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**Modern Methods to Detect Degradation in Centrifugal Compressor Performance and Use of Thermodynamic Performance Modelling to Identify Causes**

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## ABSTRACT

Detecting degradation in centrifugal compressor thermodynamic performance can be challenging. Machines can operate at different inlet and discharge conditions, different inlet guide vane positions, different speeds and even on different gases or gas mixtures. Compressor performance curves that typically show mass flow versus discharge pressure and power can limit one's ability to analyze performance at different operating conditions. The use of more general and invariant compressor performance parameters, such as inlet volumetric flow versus head and efficiency can be more useful, especially when evaluating the cause(s) of a loss in compressor performance. Identifying the cause(s) is useful so that the problem can be corrected without the need of a more comprehensive repair or a trial and error approach. The purpose of this tutorial is to show how compressor performance degradation can be detected and how to identify the possible cause(s).

## INTRODUCTION

The first part of this tutorial will describe a typical compressor system. For the sake of simplicity, a typical air compressor system will be described, and this example will be used in several sections throughout the tutorial to explain various concepts.

The second part of this tutorial will cover the basic compressor performance characteristics and thermodynamic processes. These models will be based on the ideal gas equations and will include isentropic, polytropic and isothermal processes. This provides a good foundation for those interested in understanding compressor performance relevant to many compressors used in air separation plants. ASME PTC-10, Performance Test Code on Compressors and Exhausters, will be used as a reference for the thermodynamic equations. The different types of tests in the code will be discussed as they relate to field testing.

The third part of the tutorial will discuss instrumentation used for condition monitoring and performance testing

The fourth part of this tutorial will cover modern equipment condition monitoring (ECM) models that can be used to detect overall compressor performance degradation at an early stage. A case study of an air compressor that suffered a loss in performance will be used to illustrate this. Not only is this important because of the loss in capacity and reduction in efficiency, but degradation can drive a compressor into an unstable operating condition that can lead to a mechanical failure.

The fifth part of the tutorial will cover three case studies where compressors performance degradation was detected. These case studies will be used to illustrate how field performance testing was performed to identify the cause(s) of the loss in performance. As described above, identifying the cause(s) is useful so that the problem can be corrected without the need of a more comprehensive repair or a trial and error approach. Some common causes of a loss in compressor performance include:

- Impeller and/or diffuser fouling
- Impeller and/or diffuser erosion
- Excessive inlet shroud erosion (open impellers)
- Excessive impeller eye seal erosion (closed impellers)
- High intercooler pressure drop and/or high intercooler approach temperature

These case studies will be based on ideal gas equations.

While many compressor applications in air separation plants can be modelled using ideal gas equations, many compressor applications in the oil and gas industry cannot. The inclusion of real gas equations of state is critical in evaluating compressor performance in these applications. The last and sixth part of the tutorial will cover real gas relationships. This will follow the current methods described in ASME PTC-10 (1997). A simple case study will be used to illustrate the differences in performance results for a high pressure $CO_2$ compressor based on ideal gas equations and real gas equations.

## BASIC COMPRESSOR SYSTEM

A typical P&ID of a compressor system is shown in Figure 1. This is a three stage compressor driven by a fixed speed electrical motor. The primary components of this compressor system include:

- Three compressor process stages with intercoolers downstream of stages one and two and an aftercooler downstream of stage three. In this example, one process stage has one impeller. This is not always the case.
- An inlet filter to remove particulates before the gas enters the compressor with a downstream differential pressure indicator (DPI) used to measure the pressure drop across the filter
- Condensate traps used to remove condensate from the coolers
- An inlet guide vane (IGV) used to control the compressor flow
- Temperature indicators in the process piping downstream of the intercoolers to measure cooler approach to cooling water temperature
- A discharge pressure indicating controller (PIC)
- A discharge mass flow indicating controller (FIC)
- A discharge blow-off valve that is used for start-up, shutdown and surge protection
- A motor power meter
- A discharge check valve used to prevent back flow into the compressor on a shutdown

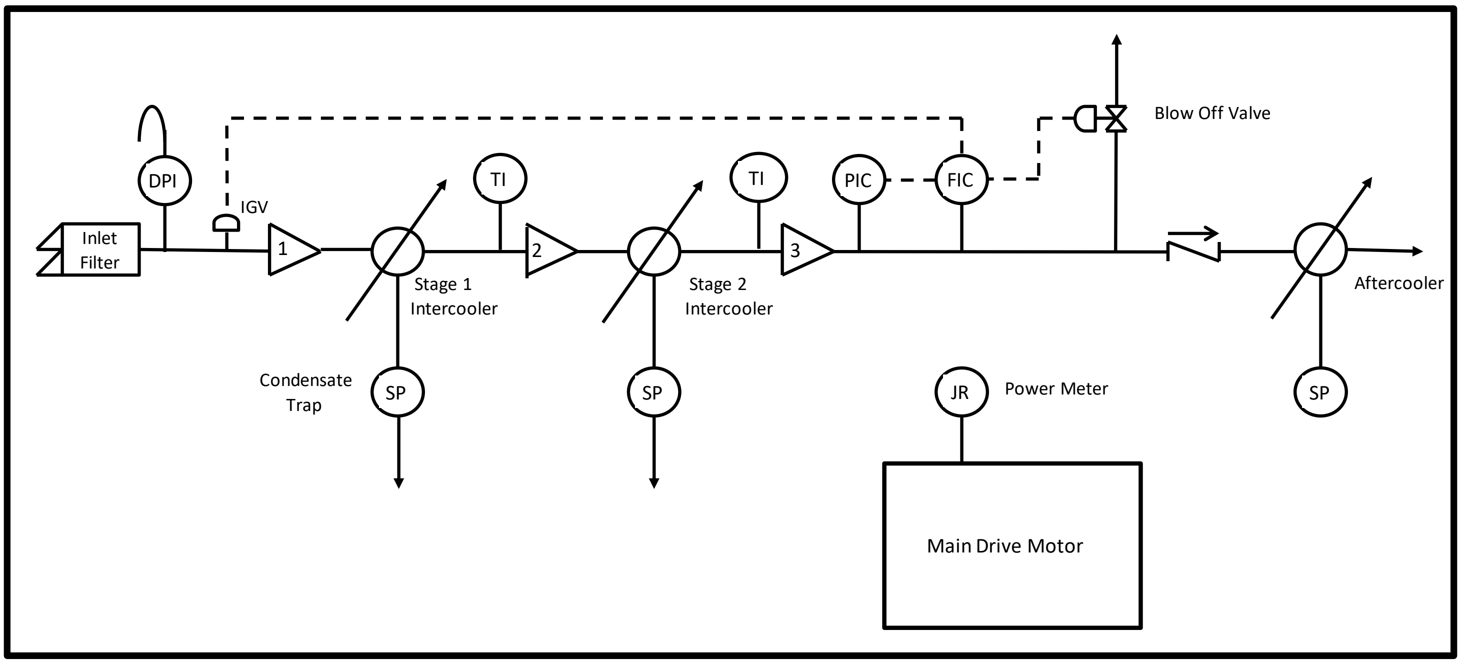


Figure 1: P&ID

These instruments are the basic components used to operate and monitor the thermodynamic performance of the compressor. There are other components, such as vibration probes and bearing temperatures, that are not shown and are used to monitor the mechanical performance of the compressor. These can also be useful when diagnosing a compressor performance problem, but will not be addressed in this tutorial.

## PERFORMANCE CHARACTERISTICS AND BASIC THERMODYNAMIC PROCESSES

Compressors operate based on a volumetric flow rate. This is the mass flow divided by the local total density. The term "total" will be used throughout the tutorial. Total pressure, for example, is the sum of the static pressure and the dynamic pressure. In a nonmoving fluid the dynamic pressure is zero and the total pressure is the static pressure. In a moving fluid the velocity creates a dynamic pressure, which is the pressure resulting from motion. If there is fluid moving in a pipe and the pressure is measured at the wall of a pipe, the measured pressure is the static pressure. To determine the total pressure, the gas velocity has to be determined to calculate the dynamic pressure, which is then added to the static pressure. Total temperature is the same. Static temperature is the is the temperature measured with a sensor traveling at the same velocity as the fluid. So, when a static temperature probe is used to measure the temperature in a moving stream, the measured temperature is the static temperature. To determine the total temperature, the dynamic component has to be added to the static component. In this tutorial, the total pressures and temperatures are calculated based on the equations in ASME

PTC-10. Most heat and material balances (HMB) from process engineers are based on static conditions, consistent with what is measured in the field with normal pressure and temperature indicators and transmitters.

Density is a function of pressure, temperature and molecular weight. So, in an air compressor, if the ambient temperature changes, the density changes and this affects mass flow.

In an air compressor, relative humidity affects the molecular weight of the gas. And, when air is compressed and then cooled, condensate can form. This condensate needs to be removed and the gas entering the next stage will have a different molecular weight. Various process gas compressors also will have a condensable component. In compressor thermodynamic calculations, the gas specific heat and heat ratio are used and have to be calculated at the inlet and discharge of each stage. Again, ASME PTC-10 can be used as a reference along with steam tables and/or an online property calculator.

IDEAL GASES

As described in ASME PTC-10, thermodynamic properties can be calculated based on the ideal gas equations of state depending on the gas composition and pressure. This will be described in more detail later in the tutorial. The gas in a typical three stage air compressor can be treated as an ideal gas, meaning that it conforms to the equation:

$$144Pv = RT \tag{1}$$

In this equation, “P” is absolute pressure, “v” is specific volume per unit mass, “R” is the gas constant, and “T” is absolute temperature. The factor 144 is included as a constant for US customary units (psia as pressure).

If “v” is replaced with “Q” (volume per time) divided by “w” (mass per unit time), the volume flow can be determined by rearranging the equation as:

$$Q = \frac{wRT}{144*P} \tag{2}$$

Recall that volume flow is the mass flow divided by the density (w/ρ) and the density based on the ideal gas law is:

$$\rho = \frac{144*P}{RT} \tag{3}$$

To determine the volume flow at the inlet of a compressor, the total inlet pressure and total inlet temperature are used in this equation. Compressor flow can also be characterized based on the flow coefficient. The flow coefficient is a dimensionless parameter defined as the compressed inlet volume flow rate divided by the product of rotational speed, and the cube of the blade tip diameter. Compressed mass flow rate is the net mass flow rate through the rotor. In equation form, this is:

$$\varphi_t = \frac{w_{rotor}}{\rho_i 2\pi N\left(\frac{D}{12}\right)^3} \tag{4}$$

As described in ASME PTC-10, “…$w_{rotor}$ is the mass flow rate which enters the rotor and is compressed. It differs from the measured mass flow rate by the amount of leakage and sidestream flow which occurs between the rotor entry and the flow measurement station.” Compressors also produce pressure, but this is a function of the head that the compressor generates. The head that a compressor generates can be calculated based on the first law of thermodynamics. Figure 2 shows a single compressor stage.

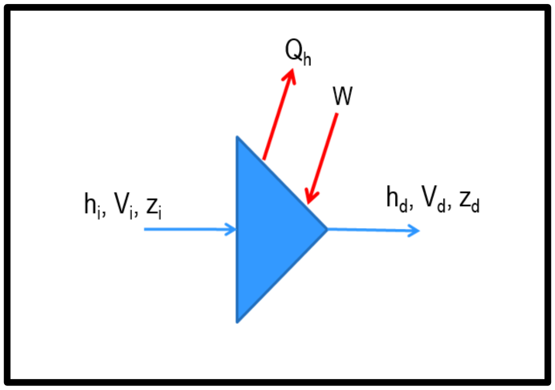

Figure 2: Single Compressor Stage

The general equation of state for a steady flow process based on Bernoulli's equation can be expressed as:

$$h_i + V_i^2/2g + z_i + Q_h = h_d + V_d^2/2g + z_d + W \tag{5}$$

Subscripts "i" and "d" refer to the inlet and discharge conditions, respectively. The inlet and discharge flanges may be considered to be at the same elevation so that $z_i$ and $z_d$ cancel out. The velocity term can be considered to be part of the enthalpy if the enthalpy is defined as the stagnation or total enthalpy, in which case the equation simplifies to:

$$h_d - h_i = -W + Q_h \tag{6}$$

So, the head produced by the compressor is the difference in enthalpy, which is the difference between the work done by the system and the heat loss. Compressor performance can be modelled after several thermodynamic processes, which include isentropic, polytropic and isothermal. Brown (2005) is a reference that illustrates how the different thermodynamic equations are derived. This will not be covered in this tutorial. The resulting equations for each thermodynamic process are shown below.

An isentropic process is an idealized thermodynamic process that is both adiabatic and reversible and obeys the equation:

$$pV^k = C \tag{7}$$

For an ideal gas with constant specific heats, the isentropic work (or head), isentropic work coefficient, isentropic efficiency and power equations are:

$$H_s = h_d - h_i = R*T*k/(k-1)*((p_d /p_i)^{((k-1)/k)} - 1) \tag{8}$$

$$\mu_s = H_s / (\Sigma U_2^2 / g) \tag{9}$$

$$\eta_s = Tin* (r_p^{((k-1)/k) - 1})/(Tout - Tin) \tag{10}$$

$$\text{Power} = \text{Mass Flow} * H_s / \mu_s \tag{11}$$

A polytropic process is a reversible process that obeys the equation:

$$pV^n = C \tag{12}$$

"n" is the polytropic component and C is a constant. For an ideal gas with constant specific heats, the polytropic work (or head), polytropic work coefficient, polytropic efficiency and power equations are:

$$H_p = R/MW *T*n/(n-1)*(r_p^{((n-1)/n)}-1) \tag{13}$$

$$\mu_p = H_p / (\Sigma U_2^2 / g) \tag{14}$$

$$\eta_p = (k-1)/k * \ln(r_p) / \ln(T_d / T_i) \tag{15}$$

Where:

$$(n-1)/n = (k-1)/(k * \eta_p) \tag{16}$$

$$\text{Power} = \text{Mass Flow} * H_p / \eta_p \tag{17}$$

For given suction and discharge temperatures and pressures, the isentropic head and efficiency will be different than the polytropic head and efficiency. The isentropic efficiency considers only the start and end states of a compression process, whereas the polytropic process is a function of the path. There are an infinite number of polytropic paths depending on the value of "n" in equation (12). The exponent "n" may have any value from zero to infinity depending on the particular process. If "n" is equal to "k", the process is isentropic, which is a special case and represents an ideal, frictionless adiabatic process. As mentioned above, this only considers the start and end states. When modelling a compression process as polytropic, heat transfer and friction are factored in using equation (16) assuming an ideal gas with constant specific heats. While the isentropic process is simpler to use when making performance calculations, the polytropic process is more accurate across varying compression ratios. Figure 3 shows the how isentropic and polytropic efficiency compare as a function of pressure ratio for a typical compression process.

