Our experience has proven that the interests of end users, dealers and Hycomp are best served when we approach an opportunity with engineering involvement from the outset. We follow an Engineer-to-Engineer (E2E) process as we work through the specifics of the application and compressor design. The following information provides an overview of gas compressor selection.

COMPRESSOR TERMS

Certain terms are commonly used in gas compression, and should be understood and used properly for effective communication.

Compressor Displacement

Compressor Displacement is based upon the actual volume swept (or displaced) by the piston(s) during a certain number of revolutions of the crank shaft. It does not take into account any efficiencies, and is based solely upon the piston diameter(s), the stroke of the compressor and the RPM.

Volumetric Efficiency

In a perfect compressor, the compressor would deliver an amount of gas equal to its displacement. However, since no compressor is perfect, it will always deliver less gas. The ratio of delivered cubic feet divided by the displaced cubic feet is known as the volumetric efficiency. This efficiency will decrease with increasing compression ratio. The gas being compressed will also have an effect upon the efficiency.

Absolute Pressure

Most gauges read in PSI which is technically pounds per square inch gauge or PSIG, meaning that the effect of the local atmospheric pressure has been removed from the reading. This pressure is approximately equal to 14.7 pounds per square inch at sea level. Absolute pressure is equal to the total pressure exerted, including the pressure due to the atmosphere, which is generally equal to the gauge pressure plus barometric pressure, giving a pressure known as pounds per square inch absolute or PSIA.

Compressor Capacity

The amount of gas a compressor discharges is generally known as the compressor capacity. Two of the more common ways to reference this capacity are as follows: The first is to state what volume the gas would occupy at a "standard" temperature and pressure. In the United States, this is 60°F and 14.7 PSIA, and the capacity may be referred to as standard cubic feet per minute (SCFM). The second reference is actual cubic feet per minute (ACFM), which is referenced to a specific pressure and temperature, often (although not necessarily) the inlet condition. At atmospheric inlet pressures, the difference between these two is very small, but at elevated suction pressures, the difference can become quite large. For example, a compressor with a suction pressure of 14.7 PSIG (29.4 PSIA) and 60°F would have a SCFM exactly twice its ACFM.

Compression Ratio

The compression ratio is the absolute discharge pressure (PSIA) divided by the absolute inlet pressure (PSIA). In a compressor at sea level with an ambient intake and 100 PSIG discharge, the absolute discharge pressure is 100+14.7=114.7 PSIA, while the absolute inlet pressure is 0+14.7=14.7 PSIA, for a compression ratio of 114.7 divided by 14.7, which is equal to 7.80.

Exponent of Compression

Each gas has a specific property known as the exponent of compression, known as the "n" value of the gas. This value is very close to the ratio of the specific heats of the gas. This property of the gas affects required horsepower, compressor discharge temperatures and volumetric efficiencies.

Stages of Compression

The number of stages in a compressor describes the total times a gas is taken to an intermediate pressure, cooled, and taken to a higher pressure inside the compressor. Generally, as the number of stages increase, the compressor delivers a larger amount of cooler gas with less horsepower. The Hycomp compressor line is standardized around single, two and three stage compressors. A single stage compressor delivers gas at the final discharge pressure in a single stroke of the piston. When the compression ratio in this single stage unit becomes too high, the temperature of the gas becomes excessive, and a two stage unit should be used. Multi stage compressors take gas into the larger first stage cylinder, compress it to an intermediate pressure, then pass it through an intercooler to bring the gas temperature back down. The gas is then passed into the smaller second stage cylinder for compression to final discharge pressure. The cycle is repeated for a third stage on three stage compressors.

COMPRESSOR SPECIFICATION

Specifying a gas compressor can be a complicated task. However, knowing what is required and what can be done will simplify this greatly. Hycomp has produced many styles and sizes of compressor that are not standard because of the variety of standard bore and crankcase sizes available. Combining this with air or water cooling, multiple staging and gas packing options allows Hycomp to meet your needs and not force you to meet ours.

There are three limiting factors to every compressor selection: horsepower, pressure and temperature.

Every crankcase will only handle a certain amount of horsepower. This protects the crankshaft, main bearings, journal bearings and wrist pin bearings, as well as keeping the lubricating oil cool. Associated with horse power is maximum allowable torque, which is dependent upon the speed of the compressor and the required horsepower.

The maximum pressure rating is the absolute maximum amount of pressure the compressor is structurally capable of handling. Exceeding this limit will put the user at risk of having cylinder, head or intercooler failures. This does NOT mean, however, that every application can generate this pressure. As the compression ratio increases, so does the discharge temperature and required horsepower.

The discharge temperature of the gas increases as the compression ratio increases, and is often the limiting factor of any gas compressor application. In continuous duty operation, the discharge temperature must not exceed 340°F. Continuous duty is defined as any duty cycle with a load period exceeding 30 minutes. Intermittent duty is defined as a load period of less than 30 minutes, with an unload time equal to or greater than the load cycle. In special situations, Hycomp has produced compressors meant to operate in excess of these temperatures. Contact the factory for further information.

