TUTORIAL ON CRYOGENIC TURBOEXPANDERS

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ABSTRACT

Cryogenic turboexpanders are used in many applications, including ethylene plants, refineries, gas processing plants, and air separation facilities. They frequently produce the coldest level of refrigeration available in the plant, and often operate with turbine outlet conditions where large quantities of liquid are condensed during the expansion process.

Turboexpanders have come a long way in the last 20 years. Their high efficiency and reliability have led to widespread use around the world. The use of magnetic bearings in many critical turboexpander applications such as ethylene plants has become common, and in the natural gas processing market, they are making inroads as well. Computational fluid dynamics (CFD) programs are providing windows into aerodynamic improvements and new applications that were unheard of in the past.

API 617 (2002) now addresses cryogenic turboexpanders in Chapter 4 of the Seventh Edition. Annex 4F of the same document provides guidelines for purchasing equipment using magnetic bearings, and this section is particularly applicable to turboexpanders. The upcoming Eighth Edition will continue this trend with further enhancements to the sections on turboexpanders and magnetic bearings.

This tutorial will attempt to broaden the knowledge of engineers regarding cryogenic turboexpanders, briefly covering both their design and application.

TURBOEXPANDERS

For those unfamiliar with turboexpanders, a brief description will be given. For the purposes of this paper, a turboexpander will be considered to be a radial inflow turbine connected to a load device (typically a centrifugal compressor) by means of a single rigid shaft. The radial and thrust bearings discussed herein are thus located between the expander and compressor impellers, as in Figure 1. The shaft speed of a typical turboexpander in hydrocarbon service is typically around 5,000 to 10,000 rpm for large machines and 50,000 to 100,000 rpm for small machines. In most applications, the turboexpander normally runs faster and operates with colder temperatures than any other equipment in the plant. Other forms of turboexpanders, such as those in which the expander drives a generator, are similar in design, as shown in Figure 2.

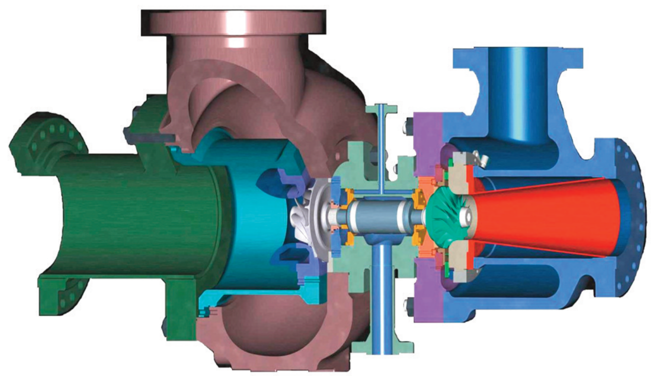

Figure 1. Typical Turboexpander.

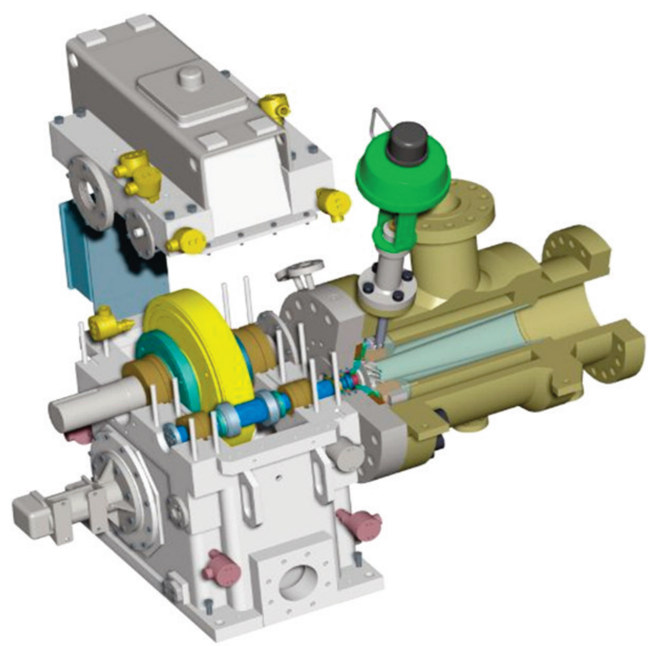

Figure 2. Typical Expander-Generator.

An interesting aspect of most turboexpanders is that the bearings, whether they are oil, gas, or magnetic bearings, typically operate in a filtered process fluid environment, and at pressures substantially above ambient. This means that most turboexpanders operate with a pressurized lube oil system, and that there is no high speed rotating shaft seal needed to prevent leakage of the process gas or oil to the atmosphere. In applications where there must be a high speed rotating shaft seal, such as expander generator systems that do not have the generator rotor immersed in the process fluid, then the shaft seal is generally put between the expander wheel and the expander bearing, so that the bearings and lube oil system can operate at atmospheric pressure.

While specific processes vary greatly, almost all turboexpanders are used to remove energy from a gas stream, thereby producing power and cooling the gas. The refrigeration effect of this cooling process is normally the main reason that the turboexpander is purchased. In addition, the main technical reason that a turboexpander is purchased with magnetic bearings is this eliminates the possibility that the cold box (heat exchanger) will be fouled with oil in the event of a major machine failure or operator error. This oil is difficult if not impossible to completely remove from the heat exchanger. The main commercial reason that magnetic bearings are used is that often the magnetic bearing system is cheaper! In the ethylene industry, for example, most new plants are purchasing magnetic bearing equipped turboexpanders.

In most instances, the inlet gas to the expander is very cold and at or near saturation. This means that the gas passing through the expander will not only get colder, but some of the heavier components will also liquefy. Usually, this liquid contains valuable product that is recovered as a result of the condensation process, making the plant more efficient. The amount of this liquid can vary from about 0 to 50 percent of the inlet stream (weight percentage basis), depending on the process conditions. A typical process is shown in Figure 3.