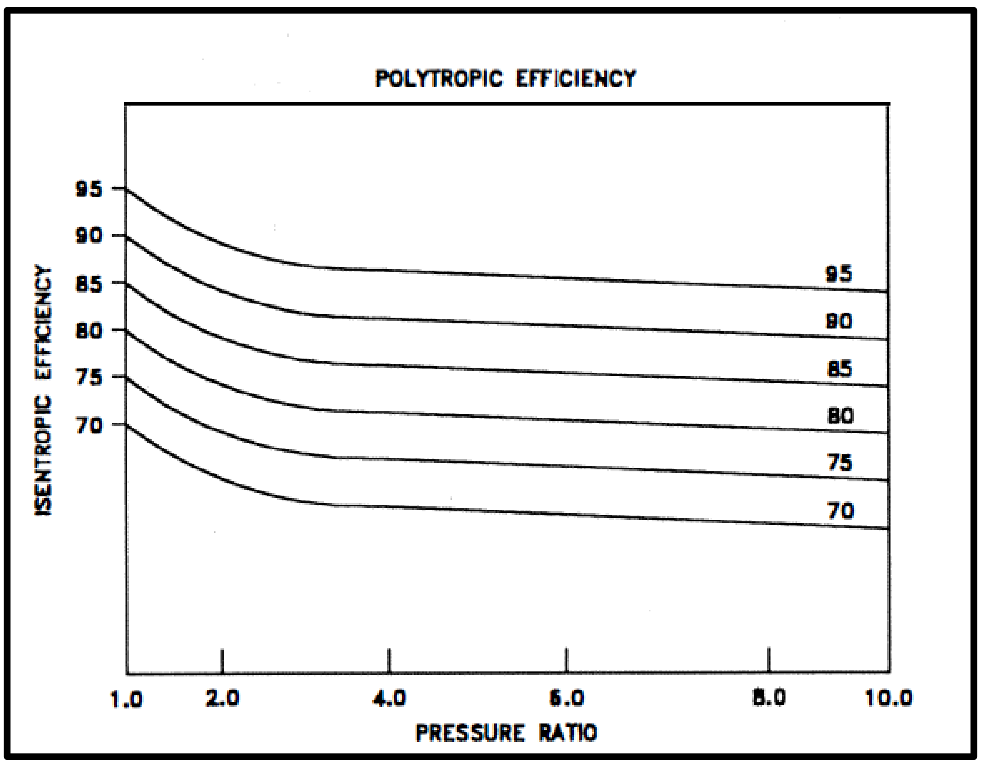


Figure 3: Relationship between Isentropic and Polytropic Process (ref. Boyce 1993)

Isentropic and polytropic processes can be used to model compressor performance stage by stage. The head is used to determine the pressure ratio, which determines the inlet pressure to the subsequent stage (minus the intercooler pressure drop). The inlet temperature of the subsequent stage is the temperature downstream of the intercooler. The powers for all the stages are added together and this is the total gas power. In addition to this power, there are parasitic losses in a compressor. These are comprised of mechanical losses and other power requirements which do not contribute to the energy rise of the gas. Mechanical losses may be estimated based on the heat rejection to the lubricating oil. The equation is:

$$\text{Mechanical Losses} = \text{Oil Flow} * c_p * \Delta T \tag{18}$$

Other parasitic losses can include power to drive auxiliary equipment. The total compressor power is the total gas power plus all the parasitic losses.

An isothermal compression process is one in which the gas is compressed, and the temperature remains constant. So, this would be the equivalent of having an infinite number of intercoolers and this represents an ideal case. The isothermal head and efficiency equations are:

$$H_{iso} = R/MW *T* \ln(r_p) \quad (19)$$

$$\eta_{iso} = \text{Mass Flow} * \text{Hiso} / \text{Total Compressor Power} \quad (20)$$

An isothermal process can be used to evaluate the total compressor efficiency and as will be described in the next section, it can be used in equipment condition monitoring to detect degradation in thermodynamic performance of a compressor.

All methods shown here can be used to evaluate the health of the compressor. The accuracy of the analysis will be different based on type of compressor and assumptions made in the equations.

As stated in ASME PTC-10, "This Code contains provisions for two different types of tests. A Type 1 test must be conducted on the specified gas with a limited deviation between test and specified operating conditions. A Type 2 test permits the use of a substitute test gas and extends the permissible deviations between test and specified operating conditions." Also as stated in ASME PTC-10, "Type 1 tests are conducted with the specified gas at or very near the specified operating conditions." For most field tests, the test will be performed close to the specified operating conditions and so a type 1 test procedure can be used. The calculations discussed up to this point assume the gas can be treated as an ideal gas. However, many gases in the oil can gas industry do not conform to ideal gas laws and calculation procedures based on real gases need to be used. Whether a gas can be treated as an ideal gas or not depends largely on the compressibility of the gas at the inlet and discharge conditions, although the compressibility within the compressor also has a factor. The compressibility factor "Z", also known as the compression factor or the gas deviation factor, is a correction factor which describes the deviation of a real gas from ideal gas behavior. Take for example dry air at 100°F. The graph in Figure 4 shows that between 10 psia and 1200 psia the compressibility factor is close to 1 and so air can be treated as an ideal gas for most main air compressor and booster air compressor applications.

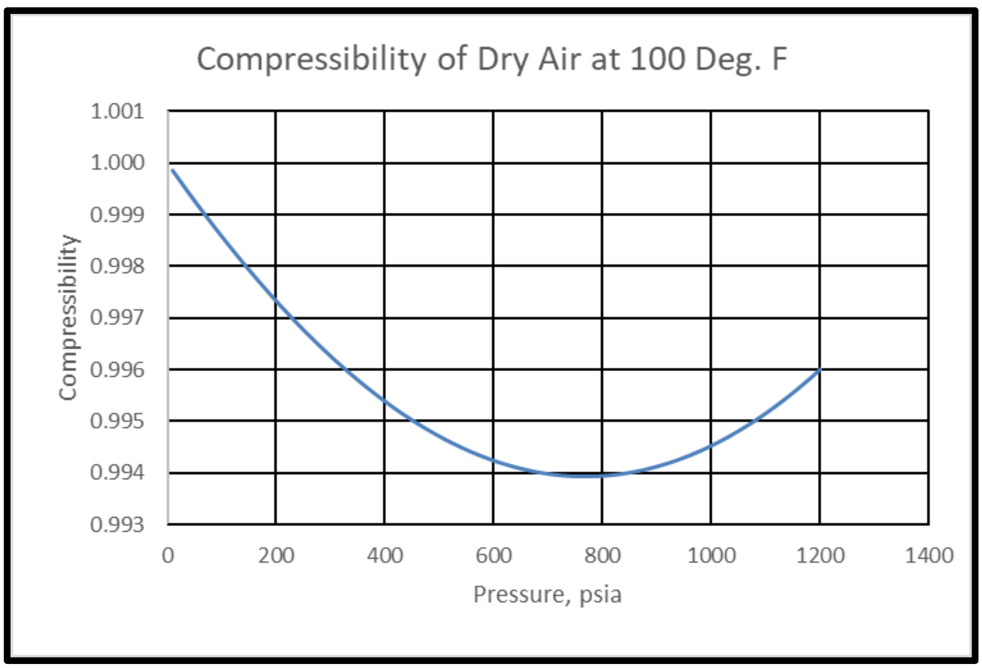


Figure 4: Compressibility of Dry Air at 100°F as a Function of Pressure

But, for $CO_2$ at the same conditions, the compressibility changes significantly with pressure as shown in Figure 5.

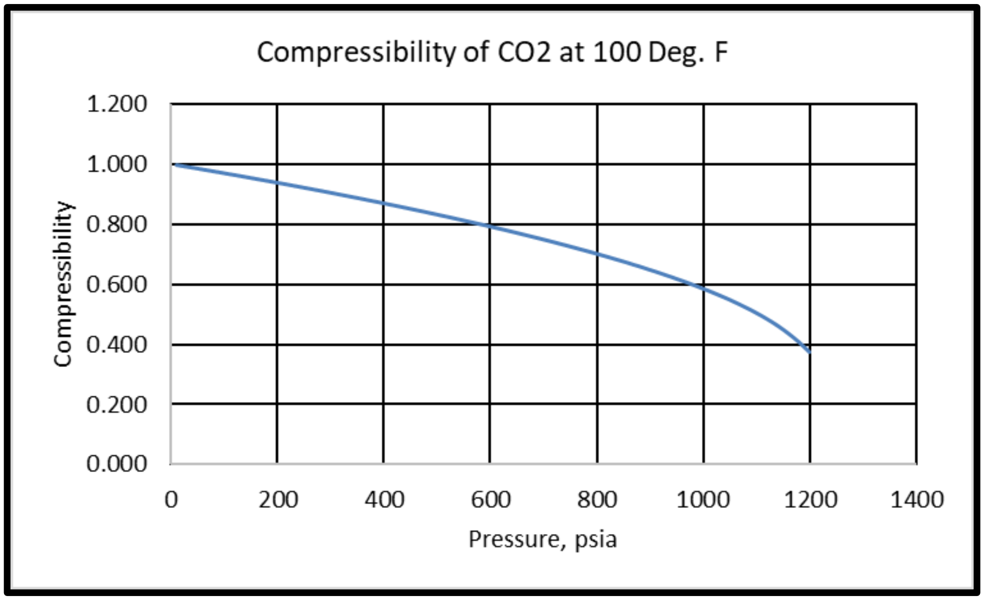


Figure 5: Compressibility of Dry CO2 at 100°F as a Function of Pressure

REAL GASES

For real gases, the equation of state includes the compressibility factor and the modified volume flow from the ideal gas equation becomes:

$$Q = wZRT / (144 * P) \tag{21}$$

For CO2 compressors, compressibility has to be included in thermodynamic performance calculations. The other issue with real gases is that the specific heat used in the previous equations was assumed to be constant. There are some gases where the specific heat is highly non-linear and/or changes significantly between the inlet and discharge conditions of a compressor stage. In these cases, simply averaging the inlet and discharge specific heat values is not good enough and real gas calculations have to be performed. Per Sandberg, "…Regardless of thermodynamic model used, isentropic discharge conditions must be estimated and used in a number of the calculations." The real gas equations for the different thermodynamic processes are shown below. The superscript **'** is the condition at discharge pressure with entropy equal to inlet entropy. As Sandberg describes, this can be an iterative process where the discharge temperature is varied until the discharge entropy equals the inlet entropy.

Isentropic Work (Head):

$$W_s = \frac{n_s}{n_s - 1} f 144 P_i v_i \left[ \left( \frac{P_d}{P_i} \right)^{\frac{n_s - 1}{n_s}} - 1 \right] \tag{22}$$

Isentropic Exponent:

$$n_s = ln\left(\frac{P_d}{P_i}\right) / ln\left(\frac{v_i}{v'_d}\right) \tag{23}$$

Polytropic Work Factor:

$$f = \frac{(h'_d - h_i)J}{\frac{n_s}{n_s - 1}} 144(Pv'_d - P_i v_i) \tag{24}$$

Isentropic Efficiency:

$$\eta_s = \frac{W_s/_J}{h_d - h_i} \tag{25}$$

Polytropic Work (Head):

$$W_p = \frac{n}{n-1} f 144 \rho_i v_i \left[ \left( \frac{\rho_d}{\rho_i} \right)^{\frac{n-1}{n}} - 1 \right] \quad (26)$$

Polytropic Exponent:

$$\mathrm{n} = ln\left(\frac{P_d}{P_i}\right) / ln\left(\frac{v_i}{v_d}\right) \quad (27)$$

Polytropic Efficiency:

$$\eta_p = \frac{W_p/_J}{h_d - h_i} \quad (28)$$

OVERALL COMPRESSOR PERFORMANCE CURVES

Compressor stage performance is typically characterized by invariant parameters such as inlet volumetric flow versus head. An example is shown Figure 6. Head and efficiency are based on an isentropic model.

