crowns in the K4X engine. Next, the crown height checking tool was forwarded to the client site and measurements obtained in each cylinder of the 410KVTR-104EP engine. The measurements thus obtained were consistent with the measurements that had been obtained on the K4X engine, thereby ruling-out any irregularities in the mechanical configuration of the 410KVTR-104EP engine.

## Valve-to-crown contact cause

Within D-R Painted Post, investigation continued to identify the root cause of the valve-to-crown contact at the edge of the valve head relief. The 410KVTR-104EP engine was disassembled, and the "B" piston crowns, along with the never-installed set of "B" crowns for the -102EP engine, were returned to Painted Post for examination. Additionally, the never-installed set of "B" piston crowns that had been supplied for the 512KVTR-101EP engine was brought back to Painted Post.

In the I-R heritage two-piece piston design, the crown is retained to the skirt by six studs, arrayed in a specific pattern in the mating surface at the underside of the crown. The six-hole pattern is located relative to the piston pin centerline in the piston skirt. The placement of the six-hole pattern is defined on both the crown and skirt drawings. Because the location of the intake valve head relief in the "B" piston crown design is also placed in relation to the piston pin boss (in the piston skirt) centerline, the first check performed on all of the returned "B" crowns was to assess the placement of the valve head relief relative to the crown's bolt circle pattern. This was performed on an inspection table, using special-made studs in the piston crown, to permit picking-up of the same reference point on each crown. Every crown checked-out as having been made accurately to the drawing.

The CAD-UG model that had been utilized in the design of the "B" crown was reviewed; picked-apart and re-constructed by a designer who had not previously been involved with the KVTR project. No substantial findings were uncovered in this exercise.

One of the advantages of the two-piece piston crown design is that it provides the ability to replace a worn or damaged crown separate from the piston skirt; a skirt usually remains suitable for service over a greater number of operating hours than the crown. Typically the top piston ring grooves in the crown will reach replacement wear limits before the ring grooves in the skirt reach a wear limit. The three Canadian KVT engines had all been equipped with the two-piece design of KVT power pistons, long before the

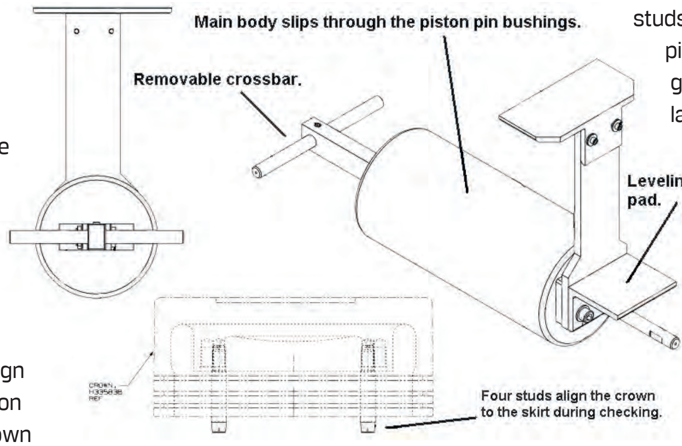
| ENGINE SN<br>DATE | 410KVTR-103EP<br>MARCH 4, 2011 | | | | | |
|---|---|---|---|---|---|---|
| Run Number | % Torque | AMP, psig. | NOx, g/bhp-hr | CO, g/bhp-hr | SFC, Btu/bhp-hr | Spark Timing, °BTDC |
| 2 | 97.9 | 12.3 | 0.94 | 3.41 | 6778 | 10.9 |
| 12 | 101.2 | 13.2 | 0.95 | 3.13 | 6668 | 10.8 |

KVTR revamp was conceived. As each of these engines was revamped, the KVT piston skirts were inspected and all skirts that were fit for continued service were matched with new KVTR piston crowns. It was the re-use of existing piston skirts that was ultimately found to be a contributor to the "random nick" crown contact incidents.