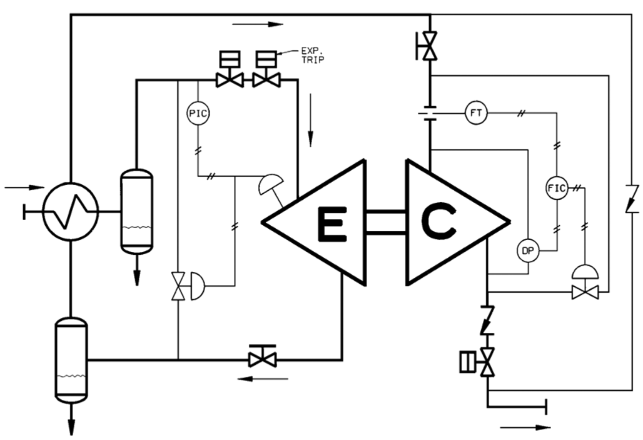

Figure 3. Typical Process Flow Diagram.

RADIAL INFLOW EXPANDER DESIGN

A turboexpander is used to refrigerate a gas stream, by removing work directly from the gas. By expanding the gas in a nearly isentropic fashion, energy in the high pressure gas entering the expander is removed, and the gas exits at a lower pressure and temperature.

A radial inflow expander is able to accomplish near-isentropic expansion by transferring energy efficiently. The gas flows radially inward, and is accelerated through inlet guide vanes and turned. The swirling, high velocity gas enters the expander impeller with relatively low incidence, because the blade tip velocity at the impeller outside diameter approximately matches the gas velocity. Work is extracted from the gas by removing this momentum: as the gas moves inward, it is forced to slow down because the blade rotational velocity decreases with the decreasing radius. The blades also turn the gas to reduce the gas velocity even further. As a result, the gas exits the impeller with low tangential velocity relative to the outside world. In this way, the angular momentum of the gas is efficiently removed. Up to 10 percent of the stage energy still remains as through-flow velocity at the impeller exit. Some of this remaining energy can be recovered with a conical diffuser. By converting the velocity into pressure, the static pressure at the turbine wheel discharge is reduced, which results in additional refrigeration.

This type of turbine is commonly a 50 percent reaction turbine, with half of the enthalpy change occurring in the rotating section (the impeller) and the other half taking place in the stationary components (the inlet guide vanes or nozzles). As an approximation, the "degree of reaction" is sometimes thought of as the pressure drop across the rotor divided by the pressure drop across the stage.

The flow of gas through an expander stage encounters several rapid transitions. Although the inlet guide vane and impeller geometry can be designed based on bulk fluid flow assumptions, a CFD analysis provides insight into important localized fluid flow patterns. A typical computational mesh is shown in Figure 4.

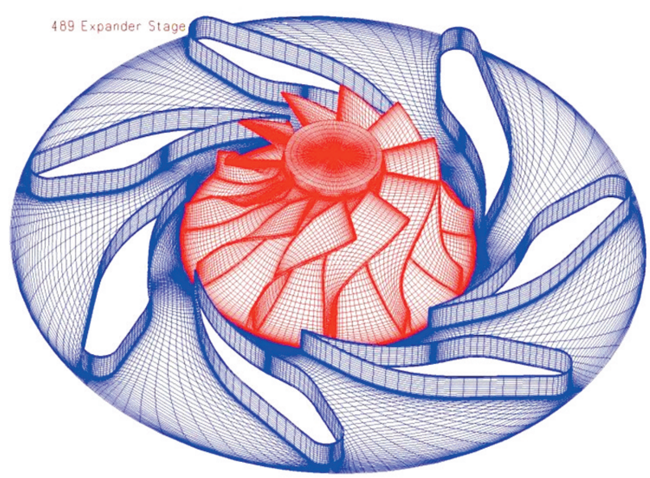

Figure 4. Radial Inflow Expander Mesh with Inlet Guide Vanes.

INLET GUIDE VANES

The pressure drop through the inlet guide vanes (IGVs) is used to accelerate the gas as it turns toward a circumferential direction. The goal is to do this with a minimal loss in total pressure, with uniform velocity and flow angle at the IGV exit, and to minimize the wake region downstream. A "nozzle efficiency" for the vanes can be defined as in Equation (1):

$$Nozzle\ Efficiency = \frac{C_2^2}{C_{2s}^2} = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \tag{1}$$

where:

h_{01} = Total inlet enthalpy
h_2 = Actual IGV static outlet enthalpy
h_{2s} = Ideal static outlet enthalpy

This process can be easily visualized on an h-s diagram, as shown in Figure 5. It should be noted that expansion through nozzles is an isenthalpic process, where the total enthalpy at the nozzle exit will be the same as the total enthalpy at the nozzle inlet (h_{01}). The static enthalpies can be used when modeling the process as isentropic. However, there will be some loss of total pressure, as a portion of the inlet pressure is not converted to velocity, but increases the static temperature (heating caused by friction). This increase in static outlet temperature makes h_2 larger than h_{2s}, and thus results in an efficiency less than 100 percent. The nozzle efficiency should generally be 95 percent or higher for well-designed vanes.

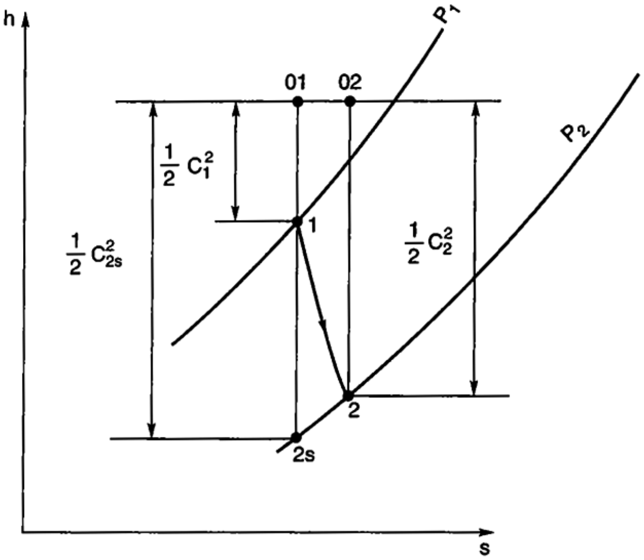

Figure 5. Enthalpy-Entropy (h-s) Diagram of Typical Nozzle.