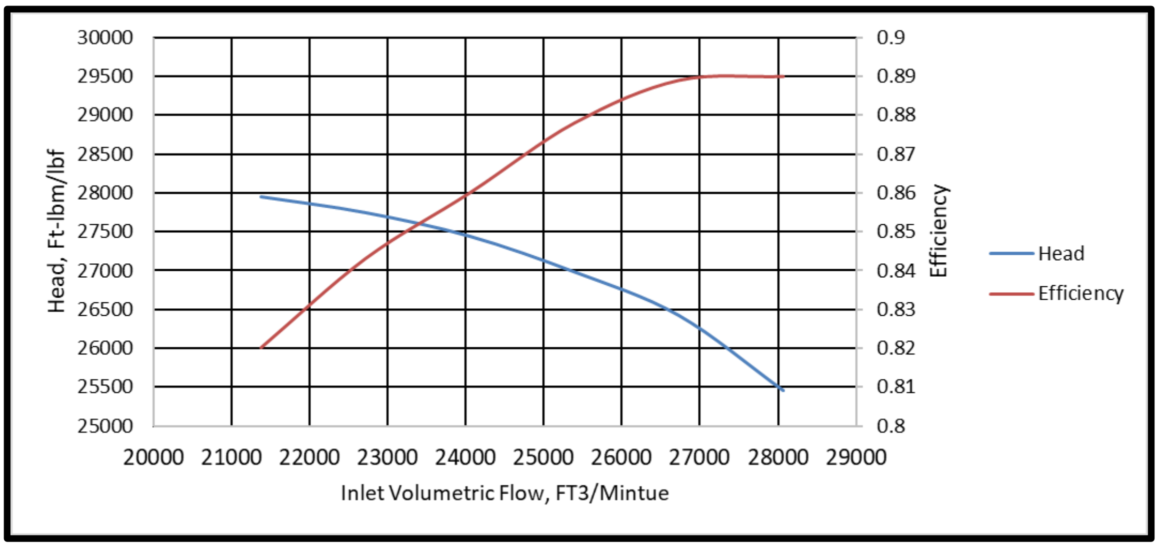


Figure 6: Compressor Stage Performance Curve

At flows below 21,400 $ft^3$/minute, the compressor is in an unstable region, such as surge or stall. Surge is an aerodynamic instability that results in violent flow oscillations. In this situation the flow through the stage stops, reverses in direction and then the flow stops again and returns to the normal direction of flow. The cycle continues until the back pressure on the stage is reduced to a point where the flow and head are at a stable operating condition on the curve. Stall is a localized flow reversal, but it is not a complete reversal in machine flow like surge. Operating in surge or stall can lead to mechanical failures and this will be illustrated in more detail in Case Study One. For this example, at flows above 28,000 ft3/minute the compressor is approaching stonewall or a choked flow condition. This occurs when sonic velocity is reached somewhere in the compressor stage and flow through the compressor cannot be increased further. Operation in a chocked flow condition can also lead to machine damage.

Typically, operators are interested in the compressor mass flow and discharge pressure. To convert the invariant compressor curve into a performance curve of mass flow versus discharge pressure, the compressor inlet pressure, inlet temperature and gas composition must be specified. Using the compressor curve shown in Figure 6, let's use the following conditions:

- Pi = 14.4 psia
- Ti = 80°F = 540.3°R
- Gas = Dry Air (MW = 28.96, R = 53.35 ft-lbf/(lbm-°R); k = 1.40)

At a flow of 21,400 ft3/minute, the head is 27,955 ft-lbf/lbm and the efficiency is 82%. Using equation (2), we can solve for the mass flow.

$w = Q*144*P_i / (RT_{in}) = 21{,}300*144*14.4 / (53.35*540.3) = 1539$ lbm/min (92,340 lbm/hour)

Recall that the compressor curve is based on an isentropic thermodynamic model. So, using equation (8) and a head of 27,955 ft-lbf/lbm, the discharge pressure can be calculated. Solving for $P_d$ yields,

$P_d = ((Head*(k-1)//(kRT)+1)^\wedge(k/(k-1)) * P_i = ((27955*(1.4-1)/(1.4*53.35*540.3)+1)^\wedge(1.4/(1.4-1)) * 14.4 = 33.9$ psia

We can also calculate the gas power using equation (11) and the efficiency.

Power = Mass Flow * $H_a / \mu_a$ = 1539 lbm/min * 27,955 ft-lbf/lbm / 0.82 / 33000 ft-lbf/min-BHP) = 1590 BHP.

If this process is repeated across the entire flow range, curves of mass flow versus discharge pressure and power can be generated. See Figure 7.

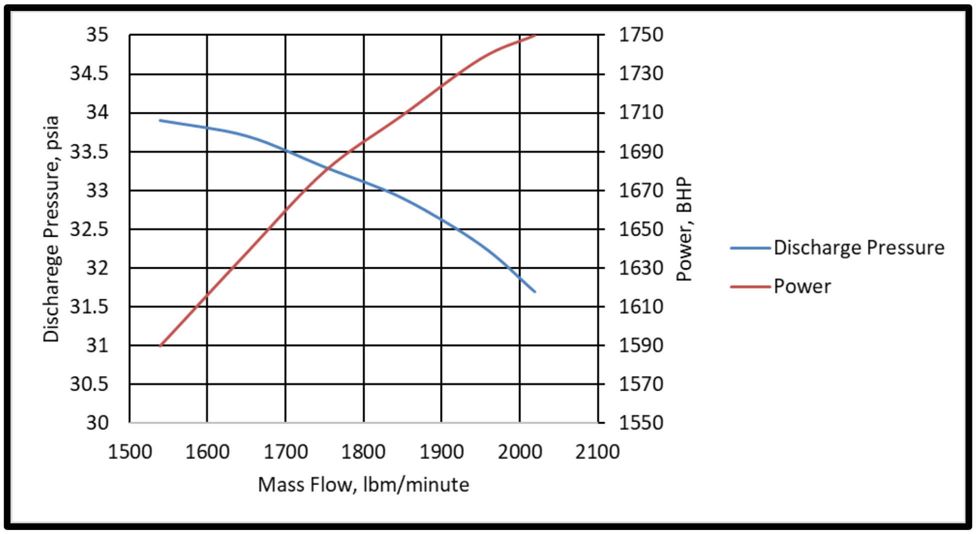


Figure 7: Performance Curve of Mass Flow Versus Discharge Pressure and Power

If the suction pressure or temperature is changed, the compressor invariant performance curves don't change appreciably (unless there are major changes in suction conditions which impact the losses within the impeller), but the curves of mass flow versus discharge pressure and power change. For example, if the inlet temperature is changed to 40°F, the compressor discharge pressure and power will change as depicted in Figure 8.

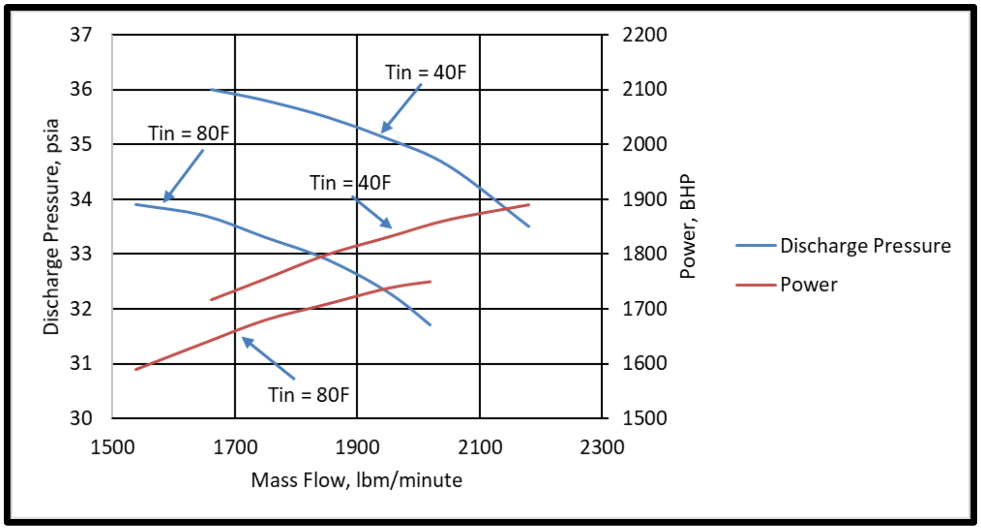

Figure 8: Effect on Compressor Performance with Lower Inlet Temperature

Note that these calculations were done on the basis that the stage curve was based on an isentropic model. If the curve was developed based on a polytropic model, the calculation process is the same except that equations (13), (15), (16) and (17) are used to determine the discharge pressure and power.

Many compressors consist of multiple stages. In-between the stages intercoolers are used to cool the gas before going into the next stage. In these machines, the discharge pressure and temperature in the first stage is determined. The pressure drop across the intercooler and piping is subtracted from this to determine the inlet pressure to the next stage. The gas temperature downstream of the intercooler would be the inlet temperature to the next stage and then the same calculation process is repeated. This process is repeated for all the stages. For the compressor to work all the stages have to able to stack together. The stages have to be designed such that they operate together over the flow range. Typically, one stage may set the low flow condition for the compressor and another stage might set the high flow limitation.

The powers of all the stages are added together to determine the total gas power. The total machine power is the sum of the total gas power and the mechanical losses. Mechanical losses include bearing losses, gear losses, etc.

A typical three stage integrally geared compressor cross section is shown in Figure 9. Many of these machines have laby shaft seals where the shafts extend through the scrolls. Some gas is lost through these seals. If the flow is measured at the discharge of the compressor as shown in Figure 1, then the mass flow at the first stage of the compressor will be the delivered flow plus the flow lost through all the stage shaft seals. The mass flow at the inlet of stages two and three also must take into account the seal losses of its stage and the downstream stages.

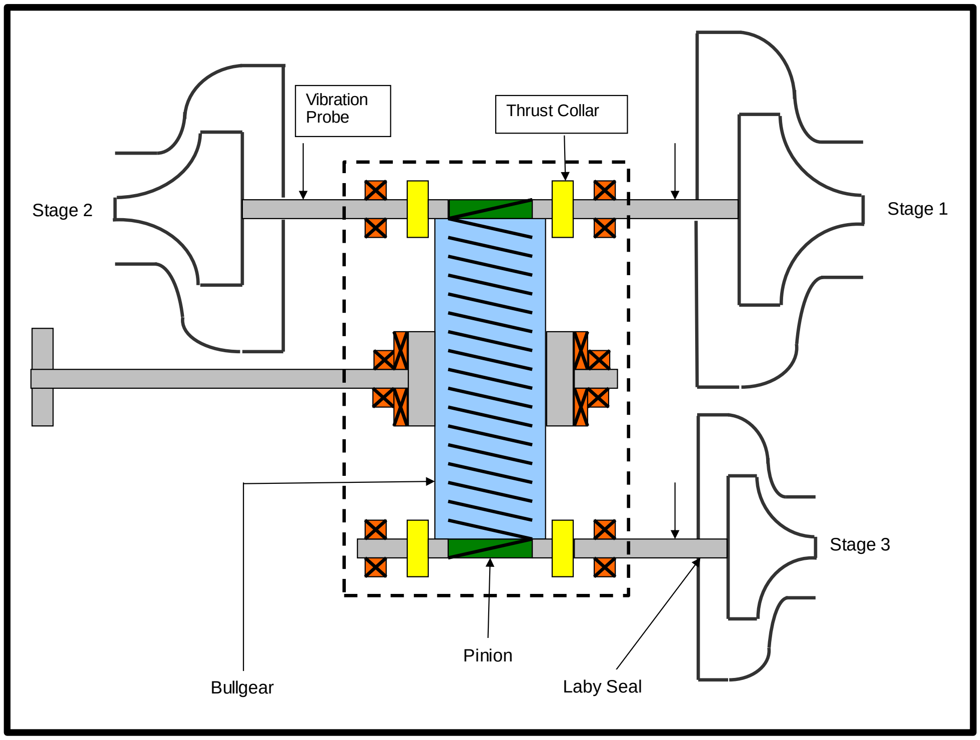

Figure 9: Three Stage Integrally Geared Compressor Cross Section

Atmospheric air is a mixture of gases that is partially saturated by water vapor. When atmospheric air is compressed and then cooled in a cooler, some of the water vapor can be condensed in the cooler. This condensate is removed from the system by condensate traps or some other means such as an orifice, partly open valve, or on/off blow down valves. Therefore, the mass of gas and water vapor entering the first stage of the compressor will be different than the amount leaving the compressor. If flow is measured downstream of the compressor, then the flow going into the first stage has to be adjusted by the amount of water vapor entering the machine. The flow going into the downstream stages has to be adjusted by the condensate removed in the upstream intercooler. This same approach applies to hydrocarbon compressors that have vapors that condense in the intercoolers.

Many compressors also use a moveable inlet guide valve (IGV) on the first stage to control the capacity of the compressor. IGVs are a series of blades circumferentially arranged at the compressor inlet. The IGV blade assembly can be adjusted with an actuator, which changes the angle of the gas entering the impeller. This adds pre-whirl to the gas which is used to control the capacity. This can effectively reduce the power consumption to increase the compressor's overall efficiency while avoiding surge to provide a better turndown. Its effect on performance is shown in Figure 10. Note that some machines use IGVs on multiple stages to achieve even better machine turndown.