With the two-piece piston design for the KVT engines, the combustion chamber surface of the piston crown requires no "orientated" feature other than two threaded lifting eye holes that are placed on the piston pin centerline. The sequence of manufacturing steps employed to make both the crowns and the skirts was reviewed with personnel in the Power Piston Cell. It was this discussion that uncovered the root cause of the crown contact. The Ingersoll-Rand heritage designs for the two-piece pistons were completed in the mid-1960's, long before the advent of Computer Aided Manufacturing. To facilitate consistency in the repetitive operations of drilling the fastener holes in the piston crowns (threaded holes) and in the piston skirts (thru holes), drill jigs were constructed. In placing the drill jig prior to drilling the thru holes in the piston skirts, the drill jig was "eyeball aligned" with the center of the piston pin bores. Accordingly, the rotational orientation of the six-hole pattern for the crown-to-skirt fasteners was subject to some variation from the true centerline of the piston pin bores.

## Further examination

On a piston crown not having a machined relief that must also relate to the piston pin centerline, having a few degrees of "clocking" of the crown-to-skirt fastener hole pattern would not result in any substantial problem. The lifting eye holes would be clocked slightly from the actual piston pin centerline, but not to an extent that lifting the piston assembly would become problematic. Further examination of the group of piston crowns plus a skirt from the 410KVTR-104EP engine indicated that 7-degrees appeared to be a maximum amount of clocking that could be expected. The "B" piston crowns used in the development testing in the K4X engine had not shown any indication of clocking because their valve head reliefs had been machined into each piston crown after the crowns had been assembled to their respective piston skirts.




**Pictured is the KVTR "B" crown go/no-go gage.**

With the understanding of what had happened to promote the "random nick" contact, the KVTR team reached a decision to increase the width of the intake valve head relief to avoid contact between the intake valve head and the relief in the crown, on a piston assembly having up to 10-degrees of clocking. At this point, the "B" piston crown drawing was revised according to the revised CAD model.

Next, a go/no-go gage was designed to enable an assessment of the suitability for an existing KVT piston skirt to be matched to a new KVTR "B" piston crown. The go/no-go gage indexes in the piston pin bushings in the piston skirt and has a tang that must enter the valve head relief in the piston crown. If the tang will not enter the valve head relief in the piston crown, the piston skirt is rejected. The Ingersoll-Rand heritage designs for the two-piece pistons require that the piston crown be lapped to the piston skirt prior to installing the crown studs, followed by tightening the fasteners to complete the assembly. This can be a time-consuming process. The KVTR go / no-go gage assembly includes a set of special studs that are temporarily installed into the piston crown so that a crown can be gaged on a particular piston skirt prior to lapping and final assembly. In this way, if a particular skirt and crown pair are not suited because of clocking of the skirt's hole pattern greater than 10-degrees, the skirt can be rejected without any time spent lapping and assembling the actual crown-to-skirt studs and nuts.

To complete the remediation of this issue, all of the never-installed "B" piston crowns from the 410 and 512 KVTR engines were re-machined to increase the width of the intake valve head relief in the piston crown to a accommodate the maximum 10-degree clocking of the mounting hole pattern. The maximum 10-degree clocking value was chosen to limit the effect upon the expansion ratio (a decrease) resulting from the increased width of the valve head relief. New piston crowns, made from the revised "B" drawing were supplied to replace any crowns that had actually been contacted by an intake valve head.

During the subsequent KVTR piston assembly operations, the go/no-go gage was used successfully to assess the KVTR piston assemblies and identify any piston skirts having greater than 10-degrees of clocking in the crown-to-skirt stud hole pattern.

## Conclusion

In original configuration, KVT engine design features related to air, fuel, and exhaust transport influence the in-cylinder air, fuel and exhaust flows in a way that engine operational stability and the ability to consistently meet exhaust emissions reduction targets can be affected. The KVTR revamp was conceived to address these issues and enable modernization of the KVT engines. The revamp work scope is substantial when compared to typical emissions reduction projects, however, after modification, KVTR engines can provide reliable and emissions compliant horsepower for years to come. KVH engines will also benefit from the KVTR revamp for emissions reduction and modernization. **CT2**

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# Compressor foundation assessment and repairs key to reducing vibrations

Costly problems can be mitigated by optimizing multiple facets related to the foundation. By **Michael Golla**

Compressors are critical to many processes, and the foundations that support compressors need to be designed, assessed and repaired properly to minimize vibration and increase compressor reliability. Although machine bearings, misalignment or other mechanical issues can cause vibration, most vibration problems stem from the foundation. The foundation may have been designed improperly or deteriorated over time or damaged. Perhaps the compressor changed without determining if the existing foundation is suitable for the new weight and dynamic forces.