Figure 6 and Figure 7 show the reduction in static pressure in the IGVs and the corresponding increase in velocity. The velocity and direction of the gas are relatively uniform at the exit of the IGV channel. The total pressure plot of Figure 8 indicates that the flow has accelerated with a minimal loss of total pressure. The plot also shows the wake regions, which are acceptably small. The resulting nozzle efficiency is 96 percent.

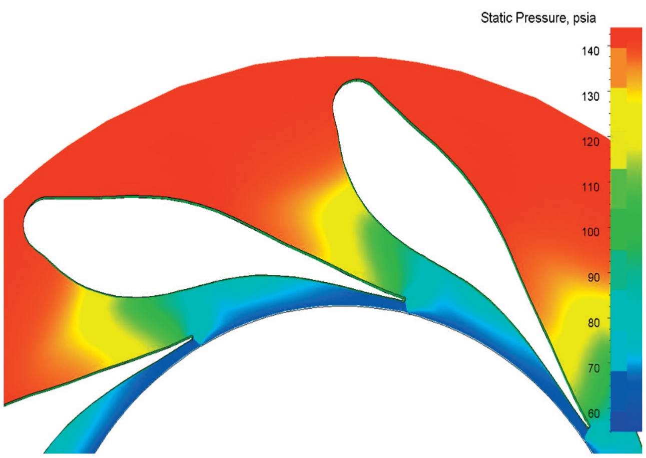

Figure 6. Static Pressure Contour in Expander IGVs.

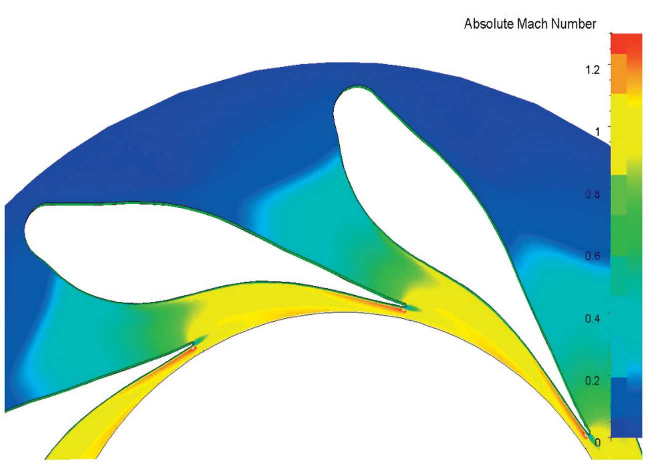

Figure 7. Absolute Mach Number Contour in Expander IGVs.

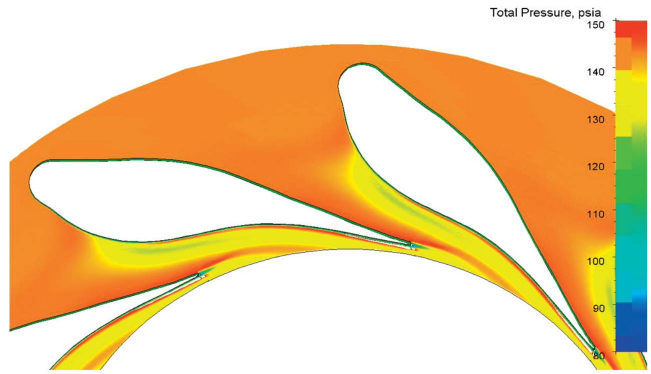

Figure 8. Total Pressure Contour in Expander IGVs.

IMPELLER

When an expander stage operates at 50 percent reaction, the velocity at the IGV exit approximately matches the blade tip velocity. For an observer "onboard" the wheel, the flow appears to enter radially, and the blades are therefore radial at the inlet to align with the relative gas angle.

For expanders, the aerodynamic performance is often expressed in terms of a velocity ratio, or U_2/C_0. U_2 is the blade velocity at the impeller outside diameter. C_0 is sometimes called the "isentropic spouting velocity," with the velocity increase being proportional to the square root of the decrease in enthalpy across the entire stage. Thus the stage enthalpy drop is termed C_0, when expressed as velocity, as defined in Equation (2).

$$C_0 = 223.8\sqrt{dhs_{stage}} \tag{2}$$

where:

C_0 = Stage isentropic enthalpy change, expressed as velocity in ft/s

dhs_{stage} = Stage isentropic enthalpy change, energy per unit mass in Btu/lbm

At 50 percent reaction, half of the stage energy is converted to velocity through the IGVs, and the nozzle exit velocity will be approximately as shown in Equation (3):

$$C_1 = 223.8\sqrt{0.5 * dhs_{stage}} \tag{3}$$

where:

C_1 = Gas velocity at nozzle exit, ft/s

dhs_{stage} = Stage enthalpy drop, Btu/lbm

The ratio C_1/C_0 will have a value of 0.707. Since the jet velocity exiting the nozzles is highly tangential and the nozzle efficiency is very high, the true tangential component of the jet stream will be very similar in magnitude to the absolute velocity of the jet stream calculated by Equation (3). The blade velocity at the wheel outside diameter is termed U_2, as shown in Equation (4), which is proportional to the product of wheel diameter and speed. For zero incidence at the leading edge of the radial blade, the tangential component of C_1 should be about equal to U_2. Therefore this is approximately where the peak efficiency will occur, which corresponds to a U_2/C_0 of about 0.7.

For mechanical reasons, it is not always possible for the impeller to run fast enough to match the peripheral velocity exiting the IGVs. In these cases, it is desired to lean the inlet of the blade to more closely align with the gas flow, although this causes the blade stress to increase, and thus may not be possible in all cases.