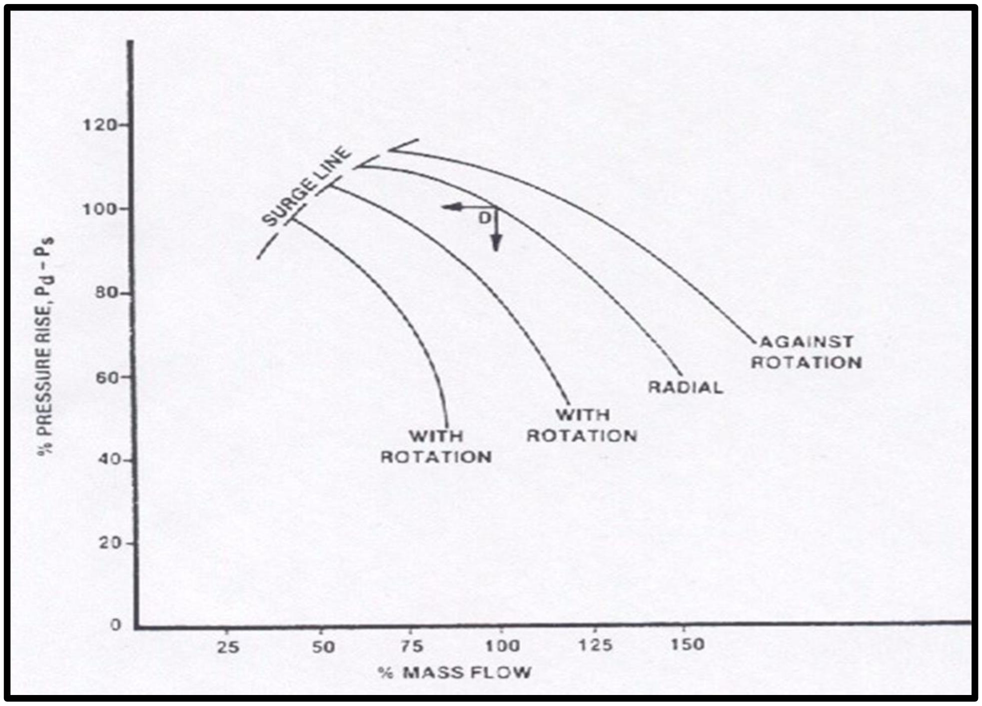

Figure 10: Effect of IGV on Compressor Performance

**A Note on Isothermal Calculations**

The isothermal head and efficiency equations were shown earlier. The use of these equations can be used in many instances, but there are some further assumptions that can be made that improves the use. The equations are restated here:

$$H_{iso} = R/MW * T * \ln(rp) \tag{19}$$

$$\eta_{iso} = \text{Mass Flow} * H_{iso} / \text{Total Compressor Power} \tag{20}$$

A couple of things that need to be considered include:

- Temperature: Normally the inlet temperature to the first stage of a compressor is used in the isothermal head calculation. An improvement is using an average inlet temperature of all the stages. This accounts for instances where the inlet temperature to the first stage changes differently compared with the inlet temperatures to the other stages. For example, the inlet temperature to an atmospheric air compressor may range from 20°F to 100°F depending on the time of the year, but cooling water may only change from 50°F to 90°F. The inlet temperature to the downstream stages is based on cooling water temperature and so these temperatures don't change as much as the inlet temperature to the first stage.
- Gas constant, R/MW: With condensation, the molecular weight of the gas will change with each stage. Consideration needs to be made on what is best MW to use. For the atmospheric air, multi-stage compressor example, the gas constant changes through the machine.
- Mass Flow: Mass flow is normally assumed as mass flow delivered or at inlet. Consideration has to be done to be consistent in any analysis. Delivered flow will already have seal gas leakages "removed", while inlet does not.

For a comparison of performance in the field, using consistent assumptions will make the analysis more accurate.

**INSTRUMENTATION**

ASME PTC-10 provides requirements for shop testing of centrifugal compressors. This instrumentation typically includes:

- Four static pressure instruments on the inlet of each stage (except for an atmospheric inlet); spaced 90° apart
- Four temperature instruments on the inlet of each stage; spaced 90° apart and 45° apart from the pressure instruments
- Four static pressure measurements on the discharge of each stage; spaced 90° apart

- Four temperature instruments on the discharge of each stage; spaced 90° apart and 45° apart from the pressure instruments
- Flow measuring device that can be located on either the inlet or discharge side of the compressor.

ASME PTC-10 provides requirements on the locations of the pressure and temperature measuring points based on a minimum straight run of piping upstream and downstream of the compressor stage. The code does allow the use of flow straighteners in case the straight run of piping cannot be achieved, but it is required that the OEM and end user mutually agree on this.

The test gas composition has to be measured, compressor speed has to be measured, and the shaft power has to be measured using either a torque measurement or by electrical measurements. In addition, unless the mechanical losses are well known and documented by previous tests, the mechanical losses (integral gears, bearings, and seals) can be calculated by measuring the cooling fluid flow and temperature rise.

ASME also makes reference to ASME PTC 19.3 for guidance on the instruments used and the calibration requirements.

In the field, the instrumentation is not the same as the instrumentation used in a shop test. Compressors are instrumented to be able to operate, monitor the condition and protect the machine from mechanical damage. Maintaining good compressor mechanical and thermodynamic performance can be achieved by having a good machinery condition monitoring program, a good machinery protection system and good maintenance practices. The purpose of a machinery condition monitoring system is to identify machine degradation early so that action can be taken before it progresses to a point of machine failure. Sometimes the signs of progressive damage are very subtle and difficult to detect. In these cases, a machine may appear to suddenly fail without warning. However, a robust condition monitoring system that detects thermodynamic and/or mechanical degradation at an early stage can allow for action to be taken before the condition worsens to a point that causes a machine failure. And, as described above, the pressures, temperatures, flow, cooling water, IGV position, etc., all affect the thermodynamic performance. So, how can thermodynamic performance degradation be detected?

## EQUIPMENT CONDITION MONITORING

It wasn't too long ago that compressor condition monitoring was largely based on operators taking log sheets and looking for signs of degradation. In many cases, compressor performance degradation was not detected until the compressor couldn't provide enough flow for the process or the compressor went into an alarm condition. As described by McCusker, "The problem is that early warning of equipment failures is difficult to achieve with conventional process alarm management techniques, as the alarm settings are fixed and need to be suitable for all operating conditions. As such, alarms are set broadly so when they are eventually triggered, the equipment is already on the way to failure." As further described by McCusker, "By using multi-variable data points fed into a condition algorithm, the abnormal or alarming points can be 'conditioned' to reflect the natural variability that is associated with the plant's specific operating condition." So, how do we use this concept to monitor the thermodynamic performance of a compressor?

While isentropic and polytropic models are good for evaluating the stage performance of a compressor, these models do not work well when evaluating the overall thermodynamic performance of a multi-stage compressor with intercooling. In this case, an isothermal process may be a better means to evaluate overall compressor thermodynamic performance. Isothermal efficiency will vary at different conditions. For example, Figure 11A shows how isothermal efficiency varies for a particular three stage air compressor along the wide open IGV curve. Figure 11B shows how isothermal efficiency varies for the same machine if it is turned down at constant discharge pressure (different IGV angles). The performance is based on a specific inlet pressure and temperature, and cooling water temperature.

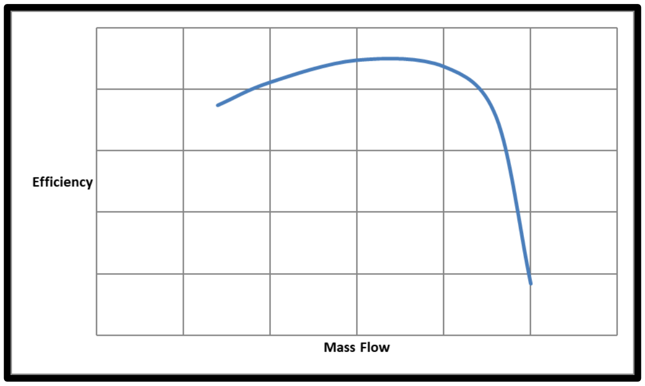


Figure 11A: Flow Versus Isothermal Efficiency – 100% Open IGV

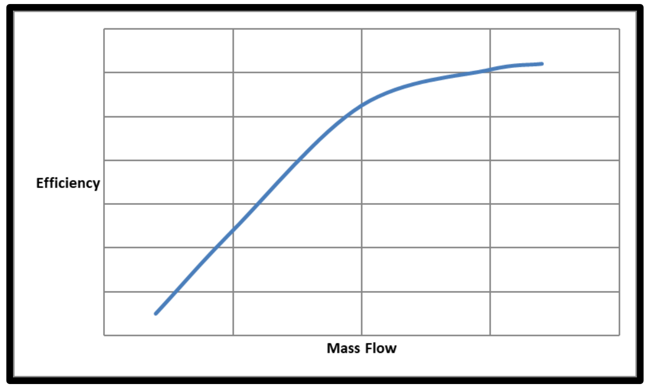


Figure 11B: Flow Versus Isothermal Efficiency – Constant Discharge Pressure (Varying IGV Angles)

Having some key process variables available from the DCS and then ultimately stored in a data historian for long term use will allow isothermal efficiency to be calculated and trended for continuous evaluation. The key parameters needed to perform the calculation include suction temperature, suction and discharge pressure, power, and flow of the gas. If the gas is wet, such as with an atmospheric air compressor, the effects of moisture should be included in the flow and MW. To determine this, the dry bulb temperature, barometric pressure, and relative humidity need to be known. Having weather data available on a continuous basis for an atmospheric air compressor will allow for a more accurate calculation to be performed. If weather data is not available, atmospheric pressure and relative humidity can be represented by a constant value based on a location's average conditions. There are programs available today that will calculate the wet properties based on this information. For an atmospheric air compressor, the following equations can also be used:

$$SH = 0.6219 * (RH/(p_i/p_{sat} - RH)) \tag{28}$$

$$w_{wet} = (1 + SH) * w_{dry} \tag{29}$$

$$MW\ (wet) = (P_{sat} * RH/P_i) * 18.01 + (1 - (P_{sat} * RH/P_i)) * 28.96 \tag{30}$$

As described above, isothermal efficiency will vary based on the operating conditions, IGV position, etc.

Figure 12 is a two year trend of an air compressor's isothermal efficiency utilizing weather data, but excluding some of the mechanical losses such as air seal leaks. As shown, the isothermal efficiency does vary with changes in temperature and conditions. Even so, by monitoring the trend over time, a decrease in the operating upper and lower peak isothermal efficiencies can be observed.

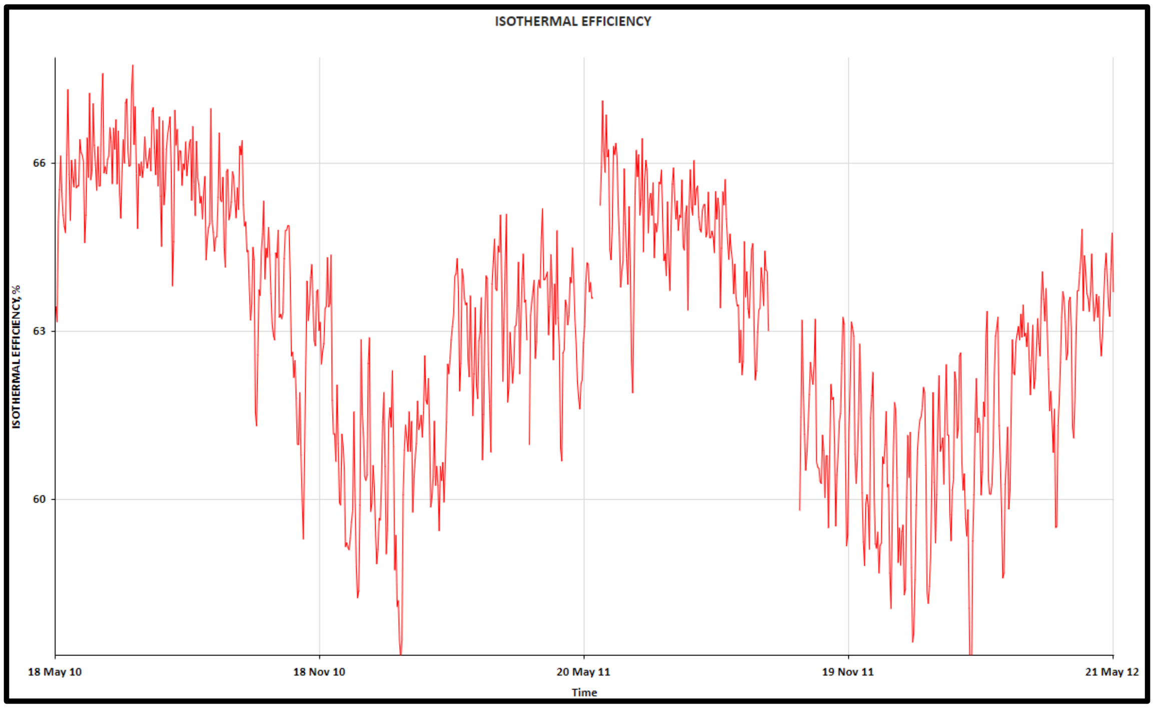

Figure 12: Isothermal Efficiency Trend

In addition to calculating, trending and then alerting on a decreasing isothermal efficiency, using statistical multivariate analysis techniques to predict machine flow is another way to effectively detect and quantify machine performance degradation. As described by Verderame and Metha, if enough data is taken when the machine is in good condition, the machine performance can be accurately predicted using state-of-the-art statistical multivariable techniques based on the actual operating conditions. Machine degradation can then be detected by comparing the predicted performance metric versus the actual.

Evaluating the flowrate of the machine as a function of inlet guide vane position, power, suction temperature, discharge pressure, and cooling water temperature, or even a smaller subset of these variables, is a powerful performance metric that can be used to notice very early machine degradation. A predictive model for compressor flow using multivariate regression or other machine learning techniques can be developed with the inclusion of the mentioned process and machine variables. Once the flow model is developed based on healthy and variable historical operating conditions over a suitable period of time, the predicted flow value can be compared to the current actual flow value. This comparison results in a residual trend of the predicted minus the actual flow value. An increasing residual will then signify clearly the degradation to machine performance. In fact, in many instances machine degradation can be seen more clearly in the residual flow trend than the isothermal efficiency trend. As shown in Figure 13, there was an incident, where the calculated residual flow increased.