Many old foundations were not designed properly for vibration and have exceeded their design life. Vibration may also occur due to improper design and age. Sources of vibration can be identified with various techniques, and foundations can be repaired to reduce vibration from 50 to 100% in many cases.

This article will focus on reinforced concrete foundations that are commonly used to support compressors and absorb vibration and will discuss design, assessment and repair to reduce vibration.

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## Foundation design

For compressors, a reinforced concrete foundation typically consists of grout, concrete, anchor bolts, jack bolts and soil (see Figure 1). The compressor frame is typically bolted to a baseplate or soleplate attached to the grout and concrete foundation. Jack bolts, chocks or shims might be used at the anchor bolt locations to assist with alignment.

Old foundations built more than 40 years ago might not feature designs that can properly handle vibration based on today's standards. Without considering vibration, they might have been designed for static conditions using rules of thumb, such as foundation weight at three to five times compressor weight, with improper rebar design and spacing to minimize vibration forces. Additionally, grout, soil and anchor bolt material and designs have improved over the years to reduce vibration.

API 686 and ACI 351 provide good guidelines for foundation design (and repair) using modern standards and best practices. API uses the phrase "system" often to reinforce the importance of a unified foundation where all parts work together to minimize vibration.

## Grout design/installation

Precision grouts (for applied loads) are a combination of hydraulic cement, fine aggregate and other ingredients. These grouts are designed to uniformly transfer machine load and forces to the foundation. They help resist applied forces as concrete cannot do this alone. The grout also helps to minimize vibration by filling voids due to irregularities between the compressor frame or soleplates and the concrete. Grout is critical for compressor alignment too, and many vibration issues have been resolved by simply replacing

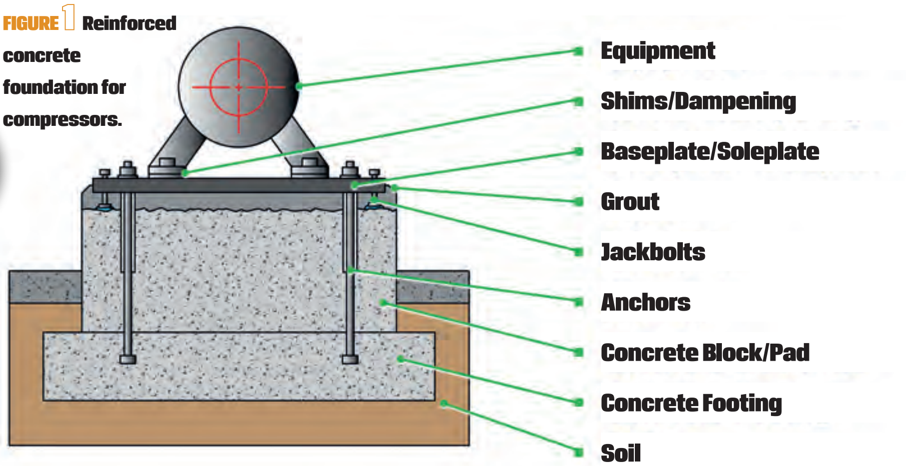


**FIGURE 1 Reinforced concrete foundation for compressors.**

grout and re-installing the compressor on the new grout for re-alignment. Grout-induced misalignment is a common cause of excessive vibration.

Grout technology has improved significantly over the years and there are many factors to consider when choosing a precision grout from the various manufacturers. Compression strength affects maximum support load. Creep resistance helps avoid misalignment and loss of bolt torque. Modulus of elasticity is a measure of stiffness and deflection of under load. The coefficient of thermal expansion is important for temperature change effects. Flowability and bearing area are critical for the baseplate contact to evenly distribute the load. The coefficient of thermal expansion and flowability is most important for rotating equipment.

In addition to choosing the proper grout, the grout installation process is also important. There are good guidelines and best practices for formwork, mixing, placement, expansion joints and others. Grout failures are common in the field and include design and installation issues. From a design perspective, point load cracking can occur due to sharp or square edges in the grout design, and grout can separate from the concrete if large grout shoulders are used, causing grout "edge lifting" (see photo #1). Additionally, grout can crack if expansion joints are not used or are not spaced properly. Poor installation can lead to a lack of bonding and voids. Grout should touch the steel above and the concrete below. Steel and concrete surfaces must be prepared properly per guidelines, and grout placement techniques by an experienced contractor must be used to ensure a good bond.