DETERMINING NUMBER OF STAGES REQUIRED

The first step is to determine the discharge temperature based on the compression ratio and 'n' value of the gas. This can be approximated with the following formula:

$$Td = Ti\,(Pd / Pi)^{(n-1)/n} \tag{1}$$

Td = discharge temperature °R
(°R = °F + 460)
Ti = suction temperature °R
(°R = °F + 460)
Pd = discharge pressure PSIA
Pi = inlet pressure PSIA
n = exponent of compression.

Any time the theoretical discharge temperature exceeds 340°F, the compressor should be considered as intermittent duty unit only. If continuous duty is required and the compression ratio exceeds 5.5, a two stage compressor may be required. If the compression ratio does not exceed 5.5 in a two stage unit, unequal rod loading will result which may cause extreme vibration in two stage units. Once the compression ratio is raised above 5, the rod loads tend to even out and vibration is reduced.

Low 'n' value gases can sometimes be compressed to ratios far in excess of 5.5 without exceeding the 340°F limit. However, the volumetric efficiency may fall below 20%. In these cases, more stages of compression are required to bring the efficiency to an acceptable level.

Water-cooling may be applied in situations where the discharge temperature exceeds 340°F, but the compression ratio is below 5. Generally water cooling will increase the acceptable compression ratio to 7 times for single stage compression of air ('n' value = 1.4), or 14-15 for two stage compression of air. This will vary as the gas changes due to differing 'n' values. Contact the factory before making final decisions based upon the above formula.

CHOOSING FRAME SIZE

First the required piston displacement must be calculated. Start by finding the Actual Cubic Feet per Minute (ACFM), at inlet conditions. The following formulas maybe used to convert common capacity references to ACFM:

SCFM to ACFM

$$ACFM = \frac{(SCFM)\ (14.7\ psia)\ (Ti\ °F + 460°R)}{(Pi\ psia)\ (520°R)} \qquad (2)$$

MSCFD to ACFM

$$ACFM = \frac{(MSCFD)\ (14.7\ psia)\ (Ti\ °F + 460°R)}{(1.44)\ (Pi\ psia)\ (520°R)} \qquad (3)$$

Lb/Hr to ACFM

$$ACFM = \frac{(lb/hr)\ (Ti\ °F + 520°R)(0.1787)(4)}{(Pi\ psia)(MW)}$$

Where: Pi = suction pressure in psia
Ti = suction temperature in °F
MW = molecular weight of gas

Once the required ACFM is known, the piston displacement can be calculated from the following formula:

Piston Displacement = ACFM / VE (4)
Where: VE =Volumetric efficiency

The approximate volumetric efficiency can be determined from the charts (T105, T106). Once the piston displacement has been determined, refer to the specifications charts (T110) to determine compressor size and required speed.

PRESSURE RATINGS

Check the specifications charts (T110) to be sure the required discharge pressure will not exceed the rating of the machine. Remember this is the MAWP only. It does not mean the compressor is actually capable of delivering the pressure. The maximum pressure a booster can deliver is set by the maximum discharge temperature of 340°F, as well as the MAWP.

HORSEPOWER

Calculating an exact horsepower requirement for a particular application is exceedingly difficult. Power used by a compressor can be traced to four main sources:

- Gas compression
- Valve losses
- Frictional losses
- Slippage losses

The power required to compress the gas is the primary requirement of any compressor. This is the actual power required to raise the pressure of the gas from the suction to the discharge pressures. The following formula may be used to find the gas horsepower requirement for a single stage, polytropic compression process:

$$Gas\ HP = \frac{ACFM\ (n)\ (Pi\ psia)\ ((Pd/Pi)^{(n-1/n)} - 1)}{(229\ (n-1))} \qquad (5)$$

The horsepower for Hycomp's gas compressors can be approximated by calculating the gas horsepower based on the piston displacement and adding 30%. This works reasonably well when small horsepowers are involved, but becomes increasingly inefficient as the horsepower requirement increases. It is also inaccurate when the frictional losses exceed 30%, as in the case of a low compression ratio (CR<1.5) and elevated suction pressure. Contact the factory to have the motor properly sized for these cases.

Horsepower requirements can be approximated with the following formulas:

SINGLE STAGE: (6)
$HP = .0057\ (PD)\ (Pi)\ (n/(n-1))\ (CR^{(n-1)/n} - 1)$

TWO STAGE: (7)
$HP = .0057\ (PD)\ (Pi)\ (2n/(n-1))\ (CR^{(n-1)/2n} - 1)$

As power requirements increase, these approximations become more inefficient. Contact the factory for more accurate horsepower calculations.