INTERPRETING EXPANDER PERFORMANCE CURVES

Most turboexpanders use a curve for the expander performance that is a plot of expander efficiency versus U_2/C_0, as shown in

Figure 9. To estimate the performance of an expander operating in the field, the wheel tip speed is first calculated, which simply requires knowing the expander wheel diameter and the speed it is running, as shown in Equation 4:

$$U_2 = D * N / 229 \tag{4}$$

where:

U_2 = Expander tip speed, ft/s
D = Expander wheel, inches
N = Speed of expander wheel, rpm

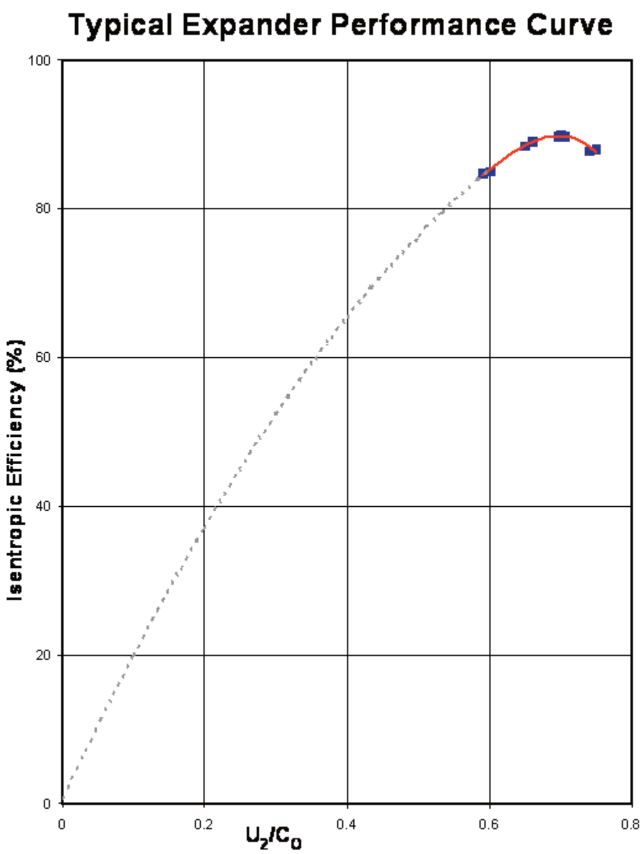

Figure 9. U_2/C_0 Versus Efficiency Curve for Typical Expander.

Once the expander tip speed is calculated, the expander inlet gas stream composition, pressure, and temperature are measured, along with the expander outlet pressure, which allows the isentropic enthalpy drop across the expander stage to be calculated. This is most easily done with a computer program, but can be obtained through laborious hand calculations if necessary, which are beyond the scope of this tutorial. This topic will be covered in more detail during the lecture part of this tutorial, but will not be discussed further in the text. Once the isentropic enthalpy drop is known, the C_0 can be calculated using Equation (2). The final step is then to divide U_2 by C_0, then use the performance curve for the expander (similar to that shown in Figure 9) to find the estimated efficiency for the field operating conditions.

It should be noted that this method does not include corrections for the discharge Q/N, but should still result in a close approximation for the predicted efficiency.

LOAD DEVICES

When gas is passed through the expander, the temperature of the gas is lowered, even for an ideal gas. This is because the gas produces work on the expander wheel, causing it to rotate. That work must be absorbed somewhere in the system, even if it is simply dissipated as heat. For some small turboexpanders, often called "hydrobrakes," the work is dissipated in shearing action of a fluid (usually the same oil that is used to lubricate the bearings). These systems are economical only when the power level is low and the cost of a compressor loaded machine is considered too high.

Similarly, the power can be dissipated in a compressor loaded machine that uses atmospheric air as the working fluid. This type of system has the advantage of using the atmosphere as a heat sink, since fresh air is drawn in on a continuous basis, and the hot discharge air is vented to atmosphere. Still, the power is completely wasted, even though the desired refrigeration from the expander is utilized.

For most process applications, the refrigeration effect is the main reason that the turboexpander is purchased. In order to get the most refrigeration from the process, the power generated by the expander can be used to drive a centrifugal compressor, which is used to increase the differential pressure across the expander, thus allowing it to provide more refrigeration than the expander alone can provide. In addition, if the amount of gas flowing through the compressor is generally in proportion to the amount of gas flowing through the expander, then the system will generally respond favorably to changes in throughput, since when the expander power that is produced is increased due to higher flowrates, so too is the work absorbed by the compressor.

Most compressor loaded turboexpanders do not use speed control systems for this reason. Even though they are "turbine driven" compressors, there is usually not a traditional governor for speed control. The speed is then free to vary up or down to achieve a power balance between the expander and the compressor.

In some instances, the plant has a need for electrical power to be produced, and the refrigeration effect of the expander is either of lesser importance or is actually considered undesirable. In these instances, the most common load device is an electrical generator (Figures 10 and 11). In most applications, the generator is not isochronous, meaning the expander does not have a speed control system capable of maintaining the line frequency. Instead, an induction generator is used, allowing the machine to operate within a narrow speed range dictated by the power produced by the expander wheel, and the system frequency is maintained by the rest of the electrical grid.

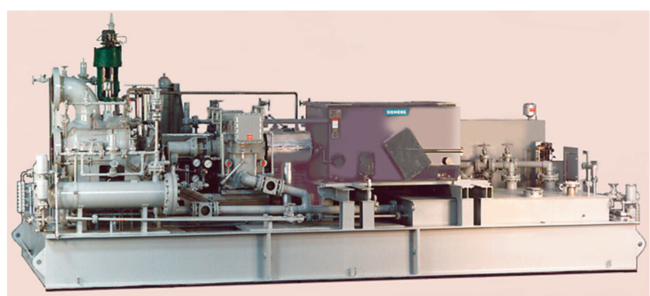

Figure 10. Expander-Generator.