## Anchor bolt design/installation

Anchor bolts are another critical part of the foundation to resist forces and minimize vibration, and they are just as important as the grout. The grout prevents downward movement and the anchor bolts prevent upward movement. Concrete is strong in compression but weak in tension. Anchor bolts provide the tensile strength. Key considerations include the bolt material and fabrication, the type of termination, tension/torque relationships, allowed stretch, and adequate concrete cover. Many foundations designed prior to the 1980s do not follow today's standards for anchor bolts, potentially causing vibration issues (see Figure 2).

The importance of anchor bolts and termination design has been thoroughly researched in recent years, resulting in new standards for anchor bolt material, thread design and lengths that can also help minimize foundation cracking under stress. For example, "J" and "L" bolt designs, square plates and thin washers used in the past were not strong enough, resulting in concrete cracking and increased vibration. New designs are much improved. Round plates and plain nuts, instead of square plates and thin washers, help reduce tensile stresses caused by anchor bolts in the concrete. Extra rebar can be installed at the termination point of anchor bolts to provide extra strength and prevent cracking. Additionally, the metallurgy, number, diameter and length of anchor bolts are critical. Undersized anchor bolts were common on older machines. Bolts should now be as long as possible.

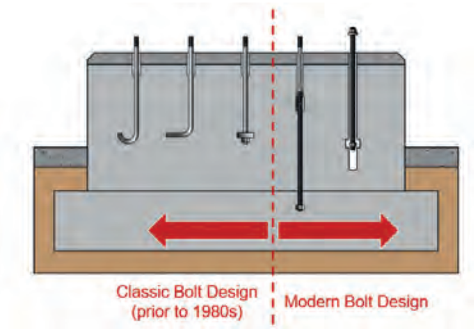


FIGURE 2 **Anchor bolt design evolution.**

## Foundation assessment

If the foundation is causing vibration or if machine vibration or other factors are causing deterioration of the grout and concrete, there are numerous methods for assessing the situation, especially for finding the source of vibration and designing the proper repair.

The foundation may need to be assessed and repaired for several reasons:

- Process-driven changes (change of equipment, etc.)
- Defects (design, materials, construction, past repairs, etc.)
- Deterioration (dynamic load effects or vibration, thermal effects, chemical attack, foundation settlement)
- Damage (equipment failure, impact, spills, etc.)

Assessment methods include a review of drawings and past repair methods; a site inspection of the foundation (concrete block, grout, alignment, anchor bolts, soil, crack patterns); and numerous non-destructive, semi-destructive and laboratory testing techniques.

Different techniques can be used to determine sources of vibration; rebar design (if drawings don't exist), crack depth in concrete; voids in concrete or grout; and the condition, composition and strength of concrete.

## Vibration

As mentioned earlier, the foundation is a frequent cause of vibration, due to foundation design or degradation over time. The foundation serves as a unified system with all parts working together to move vibration from the compressor down to the soil. If one part is not working