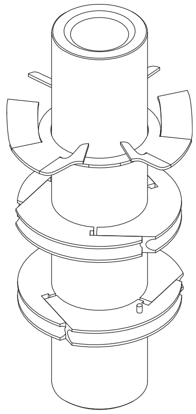

Figure 1. a) Tangent - Tangent pair of packing on piston rod

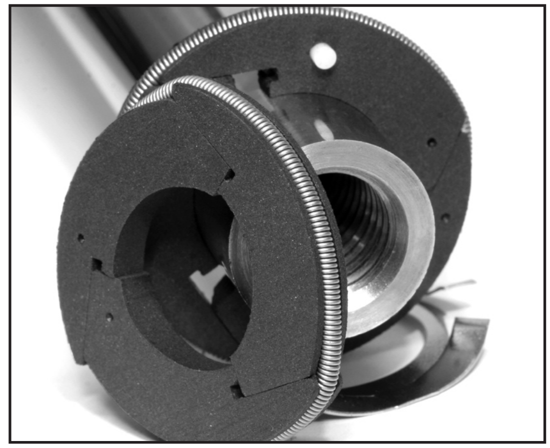

Figure 1. b) T-T packings on piston rod. Springs automatically adjust as packings wear.

ROD PACKING

Hycomp uses a proprietary segmental ring design to seal against gas leakage past the piston rods. Pairs of Tangent-Tangent packing rings are placed on the piston rod as shown (fig 1), sealing against the rod, the packing box and each other. The external force of the gas pressure keeps the packings sealed on the rod and against the packing cup, and prevents them from moving on the rod during operation.

By placing a gas plenum chamber directly below the compression cylinder, and plumbing the cool, incoming suction gas through this chamber, the pressure beneath the pistons (but above the packings) is kept at a pressure no higher than suction. This is not true of other manufacturer's designs, where the packings may see pressures as high as the discharge pressure. In addition, by bringing the cool suction gas through the plenum chamber, the packings are kept at a temperature no higher than suction temperature; again this is not true of designs that allow the packings to see discharge pressures because of the associated higher temperatures. The combination of our gas plenum design and the segmented ring gives an exceptionally long packing life, exceeding that of most competitors.

Since only Tangent-Tangent pairs of packings are used, the orientation of the packing within the packing cup is not important, eliminating the possibility of incorrect installation. Also, since the packing is a segmented ring, it is possible to replace the packing without disassembling the compressor. However, it is important that the compressor be configured properly for vacuum vs. booster service, as well as pad vs. vent/purge service. Every new machine is configured for the type of service expected, and it is nearly impossible to change this accidentally. Hycomp has three types of configurations for the gas service the packings will encounter:

Style 1:
Used for general transfer or boosting of industrial gases with a suction pressure above ambient. On the G & H series, the vent may be plumbed to an atmospheric dump or purged with an inert gas at a pressure below inlet but above 15 PSIA. It may also be plugged shut. When purged, there is virtually no chance of atmospheric air entering the process stream. When the purge gas is dumped to a safe area such as a burner, there is also very little chance of gas escaping to atmosphere. Standards packing options for this style are B201, B301, G211, G221, G321, H2211.

Style 2:
Used for general vacuum service with air or industrial gases, or when the pad gas is at a higher pressure than the compressor suction. For vacuum service the suction pressures are generally between 3 PSIA and 15 PSIA. On G & H series, the vent should be plumbed to the second stage suction or the discharge to keep a positive pressure in the vent chamber, preventing atmospheric air from entering the process stream. When high pressure padding is required, the vent should be plumbed to the pad gas source, and the compressor suction pressure may be any pressure below the pad gas pressure. Standard packing options for this style are: B202, B302, G212, G222, G322, H2212.

COMPRESSOR UNLOADING

A compressor may be unloaded by forcing the suction valve to stay in the open position. When the suction valve is held open, gas is forced in and out through the suction valve, preventing pressurization of the cylinder. Since no pressurization (and therefore no compression) is taking place within the cylinder, the load on the crankcase is reduced to inertial and frictional loads, which are very small compared to the gas load.

The unloader itself is a diaphragm actuated finger assembly. An unloader tower is attached to the cylinder head of the compressor, directly above the suction valve. When external pressure is applied to the top of the diaphragm, the diaphragm is forced down against a finger which forces the suction plate to stay open. When this external pressure is relieved from the diaphragm, it moves up, allowing the finger to move up releasing the suction plate and allowing the plate to move freely. A spring forces the finger up, ensuring it is clear of the suction plate. Because both sides of the diaphragm see pressure, the diaphragm actuating pressure must be at least 30 PSI above the suction pressure. To reload the compressor, the activation pressure must be bled from the unloader tower.

There are two main uses for unloading a Hycomp compressor. The first is to remove any load when the compressor first starts, before oil pressure is developed. This prevents damage to the crankcase components, such as bearings, crankshaft, wrist pins and crossheads. Hycomp compressors come with this style of unloading, also known as loadless start.

The second use for unloading a Hycomp compressor is to reduce the number of starts and stops on the compressor driver (motor). This is known as constant speed unloading. When a compressor is used to fill a volume such as a receiver, the pressure inside that volume will increase to the cutoff pressure, at which point the compressor shuts down. If the compressor must start and stop more than six times per hour, constant speed unloading should be installed. This will unload the suction valves when the compressor reaches the specified pressure, and reload the valves when the pressure drops by a specific differential pressure. This prevents constant starting and stopping, saving wear and tear on the motor, starter, compressor and drive components.