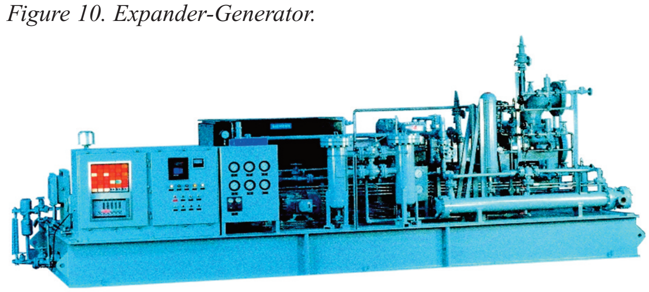

Figure 11. Expander-Generator Cross-Section.

CENTRIFUGAL COMPRESSORS

As discussed above, centrifugal compressors are the most common load device used on turboexpanders. These compressors are similar to traditional "beam" type compressors with a few notable differences.

First, the impeller material most often used is aluminum, rather than steel. Aluminum is relatively easy to machine, has a good strength to weight ratio, and is suitable for use in most of the typical gas streams that use turboexpanders.

Second, the impeller is frequently an “open” design, meaning that the wheel does not have a shroud or cover plate. This is desired for several reasons: the wheel stress is lower for a given tip speed, the machining quality and wheel integrity are higher for a one-piece open wheel, and the shroud resonance issues inherent in shrouded wheels are not present in open wheels.

Finally, they are almost always “3-D” wheels, meaning that the inducer portion of the wheel has an axial rather than radial inlet. This leads to higher efficiency and lower inlet Mach numbers.

In earlier paragraphs a radial inflow expander was described that produced power by taking a high velocity swirling gas stream and slowing it down while removing the swirl to produce work. It is then logical to think of the centrifugal compressor as taking in a low velocity gas stream with no swirl and generating a high velocity gas stream with a great deal of swirl, thus absorbing work.

This high velocity swirling gas stream produces a pressure rise that can be thought of as two distinct mechanisms. First, the absolute flow velocity at the exit of the wheel is highly tangential, creating a pressure rise similar to the one created by moving a glass of water in an arc over your head such that the water remains in the glass despite the fact that the glass is briefly upside down. The second means of pressure generation is when the high velocity gas passes through the diffuser, causing it to slow down and additional pressure rise to take place.

Most turboexpanders use a vaneless diffuser to recover the velocity energy leaving the wheel. This tends to give a wider stable range and less chance of exciting a wheel resonance than vane type diffusers.

INTERPRETING COMPRESSOR PERFORMANCE CURVES

The compressor performance curves used for most turboexpanders are very similar to the curves used for other centrifugal compressors. Generally the plot has the actual inlet flow plotted along the X axis and the head rise plotted along the Y axis (refer to Figure 12). In some cases, the flow is divided by the speed (referred to as “Q over N,” and written Q/N), and the head is divided by speed squared. This results in a plot that is basically independent of speed for “normal” Mach numbers, though the data will still produce multiple curves as a function of speed if the Mach number effects are high.

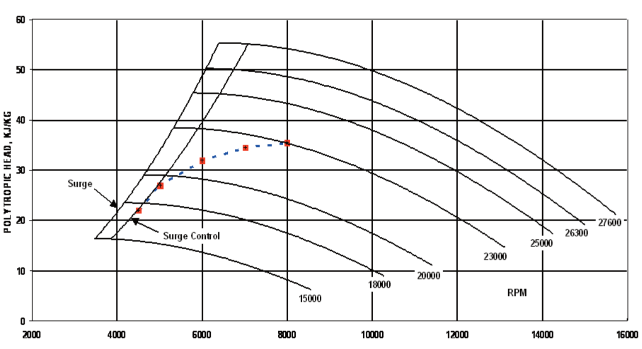

Figure 12. Typical Compressor Head Curve.

The use of head rather than discharge pressure often confuses plant personnel. However, if the curve provided by the manufacturer was discharge pressure against flow, then any time the process conditions changed at all, the curve would be inaccurate. By plotting head versus flow, the curve can be used for virtually all normal variations of process conditions.

There are three things needed to compare the predicted head and efficiency from the curve (usually based on test data) to the actual field operating conditions: machine speed, actual inlet volume flow, and head calculated from measured field data.

Almost everyone can obtain the speed without a problem. Most people can obtain the actual inlet volume going to the compressor, although many people forget to correct the data taken for variations in pressure, temperature, and gas composition, since most plant flowmeters have constant values for these parameters.

The problem usually comes in when head is to be calculated. As previously discussed for expanders, it is best to use a computer simulation program to obtain the head from the field data. This topic will be explored in greater detail in the lecture part of this tutorial, but it will not be covered further in the text.

POLYTROPIC VERSUS ISENTROPIC

Both efficiency and head can be expressed as isentropic or polytropic (or isothermal, but this is rarely used with turboexpanders). By convention, isentropic head and efficiency are normally used for expanders and polytropic head and efficiency are normally used for compressors.

The difference between the two relates to what the “ideal” enthalpy change would be for a perfect machine of 100 percent efficiency. Isentropic head (enthalpy) is just that: at a constant value of entropy, the enthalpy difference is calculated as simply the difference between the inlet and outlet enthalpy. Polytropic head, on the other hand, seeks to break up the ideal compression or expansion process into an infinite number of small isentropic changes.

At low pressure ratios, this ideal enthalpy change is virtually the same for both isentropic and polytropic cases. However, as pressure ratio rises, the curved yet parallel nature of the constant pressure lines as viewed on an h-s diagram show that the ideal polytropic head will be greater. Similarly, lower efficiencies will yield a larger difference between isentropic and polytropic ideal head.

What does all this mean? Basically that the difference between isentropic and polytropic efficiency will be smallest for small pressure ratios and high efficiencies. For expanders, isentropic efficiency will always be higher than polytropic efficiency, and for compressors, polytropic efficiency will always be higher than isentropic efficiency. In short, either definition is acceptable as long as comparisons are made only between like measurements.