**Grout cracking and edge lifting due to excessive grout placement, lack of expansion.**

**Grout installation with expansion joints and anchor bolt sleeves.**

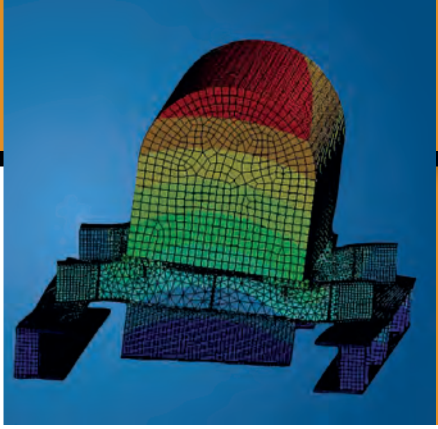

**FIGURE 3**
**ODS example of compressor motor vibration (without animation).**

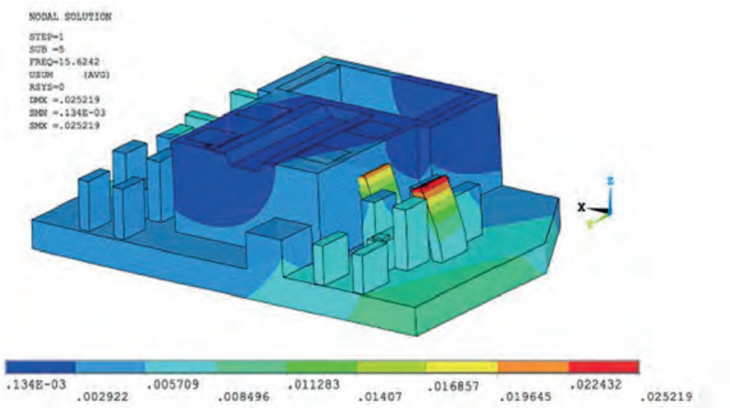


**FIGURE 4**
**FEA results to determine source of foundation vibration.**

properly, then vibration can occur.

## Vibration analysis

Various methods can be used to determine if the foundation is contributing to machine vibration.

Three other common vibration analysis methods, especially for compressor foundations, are operation deflection shape (ODS), motion amplification video (MAV) and finite element analysis (FEA).

ODS uses sensors to measure vibration at different locations on the foundation system at the compressor operating speed and frequency on a given day. Software is used to provide a 3D animation of the foundation movement (see Figure 3). It might be able to determine if a particular part of the foundation is vibrating excessively, such as a cylinder support pedestal, due to resonance, inadequate design, or deteriorated concrete.

MAV is a relatively new technology and uses a special camera and software to amplify and measure vibrations not visible to the human eye. Each camera view pixel becomes a vibration measurement sensor. MAV can sometimes identify loose anchor bolts, oil weeping from cracks, frame movement on grout (due to poor bonding) or other sources of vibration. It might also find low-frequency vibration missed by conventional vibration analysis methods.

MAV is much faster than ODS, and the resolution is more intuitive without the animated, cartoon-like image of ODS. In summary, MAV can be a cost-effective and quick method for finding sources of vibration, especially for reciprocating compressors that operate at lower frequencies than types of turbomachinery.

In general, ODS can be used for complex machinery issues requiring 3D views and trending, while MAV can be used for specific areas requiring high-spatial resolution.

FEA uses foundation drawings, compressor operating frequencies and dynamic forces, and concrete and soil condition data (if available) to create a theoretical 3D model of the foundation and then determine how the foundation reacts to various frequencies and forces. It doesn't require sensors or cameras and offers several unique benefits over other methods. For example, in addition to finding the source of the vibration without site work, FEA can be used to redesign the foundation/supports that are vibrating excessively.

Many compressors are routinely re-grouted without reducing vibration, and FEA might confirm that parts of the foundation need to be redesigned or repaired to reduce vibration, increase bearing life, reduce maintenance costs and increase the reliability of the machine.

## Case study

Here is an example of using a licensed, specialized contractor to provide high-quality assessment, repair design and repair methods to fix a compressor foundation and reduce vibration.

A four-throw, reciprocating compressor – built in 1970 with a conventionally reinforced concrete foundation and below-grade mat – experienced excessive vibration that caused frame movement at the grout interface and localized cracking in foundation components. Bearing life was reduced, causing unexpected shutdowns and increased maintenance costs.

ODS confirmed that two cylinders (one side of compressor concrete block) were moving too much and FEA confirmed that two separate cylinder support piers (not attached to the block) were resonating at compressor operating frequency (see Figure 4) and also contributing to vibration. FEA was then used to redesign the supports to eliminate the resonance. The solution included strengthening and enlarging the pier foundations, connecting them together with traditional rebar and post-tensioned bolts. The foundation block was also rebuilt to remove cracks and restore strength.

Vibration dropped substantially, which improved bearing life and decreased maintenance costs.

## Summary

Compressor foundations can be very complex and inadequate foundations can suffer from vibration and decreased compressor reliability. All parts of the foundation are critical to performance. Assessment, design and repair quality are essential, and vibration reduction can be significant after using trained and experienced inspectors, licensed and specialized design engineers with equipment foundation experience, and knowledgeable repair contractors. CT2

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# PACKAGER GUIDE 2022

A listing of global compressor packagers, along with primary contact information, types of compressors offered and the capacity range of the packages they produce. If your company is missing from this listing, please contact keefe.borden@khl.com. A PDF of this listing is available on our website.