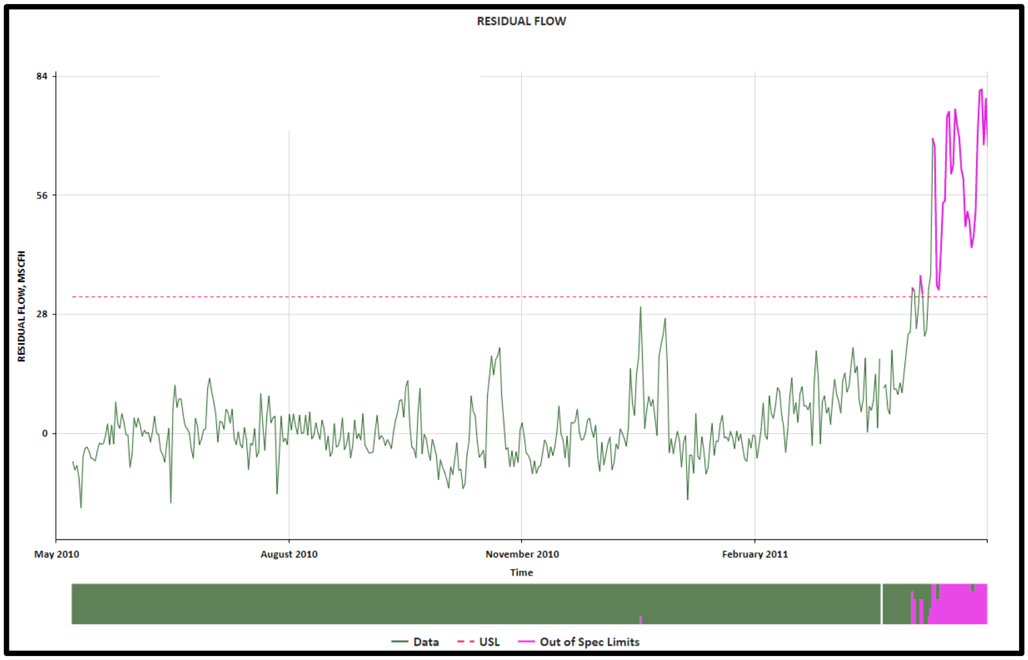

Figure 13: Residual Flow Trend

This analysis will clearly show an overall performance degradation. In order to pinpoint where the degradation has occurred on the compressor, one method to determine this is to perform stage by stage field performance testing. This will be illustrated in the subsequent case studies.

## CASE STUDY ONE

This first case study pertains to a three stage integrally geared centrifugal compressor driven by a 1780 RPM, 5000 HP induction motor. The gearbox consists of a bullgear and two rotors. The low speed (LS) rotor consists of a pinion with overhung impellers mounted at both ends. The high speed (HS) rotor consists of a pinion with an overhung impeller mounted at one end. All the impellers are a semi-open type. In this design the front side of the impeller is open, and the vanes run against a close clearance, non-contacting stationary shroud. See Figure 9. There are also vaned discharge diffusers on every stage.

The P&ID for this compressor is shown in Figure 1. The design data is shown in Table 1. Note that the seal losses are approximated as a percentage of overall flow for each stage. The composition of dry air is approximated as 78% nitrogen, 21% oxygen and 1% argon.

| MECHANICAL DESIGN DATA | | | |
|---|---|---|---|
| Number of Stages | 3 | | |
| Compressor Mechanical Losses, BHP | 112 | | |
| Motor Efficiency | 0.98 | | |
| Compressor Input Speed, RPM | 1780 | | |
| STAGE | 1 | 2 | 3 |
| Number of impellers per stage | 1 | 1 | 1 |
| Impeller Diamter Stage 1, inches | 22 | 18.3 | 13.7 |
| Rotor Speed, RPM | 13363 | 13363 | 18844 |
| Seal Losses (% of dry delivered flow) | 0.30% | 0.50% | 0.60% |
| Suction Pipe - Inside Diameter, Inches | 30 | 18 | 16 |
| Discharge Pipe - Inside Diameter, Inches | 16 | 14 | 18 |
| | | | |
| **INLET GAS COMPOSITION** | | | |
| Dry Gas Components | Composition, Mole Fraction | | |
| N2 | 78 | | |
| O2 | 21 | | |
| AR | 1 | | |

Table 1: Case Study One Design Data

Table 2 is the measured data. The barometric pressure and relative humidity were not measured in the field, but were obtained from a local weather service at the time of the test. The interstage pressures were measured with a temporary pressure transmitter. The interstage temperatures were measured with temporary thermocouples. Only single interstage pressure and temperature measurements were taken based on available NPT connections. The compressor was not taken offline during the test and only data from a single operating point was taken.

| MEASURED DATA | | | |
|---|---|---|---|
| **SITE CONDITIONS** | | | |
| Barometric Pressure, psia | 14.3 | | |
| Cooling Water Temperature, F = | 75 | | |
| Ambient Relative Humity | 40.0% | | |
| | | | |
| **MEASURED SPEED, FLOW AND POWER** | | | |
| Compressor Actual Input Speed, RPM | 1780 | | |
| Delivered Flow, lbm/hour | 91863 | | |
| Motor Power (Kw) | 3322 | | |
| | | | |
| **MEASURED STAGE DATA** | | | |
| STAGE | 1 | 2 | 3 |
| Inlet Guide Vane Opening, % | 105% | | |
| Inlet Pressure, psig | -0.650 | 16.7 | 41.2 |
| Discharge Pressure, psig | 17.6 | 42.0 | 100.8 |
| Inlet Temperature, F | 86.0 | 95.4 | 84.7 |
| Discharge Temperature, F | 258.7 | 234.5 | 244.2 |

Table 2: Case Study One Measured Data

The supplier stage curves were based on an isentropic model and so isentropic stage calculations were performed. The stage curves were shown as Q/N verses $\mu_s$ and Q/N verses $\eta_s$ where Q is the volume flow in ACFM (Actual cubic feet per minute) and N is the pinion speed in RPM. These curves were converted to ACFM verses isentropic head using equation (9), and ACFM verses isentropic efficiency. As noted, there is an IGV on the first stage. Shop test data was not available at different IGV angles. So, to compare with field performance with shop test data, it was necessary to test with the compressor with the IGV at the 100% open position. This was done and the intermediate calculations are shown in Table 3 and the overall results in Table 4.

| | | | |
|---|---|---|---|
| Inlet Static Pressure, psia | 13.7 | 31.0 | 55.5 |
| Outlet Static Pressure, psia | 31.9 | 56.3 | 115.1 |
| | | | |
| Saturation pressure at inlet temperature (Psv) , psia = | 0.6151 | 0.8254 | 0.5901 |
| Partial pressure of water vapor | 0.2460 | 0.5750 | 0.5986 |
| Inlet specific humidity (w), # H2O/# air = | 0.01141 | 0.01175 | 0.00678 |
| Inlet humidity ratio/ lbm-mole water/lbm-mole dry gas | 0.01835 | 0.01889 | 0.01090 |
| | | | |
| Dry gas MW | 28.97 | 28.97 | 28.97 |
| ACTUAL COMPOSITION | ACTUAL FLOW | ACTUAL FLOW | ACTUAL FLOW |
| N2 | 2508 | 2501 | 2488 |
| O2 | 675 | 673 | 670 |
| AR | 32 | 32 | 32 |
| wa | 58.99 | 60 | 34 |
| | | | |
| Wet gas molecular weight | 28.77 | 28.77 | 28.86 |
| | | | |
| Inlet Wet Gas Specific Heat at Constant Pressure, BTU/lbm-R | 0.243 | 0.243 | 0.243 |
| Inlet Wet Gas Specific Heat Ratio | 1.398 | 1.396 | 1.396 |
| | | | |
| Discharge Wet Gas Specific Heat at Constant Pressure, BTU/lbm-R | 0.245 | 0.245 | 0.245 |
| Discharge Wet Gas Specific Heat Ratio | 1.393 | 1.393 | 1.391 |
| | | | |
| Average Wet Gas Specific Heat at Constant Pressure, BTU/lbm-R | 0.244 | 0.244 | 0.244 |
| Average Wet Gas Specific Heat Ratio | 1.395 | 1.394 | 1.393 |
| | | | |
| Static Specific Volume at Inlet, ft3/lbm | 14.92 | 6.69 | 3.65 |
| Static Specific Volume at Discharge, ft3/lbm | 8.405 | 4.602 | 2.276 |
| | | | |
| Dry Mass Flow, lbm/hour | 93149 | 92873 | 92414 |
| Wet Mass Flow, lbm/hour | 94212 | 93964 | 93040 |
| Average Velocity at Inlet Flange, ft/sec | 79.6 | 98.8 | 67.6 |
| Average Velocity at Discharge Flange, ft/sec | 157.6 | 112.4 | 33.3 |
| Fluid MACH Number at Inlet Flange | 0.0693 | 0.0853 | 0.0591 |
| Fluid MACH Number at Discharge Flange | 0.1197 | 0.0869 | 0.0256 |
| | | | |
| Total Temperature at Inlet, R | 546.8 | 556.5 | 545.4 |
| Total Temperature at Outlet, R | 721.0 | 695.8 | 704.6 |
| | | | |
| Total Pressure at Inlet, psia | 13.70 | 31.16 | 55.64 |
| Total Pressure at Outlet, psia | 32.22 | 56.60 | 115.15 |
| | | | |
| Total Density at Inlet, lbm/ft3 | 0.06717 | 0.15013 | 0.27436 |
| Total Density at Outlet, ibm.ft3 | 0.11983 | 0.21810 | 0.43954 |
| | | | |
| Sum of squares of tip speeds | 1645463 | 1138531 | 1268884 |
| | | | |
| Power | 1571 | 1255 | 1418 |
| | | | |
| k | 1.395 | 1.394 | 1.393 |

Table 3: Case Study One Intermediate Calculations

| RESULTS | | | |
|---|---|---|---|
| STAGE | 1 | 2 | 3 |
| Flow Coefficient = | 0.04518 | 0.03503 | |
| Isentropic Efficiency | 0.861 | 0.735 | 0.781 |
| Isentropic Work Coefficent | 0.556 | 0.549 | 0.598 |
| | | | |
| ACFM = | 23378 | 10431 | 5652 |
| Isentropic Head, ft-lb/lb | 28423 | 19432 | 23580 |
| Power, BHP | 1572 | 1255 | 1419 |
| | | | |
| Sum of Stage BHP + Frictional Losses, BHP = | 4358 | 1255 | 1419 |
| Measured Compressor Power, BHP = | 4366 | | |
| Overall Isothermal Efficiency = | 0.680 | | |
| | | | |
| Cooler Approach Temperature, F | 20.4 | 9.7 | |
| Interstage Pressure Drop, psi | 0.90 | 0.80 | |

Table 4: Case Study One Field Performance Results

The measured compressor power is the measured motor power multiplied by the motor efficiency. As shown, the sum of the calculated stage power plus the mechanical losses matches very well with the measured compressor power. This provides some confidence in the test results. The first stage intercooler approach temperature is slightly higher than design; 20.4°F versus a design of 13.3°F. The second stage approach temperature and interstage pressure drops match the design values. The calculated volumetric flow, isentropic head and isentropic efficiency for each stage are shown in Figures 14 to 16.