TOTAL VERSUS STATIC CONDITIONS

If the difference between isentropic and polytropic is not confusing enough, another important parameter to understand when comparing efficiency values is the difference between total and static conditions. The efficiency number is normally given as a percent figure, as in “the efficiency is 85 percent.” This normally, but not always, means that the efficiency is 85 percent “total to static, flange to flange.” This qualifier means that when the efficiency was calculated, the inlet conditions were evaluated on a “total” basis, which includes the velocity component, as measured at the inlet flange. The outlet conditions were evaluated on a “static” basis, in which the velocity component was ignored, and it was measured at the outlet flange.

Like the isentropic versus polytropic issue, the main thing to understand is that one should always be sure to compare efficiencies relative to the same baseline. Unlike the isentropic versus polytropic issue, certain types of turbomachinery really need to use specific efficiency definitions. For example, if a compressor designer were to use a “total to total flange to flange” efficiency for his or her design basis, the design would erroneously be guided toward removing all efforts attempting to recover the velocity energy leaving the impeller, since the diffuser can only hurt the total to total efficiency. This would lead to very small outlet flanges with very high flange velocities, clearly a poor design, since too much of the energy put into the gas by the impeller will leave the stage as velocity, rather than static pressure.

By using the proper efficiency definition of “total to static, flange to flange,” efficiency improvement efforts will direct the development toward efficient diffuser designs that will hurt the total to total efficiency but will raise the outlet static pressure and thus improve the total to static efficiency.

When would "total to total" efficiency be correct? One obvious example is the turbine and exhaust nozzle of a jet engine. The desired output of this device values velocity, not static pressure, so using a "total to static" definition here would lead to the installation of a diffuser rather than a nozzle following the last stage of the turbine, which would be completely unacceptable for its intended purpose.

MAGNETIC BEARINGS IN TURBOEXPANDERS

As previously mentioned, magnetic bearings are increasingly being used in turboexpanders. At one point or another, almost everyone has taken two permanent magnets, held them with like poles facing each other, and felt the repulsive force produced by the magnets. When this principle is used as a bearing, it is referred to as a passive magnetic bearing, since there are no control devices used. For many reasons, industrial magnetic bearings are virtually never based on this principle. In some cases, permanent magnets are used to supply a bias flux in industrial machines, but never as the sole support system.

In turboexpanders, the term magnetic bearings refers to active magnetic bearings, in which the shaft is held in position using electromagnets arranged in close proximity to the shaft. The current supplied to these electromagnets is modulated by a control system. Thus it can be seen that the forces in an active magnetic bearing are never repulsive, they are always attractive.

For additional information on magnetic bearings, Schmied (1991) provides excellent information on magnetic bearing control systems in a very understandable fashion, and Jumonville, et al. (1991), provide useful photographs of actual magnetic bearing hardware, loading plots, and general design guidelines.

Figure 13 provides a reasonable pictorial of how magnetic bearings are incorporated into a typical turboexpander bearing housing. Figure 14 shows how the control system is used to locate the shaft within the bearings. When the shaft moves away from the desired position (usually in the center of the bearing), the position sensors detect this change and an error signal is generated and sent to the proportional, integral, and derivative (PID) controller. This controller is similar to the PID control loops found throughout virtually all industrial plants. Two main differences should be noted. First, the rate at which the error signal is received, processed, and the output action sent to the bearings is much faster than in standard process control. Second, the typical process PID controller usually does not need or use the derivative portion of the controller. In magnetic bearing PID control systems, the derivative action is absolutely essential. While most industrial process control applications are "open loop stable" implying that they can be placed in "manual" and the loop does not go unbounded, magnetic bearings are "open loop unstable" and cannot be placed in a "manual" mode.

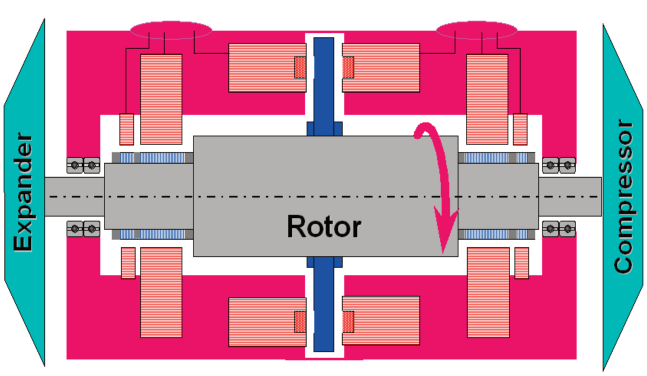

Figure 13. Turboexpander with Magnetic Bearings.

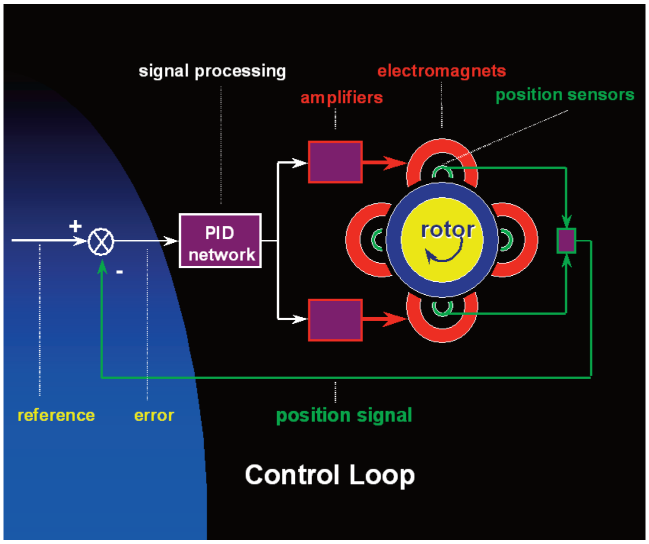

Figure 14. Magnetic Bearing Control System.