**www.compressortech2.com**

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|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| COMPANY | City | State/ Province | Country | Centrifugal | Reciprocating | Screw | Other | Min (HP) | Max (HP) | Min (kW) | Max (kW) | Contact Name | Title | E-Mail |
| 7 Compression | Tyler | Texas | USA | | x | | | 0 | 48 | 0 | 36 | Chris Forstik | CEO | chris.forstik@7compression.com |
| ABBY Services Inc. | Canonsburg | Pennsylvania | USA | x | x | x | x | 3 | 750 | 2 | 560 | D.W. Fulmer | Applications | sales@abbyservicesinc.com |
| ABC Compressors | Eibar | | Spain | | x | | | 5 | 6800 | 4 | 5000 | Lucio Arizaga | Oil & Gas Manager | larizaga@abc-compressors.com |
| Adicomp Srl | Isola Vicentina | Vicenza | Italy | | x | x | | 4 | 1350 | 3 | 1000 | Pietro De Faveri Tron | Managing Director | pietro.defaveritron@adicomp.com |
| Aerzen USA | Coatesville | Pennsylvania | USA | | | x | | 0 | 13,410 | 0 | 10,000 | Tim Grady | Lead Application Engineer Process Gas Division | tim.grady@aerzen.com |
| AG Equipment Co. | Broken Arrow | Oklahoma | USA | | x | x | | 20 | 10,000 | 15 | 7456 | Kent Bright | President | kbright@agequipmentcompany.com |
| Alegacy Equipment | Waller | Texas | USA | | x | x | | 50 | 10,000 | 37 | 7500 | Will Reyes | President | w.reyes@alegacy.biz |
| ANGI Energy Systems LLC | Janesville | Wisconsin | USA | | x | x | | 40 | 900 | 30 | 675 | Jared Hightower | Vice President, Sales | jhightower@angienergy.com |
| Applied Compression Systems | Cranbrook | BC | Canada | | x | x | x | 2 | 1000 | 1.5 | 745 | Mike Sanderman | Operations Manager | mike@appliedcompression.com |
| Arrow Engine Co. | Tulsa | Oklahoma | USA | | x | | | 15 | 550 | 19 | 410 | Shari Vanhooser | Director of Sales | svanhooser@arrowengine.com |
| INTEGRITY Compression LLC | Houston | Texas | USA | | x | | x | 20 | 8000 | 15 | 5965 | Tommy Balke | COO | tommybalke@integritycompression.net |
| Brahma Compression | Calgary | Alberta | Canada | | | x | | 5 | 400 | 4 | 298 | Monte Scott | Sales | monte.scott@foremost.ca |
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| Clauger-Technofrigo S.p.A. | Castel Maggiore | | Italy | x | x | x | | 0 | 13,412 | 0 | 10,000 | Silvana Bazzani | "Head of Sales Process, Gas and Efficiency " | sbazzani@clauger.net |
| Cobey Inc. | Buffalo | New York | USA | x | x | x | | 0 | 30,000 | 0 | 22,065 | Eric McKendry | Vice President | cobey@cobey.com |
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| Com-Pac Systems Inc. | Odessa | Texas | USA | | x | x | | 25 | 4000 | 19 | 2983 | Jack Motley | President | jackmotley@compressorpackaging.com |
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| Compass Manufacturing | Oklahoma City | Oklahoma | USA | | x | x | | 70 | 8000 | 50 | 6000 | | | compass@chk.com |
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| Dearing Compressor & Pump Co. | Youngstown | Ohio | USA | x | x | x | x | 5 | 10,000 | 3 | 7457 | Richard H. Dearing Jr. | President | rick@dearingcomp.com |
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| Elliott Co. | Jeannette | Pennsylvania | USA | x | x | x | x | 100 | 120,000 | 74.5 | 89,500 | Mike Giunta | Sales Manager | mgiunta@elliott-turbo.com |
| Enerflex Ltd. | Houston | Texas | USA | | x | x | | 0 | 10,000 | 0 | 7460 | Aaron York | Director, Sales, United States of America | ayork@enerflex.com |
| Enerproject SA | Mezzovico | Ticino | Switzerland | | x | x | x | 30 | 4700 | 20 | 3500 | Vito Notari | Business Development Manager | vito.notari@enerproject.com |
| Euro Gas Systems SRL | Targu Mures | Mures | Romania | x | x | x | | 200 | 21,500 | 150 | 1600 | Roger Wachter | General Manager | roger.wachter@eurogassystems.com |
| FIMA Maschinenbau GmbH | Obersontheim | | Germany | x | | | | 1 | 6800 | 1 | 5000 | Michael Loercher | Sales Engineer | m.loercher@fima.de |