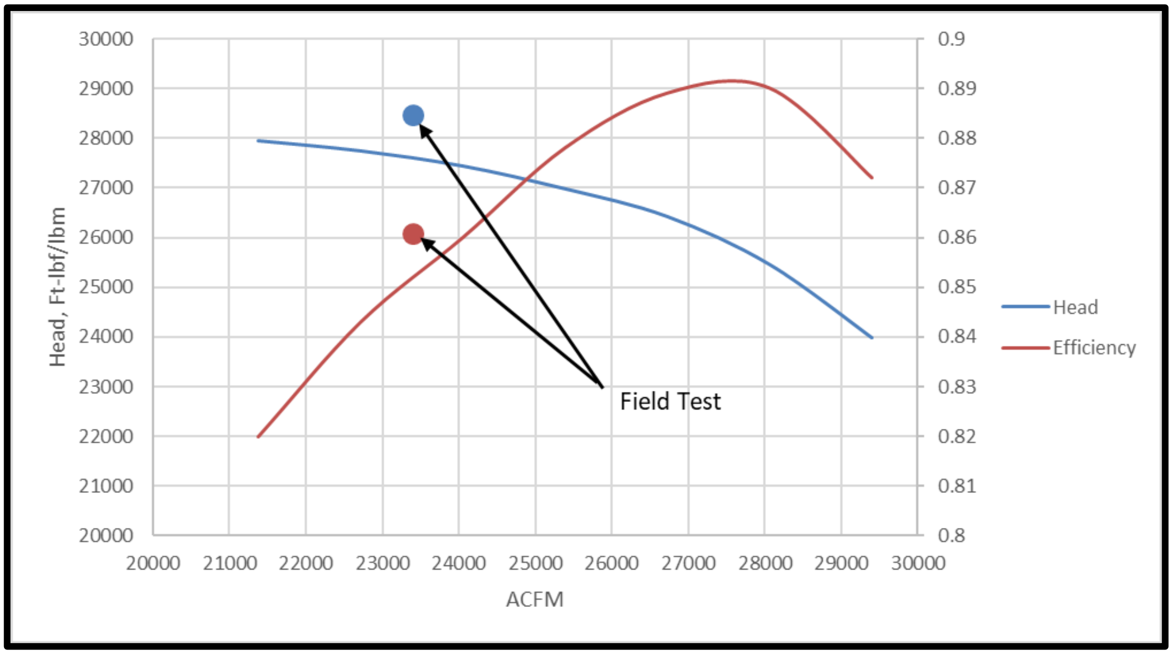

Figure 14: Case Study One – Stage One Field Test Results

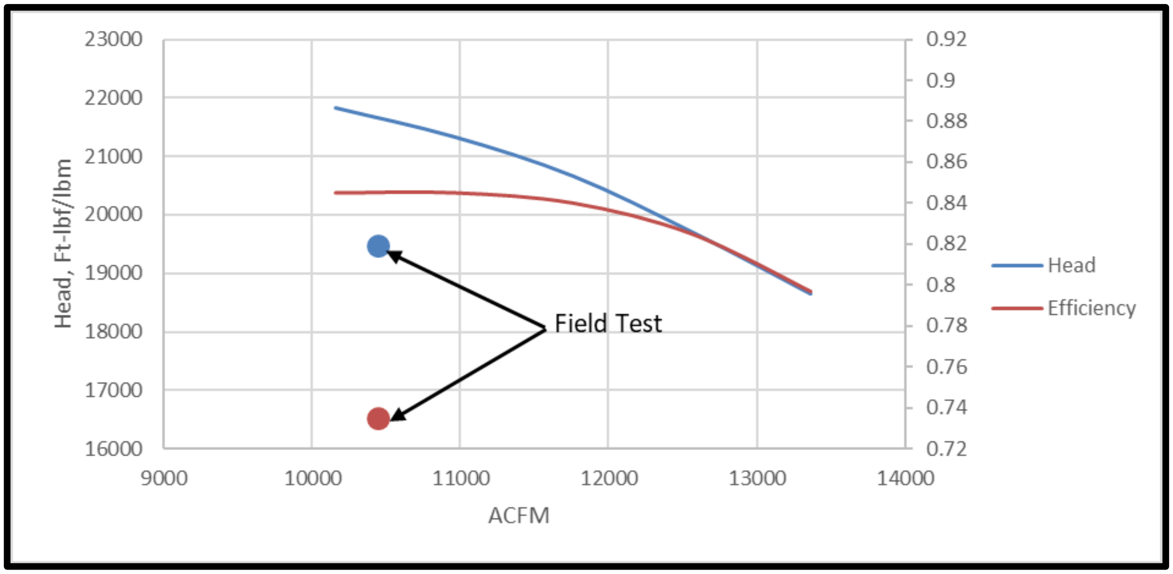

Figure 15: Case Study One – Stage Two Field Test Results

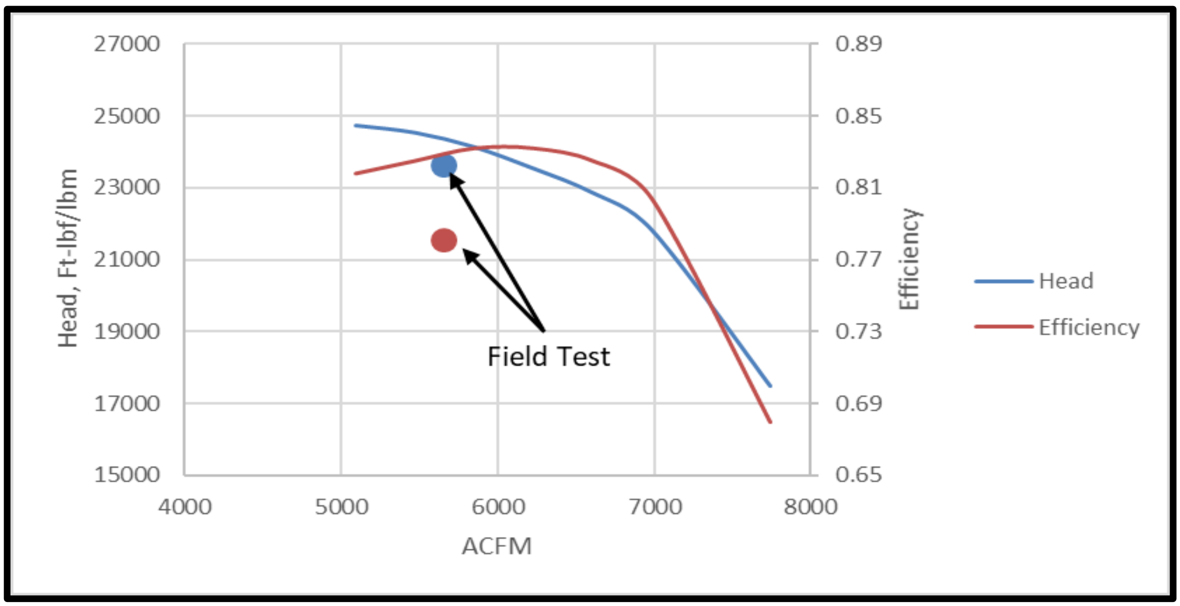

Figure 16: Case Study One – Stage Three Field Test Results

Table 5 compares the field test results for each stage with the shop test curves. The stage one field tested head and efficiency was slightly better than the shop testing curves. This could be due to the accuracy of the field measurements. Remember that only single instruments were used, and the locations in the field could have been different that those during the shop test. The second stage clearly shows degradation and the third stage shows slight degradation.

| STAGE | % Shop Test | |
|---|---|---|
| | Head | Efficiency |
| 1 | 102.5% | 100.9% |
| 2 | 89.0% | 86.4% |
| 3 | 97.1% | 94.1% |

Table 5: Comparison of Field and Shop Test Results

Based on the field test, it was decided to inspect the second stage. Unfortunately, before the planned shutdown, the second stage vibration started increasing. A spectrum analysis showed that all the vibration was at a frequency corresponding to the pinion one operating speed.

It was concluded that this was most likely due to unbalance. The first and third stage vibrations were stable until the compressor suddenly tripped on high first stage vibration. An inspection of the first stage impeller revealed that a piece was missing from one of the impeller inlet vanes. See Figure 17. Besides this, the second stage impeller and inlet shroud were found to be heavily coated with white mineral deposits. See Figure 18. This type of fouling was indicative of cooling water contamination and a check of the first intercooler revealed that several tubes were leaking. This fouling also explained the second stage performance problem.


Figure 17: First Stage Blade Failure


Figure 18: Second Stage Fouling

Several first stage intercooler tubes were plugged, which corrected the tube leaks. The first and second stage impellers were cleaned, and the rotor was balanced. Because of the broken impeller vane, a similar amount of blade material had to be removed from a blade on the opposite side of the impeller to be able balance the rotor. This was so that the compressor could be put back in service quickly as a spare rotor was not available. The third stage was not also inspected due to time and manpower constraints.

As explained by Smith (2012), the degradation on the second stage performance due to the fouling likely pushed the first stage into a low flow condition where stall cells formed in the first stage impeller. Stall can cause alternating forces which can excite natural modes of vibration and cause high alternating stresses in impeller blades. In this machine, the margin between the diffuser excitation frequency and the impeller natural frequency mode associated with this blade failure was only 3.8%.The combination of operating in stall and not having much margin between the diffuser excitation frequency and a destructive impeller natural frequency, could have caused high alternating blade stresses which led to the blade failure.

Once the machine was restarted, most of the performance was restored, but because of the damage to the first stage impeller and suspected issues with the third stage, some degradation was still evident. This is shown as the “degraded state” in Figures 19 and 20. These issues contributed to an approximate 4-5% decrease of flow in the summer months and approximately a 2% shortfall in the winter.

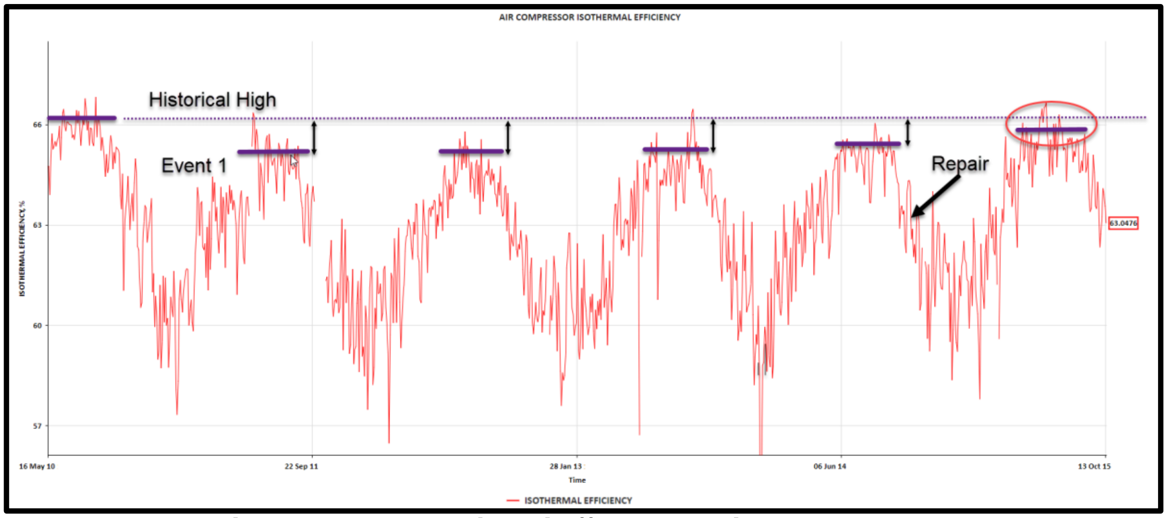

Figure 19: Case Study One Long Term Isothermal Efficiency Trend

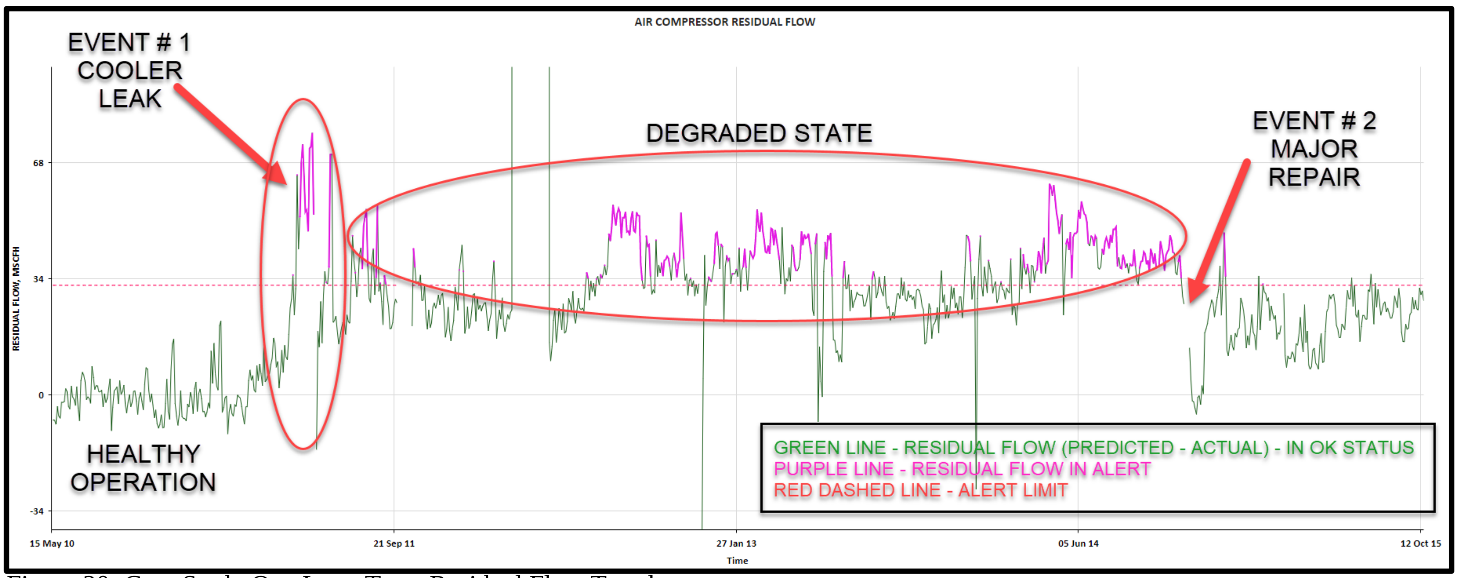

Figure 20: Case Study One Long Term Residual Flow Trends

The machine continued to run until a planned outage in November 2014, when the stage one and two rotor was replaced. Additionally, severe wear was found on the first stage diffuser vanes in conjunction with obvious impact damage on three of the vanes. The third stage impeller was also inspected and had a significant amount of hard, black fouling on the blades. The impeller was cleaned and was in excellent condition with very little wear. The inlet shroud and diffuser vanes were cleaned, but the diffuser vanes were eroded, and nothing could be done to address this because a spare was not available.

Even though the low speed rotor was replaced, and the diffusers were cleaned, performance improved but still did not get back to the desired state, as can be seen in the residual flow trend in Figure 20. In order to restore performance, it became evident that the first, second and third stage vaned diffusers needed to be replaced due to the severe wear and impact damage found during the November 2014 outage.

## CASE STUDY ONE LESSONS LEARNED

The use of isothermal efficiency and residual flow predictive modelling can be an effective method to detect degradation in centrifugal compressor performance. The change in performance due to the cooler leak in Case Study One can clearly be seen. It is important that the right input parameters are chosen and large amounts of data over a range of operating conditions are used to create the models in order for the models to be able to accurately predict the machine performance. As shown, after the initial repair, even a small degradation in performance due to the first stage impeller damage and third stage fouling was still evident.

As shown in Case Study One, the isothermal efficiency and residual flow models do not explain the cause of the performance degradation. To determine the potential cause(s), stage by stage field performance testing was performed which showed the stage that was largely underperforming. This allows for a targeted internal inspection to identify and correct the problem.

The last lesson learned was that not addressing performance degradation can lead to a mechanical failure. In this case, it is likely that degradation in the second stage pushed the first stage into a flow condition that led to a blade failure. So, not only can degradation impact machine capacity and efficiency, it can also impact machine reliability.

**CASE STUDY TWO**

The second case study pertains to a four stage, dual service integrally geared centrifugal compressor driven by a 1787 RPM, 5000 HP induction motor. The gearbox consists of a bullgear and two rotors. The low speed rotor comprises the first two stages of a supplemental main air compressor (SMAC) service. The HS rotor comprises the third stage of the SMAC service and a single stage of compressed dry air (CDA) service. The CDA flow comes from multiple sources and is a higher delivered flow than the SMAC service. All the impellers are a semi-open type. In this design the front side of the impeller is open, and the vanes run against a close clearance, non-contacting stationary shroud. The compressor configuration is shown in Figure 21.