A good analogy for this is a small ball inside of a bowl or dish. Gravity will attempt to move the ball to the center of the bowl. If you blow on the ball, it will move away from the center of the bowl, but the harder you blow, the more the gravity force pushes back, attempting to restore the ball to the center of the bowl.

Now imagine that the bowl is turned upside down, and the ball is placed on top of the "hill" formed by the outside of the bowl. It will be very difficult to balance the ball in the center of the bowl now, because gravity forces want to move the ball toward the edge of the dish rather than the center. Even if you are successful in centering the ball, if you blow on it even slightly, the ball will immediately fall off the bowl. This is what control of a magnetic bearing is like. Each competing magnet is pulling the shaft toward it, and the currents must constantly be varied to maintain the rotor position in the center of the bearing.

The control system itself can be either analog or digital. In an analog system, the elements of the PID loop are configured from actual electrical hardware components (resistors, capacitors, inductors, op amps, etc.). This has the advantage of using relatively available components having fast action with minimum time delays. However, since the initial tuning of the system can require several control loop changes, the time it takes to physically change these components can be substantial. Digital control makes it much easier to change the control tuning loop parameters. The downside is that a time delay is inherent in these systems, which contributes an undesirable phase lag that is worse at higher frequencies. It also requires the use of advanced semiconductors to achieve the fast processing times needed. These advanced components become obsolete relatively quickly compared to the lifespan of most rotating equipment, making long term availability of the repair parts a function of the inventory secured by the magnetic bearing manufacturer. This problem is not unique to magnetic bearing controllers. Virtually *all* new plants worldwide are using similar technology for their plant process control, and will increasingly find that some electronic parts must be replaced rather than being repaired at some point in the future. Years ago the author read that NASA was buying old computer chips on eBay for use on the Space Shuttle, because they could no longer buy these old chips from the original equipment manufacturer (OEM) and their inventory had finally run out after many years of use!

While on the subject of magnetic bearings, it is worth noting that the vibration alarm and trip levels are a bit different when using magnetic bearings compared to oil bearings. In fact, even the idea of "vibration" needs to be reconsidered with magnetic

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bearings. The first thing to understand is that the shaft should normally operate in the center of the auxiliary bearings, to allow the maximum motion to take place without contact between the shaft and the aux bearing. Thus both steady state motion and vibratory motion or a combination of the two are cause for a trip in a magnetic bearing. For example, if the radial clearance from the shaft to the aux bearing is 0.006 inch (150 microns) when the shaft is properly centered, and the trip level represents a value one half of this amount (0.003 inch or 75 microns), then a steady state radial sideload that causes the shaft to move off-center by 0.002 inch or 50 microns radially will leave room for only 0.001 inch or 25 microns 0-p vibration to reach the trip level.

It is also worth noting that the total allowable motion for the shaft relative to the stator is larger for a magnetic bearing. This is usually acceptable for two reasons. First, there is no Babbitt to fatigue in a magnetic bearing, so somewhat larger vibratory motions do not cause damage to the bearings the way they can in an oil lubricated bearing. Second, the magnetic bearing has provisions to almost completely eliminate the need for the amplifiers to react to synchronous vibrations such as unbalance. Thus the transmitted forces due to large vibrations of the shaft tend to be much lower for magnetic bearings compared to oil lubricated bearings.

It is interesting to note that the author has proposed, since the early 1980s, that oil lubricated bearings use similar logic to set the allowable vibration levels in oil bearing machines, but to no avail. It is obvious that lightly loaded oil bearings in which the shaft runs near the center of the shaft can tolerate much higher vibration levels without doing damage to the Babbitt or imparting large forces to the stator, since the film thickness is very large and the film stiffness is usually relatively "soft." On the contrary, heavily loaded oil bearings operate with a very small film thickness coupled with high film stiffness and thus cannot tolerate large vibration levels without increasing the chances for Babbitt fatigue and elevated stator vibration.

A commonly used rule of thumb for the maximum allowable vibration in the field is 50 percent of the bearing clearance. For a lightly loaded bearing, this might be conservative, whereas for a heavily loaded bearing, the Babbitt could be subject to severe fatigue with this level of vibration.

In fact, due to the requirement that both lightly loaded and heavily loaded oil bearings must meet the same low vibration level called out in API 617 (2002), it is not uncommon for designers of lightly loaded bearings to intentionally raise the load on the bearing by removing some of the bearing surface area, thus arriving at a smaller minimum film thickness and greater chance for Babbitt fatigue, but less chance of "high" vibration! The net effect can sometimes be an arguably "worse" bearing that may "appear" to run better (lower vibration), but may in fact be more likely to fail. Clearly, this issue is not of major concern, since high reliability is routinely achieved for heavily loaded bearings. However, when evaluating high levels of vibration in the field due to unbalance, it is still the author's opinion that the eccentricity ratio, vibration level, and transmitted forces be used to determine how high is "too high" in these cases!

AUTOMATIC THRUST EQUALIZER

Most turboexpanders, whether they use oil or magnetic bearings, employ an automatic thrust equalizer, or ATE. This device uses signals from the bearings to sense the position of the shaft that causes a valve to move, changing the pressure behind one of the two impellers to pneumatically balance the external load that is applied. With this system, thrust loads are maintained at low levels during normal operation, allowing nearly the full load capability of the bearing to be available for unexpected transient loads. Figure 15 shows a schematic of how such a system works for an oil bearing system.