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| **Flatrock Compression Ltd.** | Houston | Texas | USA | | x | | | 26 | 500 | 19 | 373 | Brian McDonald | President | brian.mcdonald@flatrockcompression.com |
| **Flogistix** | Oklahoma City | Oklahoma | USA | | | x | | 20 | 800 | 15 | 597 | Drake Andarakes | Vice President, Sales and Marketing | sales@flogistix.com |
| **FLSmidth Inc.** | Bethlehem | Pennsylvania | USA | | | | x | 10 | 500 | 7.5 | 375 | Brian Warmkessel | Market Manager | brian.warmkessel@flsmidth.com |
| **Foremost** | Calgary | Alberta | Canada | | x | x | x | 5 | 400 | 3.7 | 298 | Steve Moe | Sales Manager | steve.moe@foremost.ca |
| **Gas & Air Systems, Inc.** | Hellertown | Pennsylvania | USA | x | x | x | x | 1 | 2000 | 1 | 1500 | Stephen St. Martin | President - General Manager | sestmartin@gasair.net |
| **Gas Compressors Ltd.** | London | | UK | x | x | x | x | 7 | 20,000 | 5.5 | 15,000 | Tony Silk | Head of Sales | info@gascompressors.co.uk |
| **GEA North America** | York | Pennsylvania | USA | | | x | | 0 | 5623 | 0 | 4193 | Todd Kennedy | Process Systems Sales Manager | todd.kennedy@gea.com |
| **GRAF GASTECH** | Nonantola | Modena | Italy | | x | | | 16 | 750 | 22 | 1000 | Mario Mormile | Oil &Gas manager | m.mormile@grafspa.it |
| **Great Plains Gas Compression Holdings, LLC** | Hugoton | Kansas | USA | | x | x | | 30 | 5000 | 4 | 3729 | Gloria Pollok | Vice President, Sales and Marketing | gpollok@thegpgc.com |
| **HBR Equipamentos Ltda** | Sao Paulo | | Brazil | x | x | x | | 0 | 5000 | 0 | 3729 | Valdir Zuffo | Director of Operations | valdir.zuffo@hbr.net |
| **Howden** | Renfrew | Scotland | U.K. | x | x | x | x | 0 | 20786 | 0 | 15500 | Douglas Latta | Product Director | Screw.bareshaft@howden.com |
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| **Industrias Juan F. Secco S.A.** | Rosario | Santa Fe | Argentina | | x | | | 0 | 10,000 | 0 | 7457 | Osvaldo Calvo | International Business | ocalvo@secco.com.ar |
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| **Mayekawa USA Inc.** | Houston | Texas | USA | | | x | | 0 | 4500 | 0 | 3300 | | Sales | customerservice@mayekawausa.com |
| **McClung Energy Services** | Longview | Texas | USA | | x | | | 50 | 400 | 37 | 298 | | | aforbito@estiscompression.com |
| **Mitsubishi Heavy Industries Compressor International** | Houston | Texas | USA | x | | | x | 500 | 150,000 | 375 | 111,855 | Clayton Jurica | Sales Manager | Clayton jurica@mhicompressor.com |
| **Natural Gas Compression Systems Inc.** | Traverse City | Michigan | USA | | x | x | | 0 | 2500 | 0 | 1864 | Bill Jenkins | Vice President, Sales | jenkins@ngcsi.com |
| **Natural Gas Services Group, Inc.** | Midland | Texas | USA | | x | x | | 50 | 1500 | 37 | 1119 | Jim Hazlett | Vice President, Technical Services | jim.hazlett@ngsgi.com |
| **Neuman & Esser USA Inc.** | Katy | Texas | USA | | x | | | 0 | 40,000 | 0 | 30,000 | Scott DeBaldo | | scott.debaldo@neuman-esser.com |
| **NEXT Compression** | Calgary | Alberta | Canada | | x | x | x | 25 | 6000 | 18 | 4474 | Alyssa Guiltner | Sales & Marketing Manager | marketing@nextcomp.ca |
| **NG Metalúrgica S.A.** | Piracicaba | S o Paulo | Brazil | x | x | x | | 0 | 20,000 | 0 | 14,925 | Júlio Cella | Application and Sales Engineer | jcella@ngmetalurgica.com.br |
| **OTA Compression LLC** | Irving | Texas | USA | | x | | | 0 | 125 | 0 | 93 | Vickie L. Gage-Tims | Vice President, Sales | sales@otacompression.com |