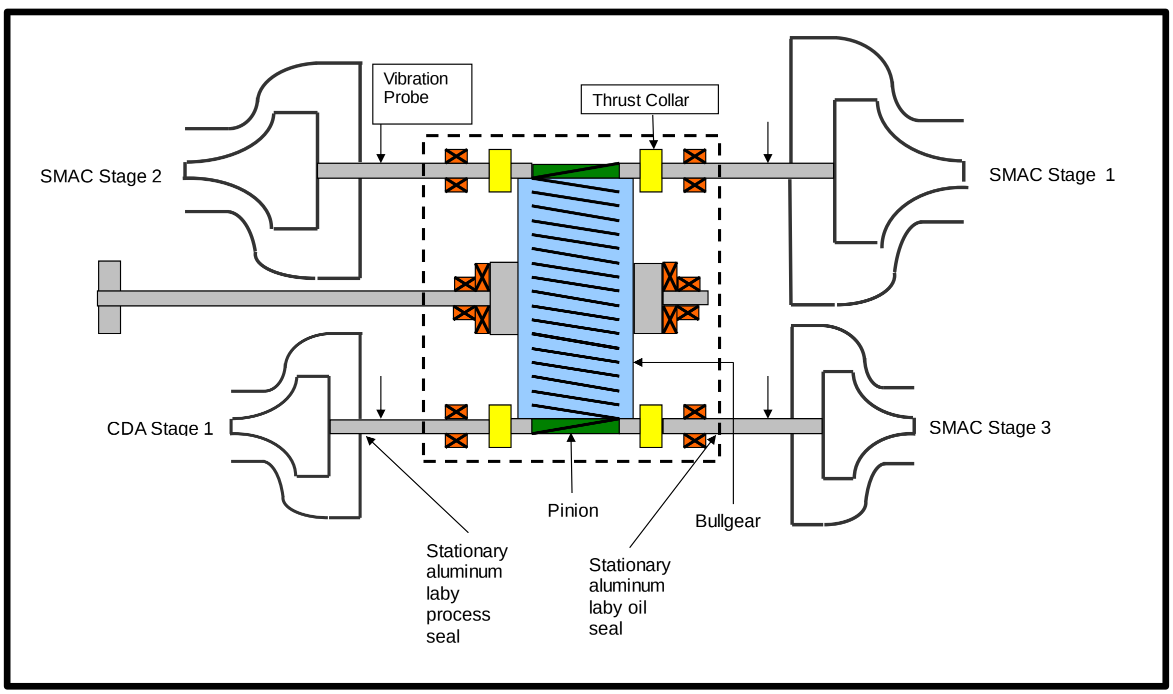


Figure 21: Compressor Configuration

The basic piping and instrumentation diagram (P&ID) for this compressor is shown in Figure 22. The SMAC control system includes a first stage IGV, a discharge flow meter, a discharge pressure transmitter, and a discharge vent valve. In this application, the SMAC capacity is controlled with the IGV and the machine discharge pressure is a function of the system back pressure. A DCS (Digital Control System) is used to operate and control the compressor. The control system also includes a surge avoidance system which opens the discharge vent valve if the machine operates within a predefined margin to surge. The CDA control system is similar except that rather than a discharge vent valve, a recycle valve is used. The recycle valve is used to recycle cooled discharge gas back to the suction so that the process gas is not lost when the valve is open.

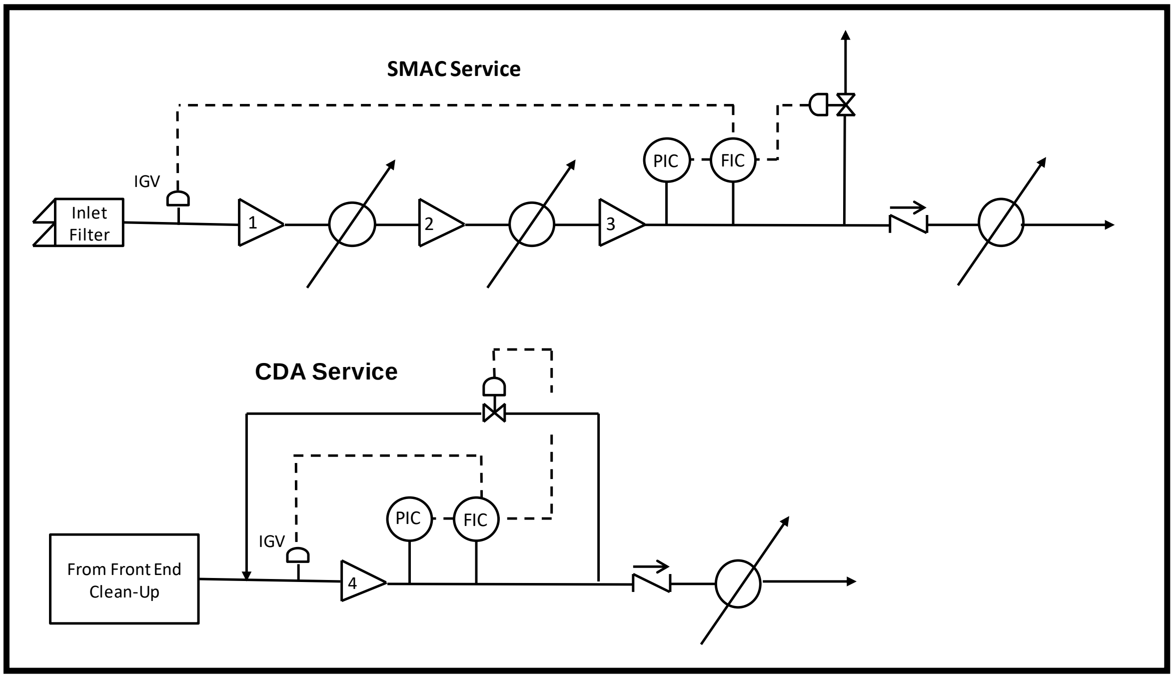

Figure 22: P&ID

This compressor was operated in continuous service for approximately 16 years before a performance problem was identified. During the summer months, the compressor was not able to maintain the same flow and pressure as it had in the past. This can also be seen in the residual flow model that is part of the Equipment Condition Monitoring (ECM) system. As shown in Figure 23, an increase in the residual flow started to occur in the summer of 2013.

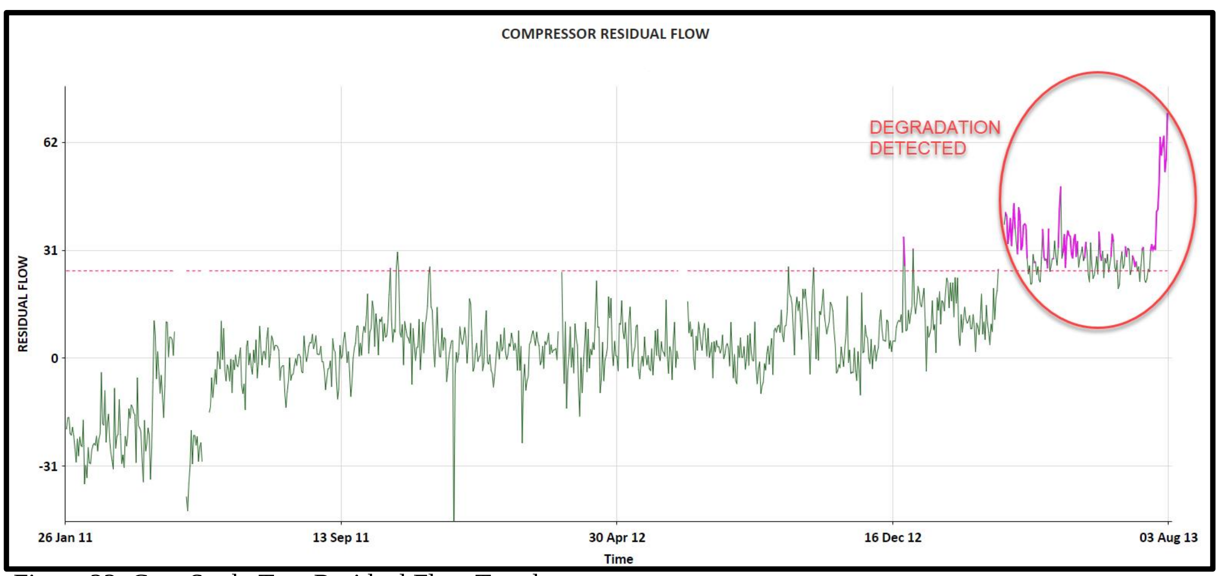

Figure 23: Case Study Two Residual Flow Trends

As with Case Study One, the residual flow trends show compressor performance degradation, but not the cause. To evaluate the cause, it was decided to perform a stage by stage field performance test. The compressor supplier shop tested stage curves were based on an isentropic process. As with Case Study One, it was not possible to take the compressors offline to take data over a complete range of flows, and the shop test data was not available at multiple IGV angles. So, field test data was taken at a single operating point with the IGV's 100% open. The results of the field test for the SMAC service compared with the shop test data are shown in Table 6.

| STAGE | % Shop Test | |
|---|---|---|
| | Head | Efficiency |
| 1 | 96% | 102% |
| 2 | 88% | 88% |
| 3 | 95% | 99% |

Table 6: Comparison of Field and Shop Test Results

As shown, there was a moderate amount of deviation from the shop tested curves for the first and third stages, but there was a large discrepancy for the second stage. This is further illustrated in Figure 24. Data from a sister machine showed similar first and third stage performance, but in this machine, the second stage field test matched well with the shop test. Minor deviations could be to the inaccuracy of the field data, which is something to be considered when evaluating field data. As with Case Study One, machines are not typically instrumented to do true ASME PTC-10 performance tests and many times compromises are made with the temporary field instruments. In any case, there was clear evidence of an issue with the second stage. Based on this, it was decided to inspect the second stage.

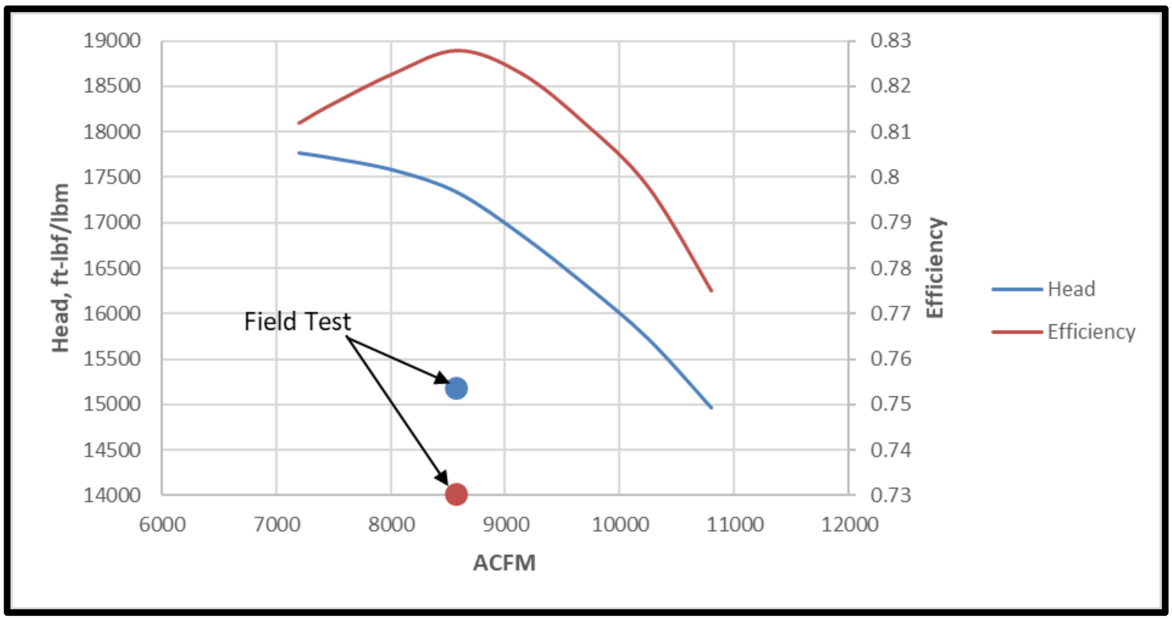


Figure 24: Case Study Two – Stage Two Field Test Results

As shown in Figures 25 and 26, there was some light fouling and some minor erosion of the leading edge of the diffuser vanes. The diffuser vane damage was reviewed with the compressor supplier, who advised that this erosion could account for the measured degradation in stage performance. The impeller was cleaned up and the diffuser was replaced. The diffuser was procured prior to the inspection based on the suspicion that there could have been diffuser damage. During the repair, several first stage intercooler tube leaks were discovered. Water carryover likely caused the fouling and erosion damage. The leaky tubes were plugged.


Figure 25: Second Stage


Figure 26: Second Stage Diffuser Vane

The machine was re-tested after the repair and as shown in Figure 27, the second stage head was slightly higher than the shop tested head. But the efficiency tested much higher than the shop tested value. This likely due to an inaccuracy in the temperature measurements, particularly the discharge temperature measurement. There were no thermowells in the discharge piping and the discharge temperatures were taken with a hand-held infrared pyrometer used to measure skin temperatures. This is really not a good tool to use when doing stage thermodynamic calculations. The outside temperature of the scroll and discharge piping varies quite a bit and are not well representative of the gas temperature. A difference of 10°F between the measured skin temperature and gas temperature results in a 9% difference in the calculated isentropic efficiency for the second stage. However, the same technique was used before the repair and after the repair, and for isentropic calculations, the discharge temperatures does not affect the calculated head. As shown, the calculated head after the repair matched fairly well with the shop testing head. And, after the repair, and the overall machine performance was restored as shown in Figure 28.