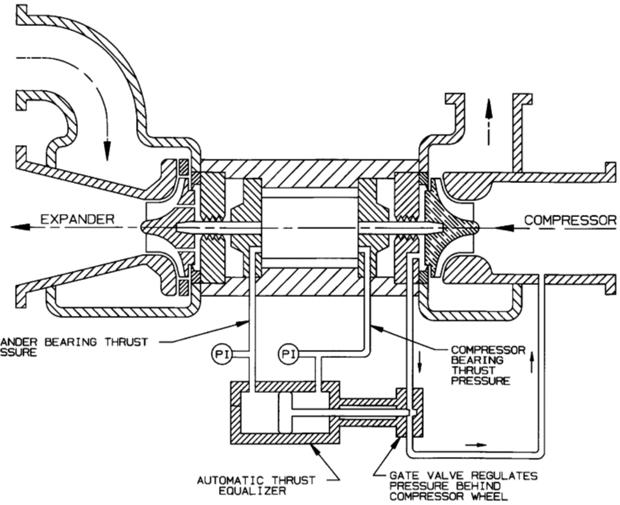

Figure 15. Automatic Thrust Equalizer Installation.

SEAL GAS SYSTEM

Despite the hermetically sealed nature of the typical turboexpander, seal gas systems are still normally required. For example, it is undesirable for the gas that is passing through the expander wheel to enter into the bearing housing. The gas in the expander is typically very cold and has not been filtered for small particles. If this gas were directly admitted into the bearing housing, the oil would become dirty, very cold and possibly freeze solid. Likewise, it is also undesirable for the oil from the bearing housing to enter the expander housing, since it would tend to freeze inside the machine and in downstream heat exchangers causing problems. It should be noted that it is very difficult if not impossible to get all of the oil out of a typical cryogenic cold box (heat exchanger) once it has gotten in.

With magnetic bearings, there are many parts that should be protected from the cold gas of the expander, so seals are also used in these applications.

For both oil and magnetic bearing systems, the most common type of seal used to separate the process from the bearing housing is the buffered labyrinth seal, Figure 16. Clean and filtered buffer gas is injected near the middle of the seal, and is forced to flow toward both the bearing housing and the expander housing (for the expander end seal). This prevents fluids in one housing from entering the other housing.

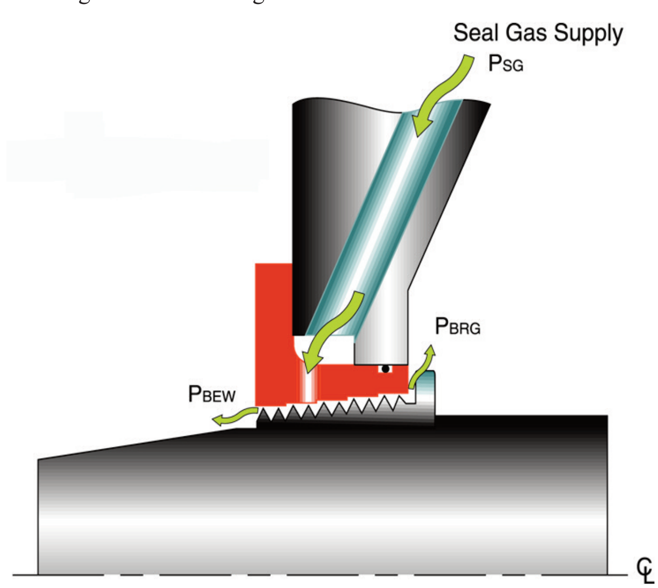

Figure 16. Labyrinth Seal.

The use of a stepped tapered labyrinth further enhances the ability of the seal to reject oil from the bearing housing, as well as lowers the buffer gas flow required by the seal. By using a stepped tapered laby instead of simply a tapered laby, the seal clearance can be maintained regardless of the seal's axial location. The flow through a normal tapered laby seal will be dependent on the axial location of the rotating and stationary parts, since the radial clearance changes with axial motion of the parts.

In order to maintain an effective barrier even if the seal wears, the buffer gas (or "seal gas") should be differential pressure controlled, not flow controlled. If the buffer gas is flow controlled and the seal wears, then the ability of the seal to provide an effective barrier is compromised. With a differential pressure controller, the worn seal can still be an effective barrier, since additional flow will be provided when needed to maintain the differential pressure.

The source of the seal gas (buffer gas) can also be important. This source needs to be sufficient pressure to provide adequate buffering of the seals under all expected operating conditions. It should be warm, clean, and dry by the time it enters the bearing housing.

The hermetic nature of the lube oil system, like the refrigerator in your home, means that the oil viscosity will be changed due to the presence of the process gas. Also like your refrigerator at home, it means the actual oil viscosity inside the system will be lower than the viscosity of pure oil at the same pressure and temperature conditions. This is referred to as "dilution," since the seal gas is "dissolved" in the lube oil. To deal with dilution, it is often necessary to increase the oil viscosity so that the resulting diluted mixture is sufficiently viscous to provide the necessary stiffness and damping that the bearings need to control the shaft motion.

Obviously, with magnetic bearings, there are no concerns about oil dilution. However, in some applications, additional cooling gas must be put into the magnetic bearing housing to remove heat due to windage, resistance, and eddy current losses in the magnetic bearings. When needed, this cooling gas is normally supplied from the same source as the seal gas supply.

The last thing to mention here is that seal gas should be turned on before the lube oil is turned on when starting a machine with oil bearings, to ensure that oil is not accidentally admitted into the expander and compressor housings. Likewise, when shutting down the machine, the lube oil should be turned off prior to shutting down the seal gas system.

CONCLUSION

Turboexpanders have come a long way in the last 20 years or so. Their high efficiency and reliability have led to widespread use around the world. In that period of time, the use of magnetic bearings in some critical turboexpander applications such as ethylene plants has gone from novelty to standard practice. In the natural gas processing market, magnetic bearing equipped turboexpanders are making inroads as well. Computational Fluid Dynamics (CFD) programs are providing windows into aerodynamic improvements and new applications that were difficult to tackle in the past.

As evidence of this trend, API 617 (2002), Seventh Edition, has added turboexpanders as a separate subject, occupying Chapter 4 of this specification. Annex 4F of the same document provides guidelines for purchasing equipment using magnetic bearings, and this section is particularly applicable to turboexpanders.

In short, the past 20 years have been filled with exciting improvements and new applications for turboexpanders, and it looks like the next 20 years will continue this trend!

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