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| PEB Engineers & Constructors | Zoetermeer | South (Z-H) | Netherlands | | x | x | | 26 | 9383 | 20 | 7000 | Duncan Naumann | Sales Manager, Gas Compressor Systems | dn@peb.nl |
| Propak Systems Ltd. | Airdrie | Alberta | Canada | | x | x | | 0 | 10,000 | 0 | 7456 | | | sales@propaksystems.com |
| PSE Engineering GmbH | Hannover | | Germany | | x | | | 100 | 10,000 | 75 | 7500 | Dirk Heyer | Division Manager, Compression Systems | info@pse-eng.de |
| Reagan Power & Compression Inc. | Broussard | Louisiana | USA | | x | x | | 0 | 10,000 | 0 | 7455 | Joe Bellon | Executive Vice President | jbellon@reaganpower.com |
| Ron Porter LLC | Carmel | Indiana | USA | x | x | x | x | 20 | 300 | 15 | 225 | Ron Porter | Founder | porter.ronald@comcast.net |
| ROOTS Systems, Inc. | Houston | Texas | USA | | | | x | 0 | 1341 | 0 | 1000 | Mike Elliyoun | Regional Sales Engineer, Americas | mike.elliyoun@roots-blowers.com |
| S&R Compression LLC | Tulsa | Oklahoma | USA | | x | x | | 0 | 400 | 0 | 298 | David Bellamy | President | dbellamy@sandrcompression.com |
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| SCFM Compression Systems | Tulsa | Oklahoma | USA | x | x | x | x | 50 | 10000 | 40 | 7000 | Happy Pendley | Sales | hpendley@scfm.com |
| SEC Energy Products & Services | Houston | Texas | USA | | x | | | 50 | 10,000 | 37 | 7457 | Brian Niessing | Senior Sales Manager | bniessing@sec-ep.com |
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| Shandong Kerui Compressor | DongYing | Shandong | China | | x | x | x | 50 | 10,000 | 37 | 7456 | David Ni | Executive Dep. General Manager | nidw@keruigroup.com |
| Siad Macchine Impianti S.p.A. | Bergamo | | Italy | | x | | | 13 | 11,900 | 10 | 8700 | Mauro Acquati | Compressor Division Sales Manager | mauro_acquati@siad.eu - c.c.: paola_piccinelli@siad.eu |
| Startec Compression and Process Ltd. | Calgary | Alberta | Canada | x | x | x | x | 5 | 8000 | 4 | 5965 | Will Van Den Elzen | Business Development Manager | wvandenelzen@startec.ca |
| Technical Thai Service Co., Ltd. | Muangrayong | Rayong | Thailand | x | x | x | x | 20 | 20,000 | 15 | 15,000 | Alex Westphal | Managing Director | alexwest@technicalgroup.com |
| UEC-Gas Turbines, JSC | Rybinsk | Yaroslavl Oblast | Russia | x | x | | | 5361 | 33,512 | 3998 | 24,989 | Igor Yudin | Managing Director | inbox@odk-gt.ru |
| UECompression | Henderson | Colorado | USA | | x | x | x | 50 | 7500 | 37 | 5592 | Greg Herman | VP Sales and Marketing | gherman@uecompression.com |
| VPT Kompressoren GmbH | Remscheid | NRW | Germany | | x | x | | 5 | 3500 | 5 | 3500 | Carsten Kollenbach | CEO | C.Kollenbach@vpt-kompressoren.de |
| Wasco Engineering Group | Singapore | Singapore | Singapore | x | x | x | x | 100 | 10,000 | 75 | 7500 | Madhana Gopal | Regional Sales Manager | madhana.gopal@wascoenergy.com |
| York Process Systems (Johnson Controls) | Waynesboro | Pennsylvania | USA | x | | x | | 100 | 15,000 | 75 | 11,200 | Robert Fahey | General Manager | robert.f.fahey@jci.com |