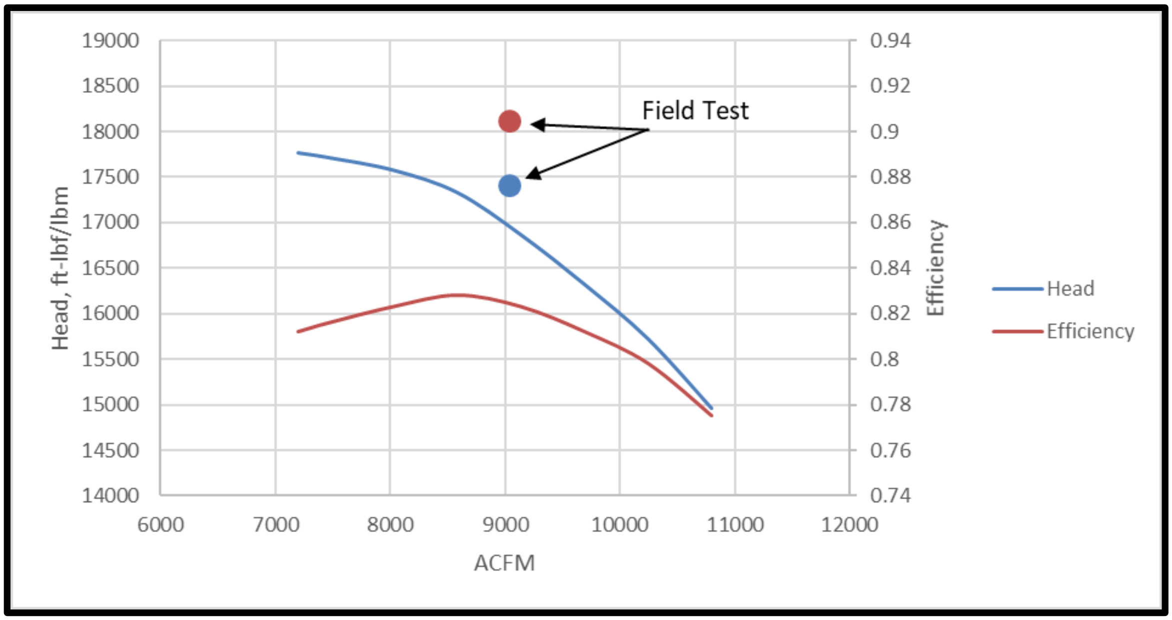


Figure 27: Case Study Two – Second Stage Performance After Repair

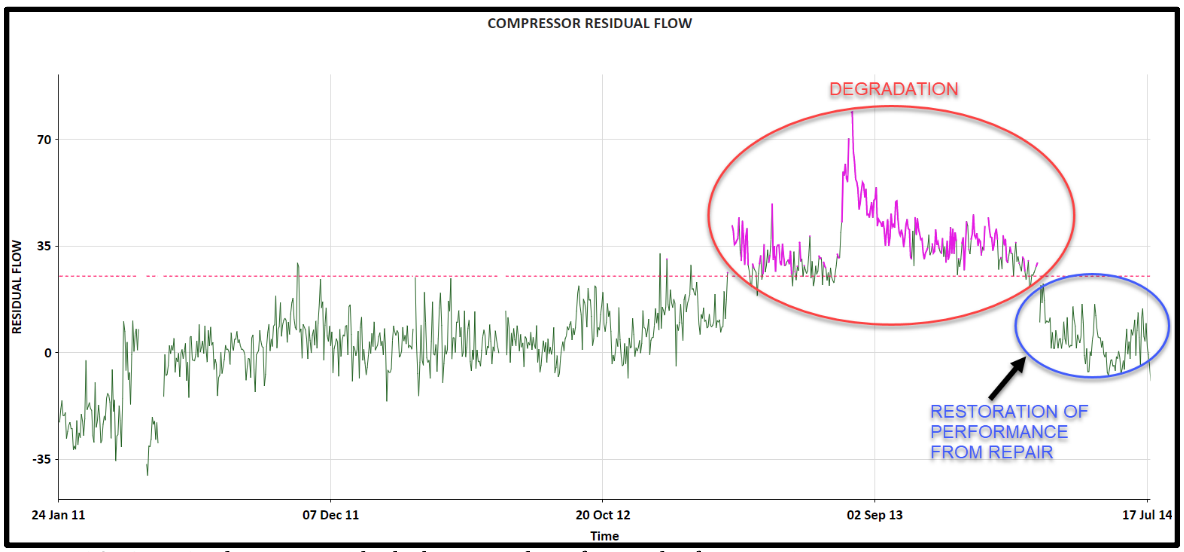


Figure 28: Case Study Two Residual Flow Trends Before and After Repair

CASE STUDY TWO LESSONS LEARNED

In this case, a compressor performance problem was identified in the early stages of degradation. Through stage testing, the problem was confirmed, and the problem area was identified making the physical inspection and corrective action much easier.

Had the type of degradation described in this tutorial gone un-noticed, it is possible that the compressor could have been driven into unstable operating conditions, possibly leading to a failure.

As discussed in the introduction and in the examples discussed, early identification of machinery degradation through a robust condition monitoring program is a key factor in a successful reliability program.

**CASE STUDY THREE**

This case study pertains to a four stage, dual service integrally geared centrifugal compressor driven by a 1200 RPM, 22,000 HP synchronous motor. The gearbox consists of a bullgear and two rotors. The low speed rotor comprises the first two stages of the main air compressor (MAC) service. The high speed rotor comprises the third stage of the MAC service and a single stage of booster air compressor (BAC) service.

In this design the front side of the impeller is open, and the vanes run against a close clearance, non-contacting stationary shroud. The compressor configuration is shown in Figure 29.

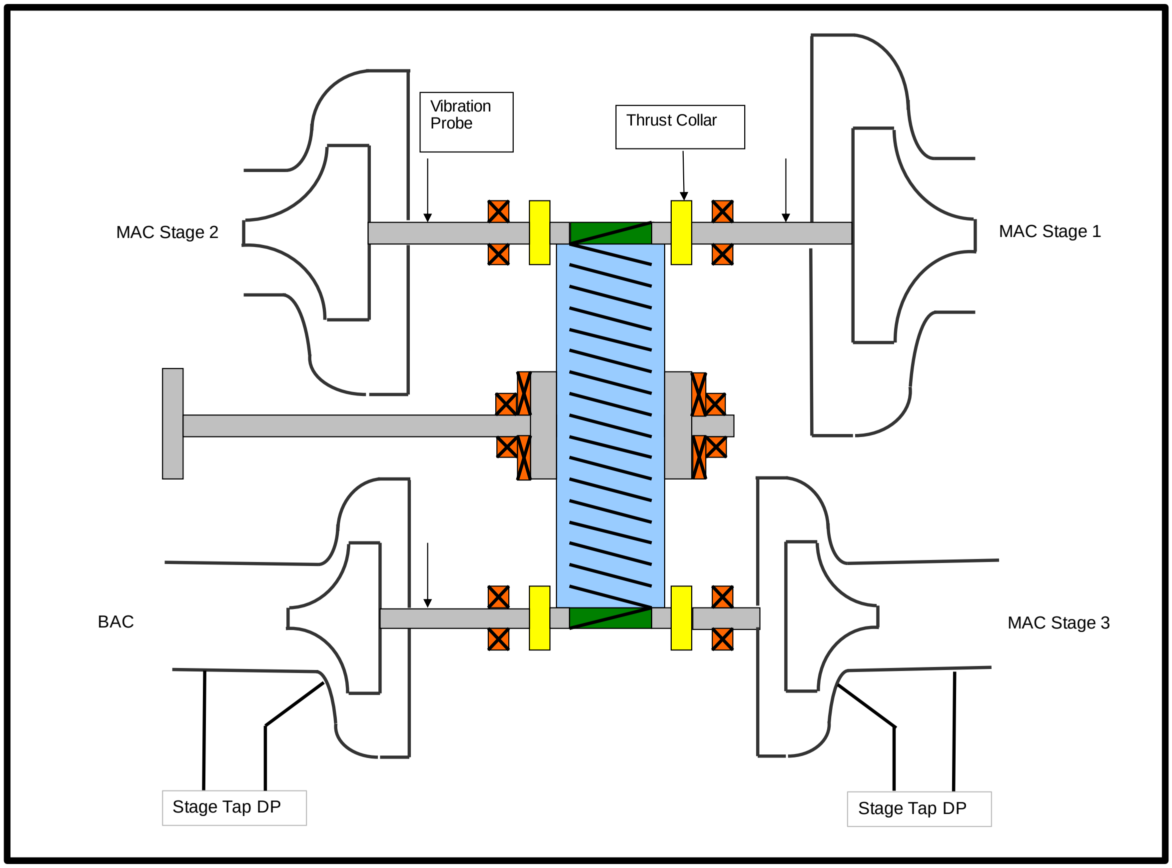


Figure 29: Case Study Three Compressor Configuration

The P&ID for the MAC section is shown in Figure 30. The primary control devices include a first stage IGV and a discharge vent valve. The primary controls include a third stage differential pressure indicating controller (DPIC), a discharge pressure indicating controller (PIC), and a flow indicating controller (FIC). The controllers are part of the DCS (Digital Control System) which is used to control the IGV and the discharge vent valve. A variable IGV was also installed on the third stage, but this physically locked in the 100% open position due to earlier control problems with the IGV.

In this application, the compressor capacity is controlled with the IGV based on a signal from the flow indicating controller. This flow is the measured flow downstream of the air clean-up system. The machine discharge pressure is a function of the system back pressure.

The third stage differential pressure measurement is the difference between the pressure at a point along the contour of the impeller and the third stage inlet pressure. See Figure 29. This differential pressure measurement is sometimes referred to as a stage tap. The square root of this differential pressure is proportional to flow. Although this is not a calibrated flow device, it provides a repeatable signal that can be used for surge control.

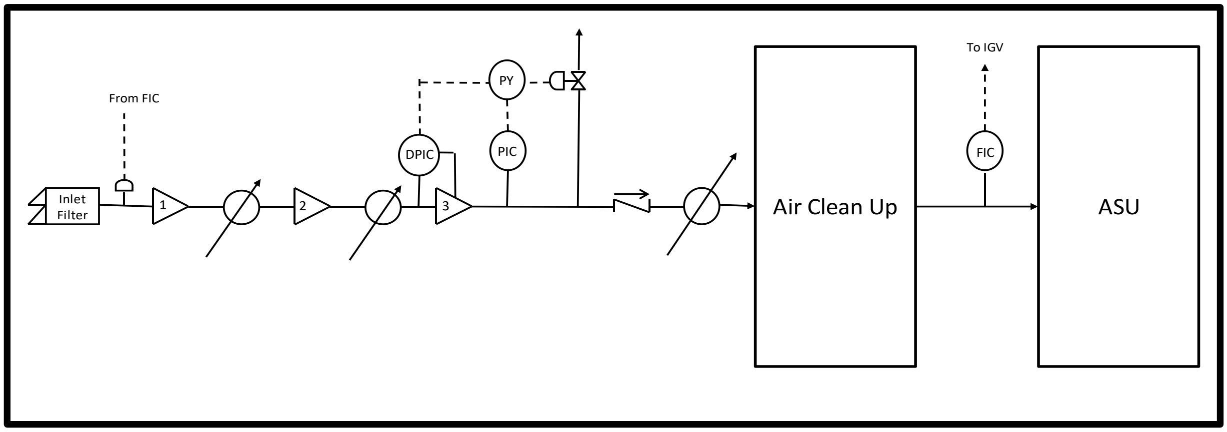


Figure 30: Case Study 3 P&ID for MAC Service

This compressor had operated for several years without any performance issues. Then, one summer, the plant operators observed a significant drop in MAC capacity as the summer ambient temperatures warmed up. At the time the problem was identified, an automated equipment condition monitoring system had not been implemented. Condition monitoring was managed locally by the operators. A review of the overall compressor performance curves showed that the machine was between 10% and 15% short in capacity. The IGVs, vent valve, instruments, etc. were all checked. And the overall plant production was consistent with the same reduction in MAC capacity. So, it appeared the MAC performance degradation was a result of some fault in the machine. An overall picture of the compressor is shown in Figure 31.

Figure 31: Case Study Three Compressor

This compressor was not shop performance tested, but the supplier did provide predicted stage curves based on a polytropic process. As with the other case studies, it was not possible to take the compressors offline to take data over a complete range of flows. Although the supplier did provide stage data multiple IGV angles, the field test data was taken at a single operating point with the IGV's 100% open. The results of the field test results data for the MAC service compared with the shop test data for each stage are shown in Figures 32 to 34.

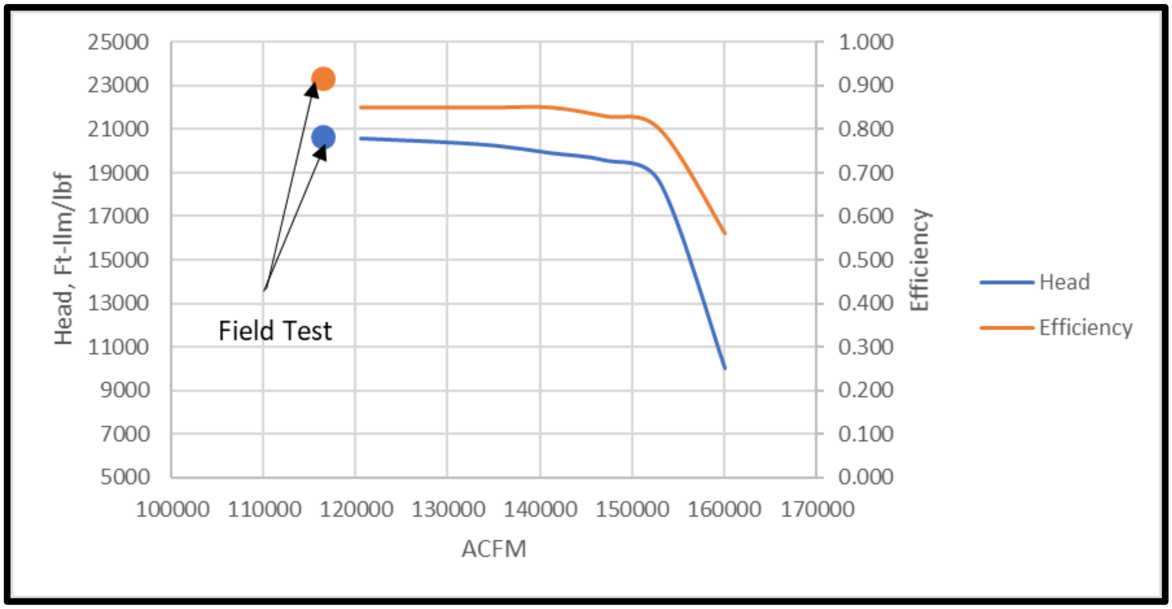

Figure 32: Case Study Three -First Stage Performance

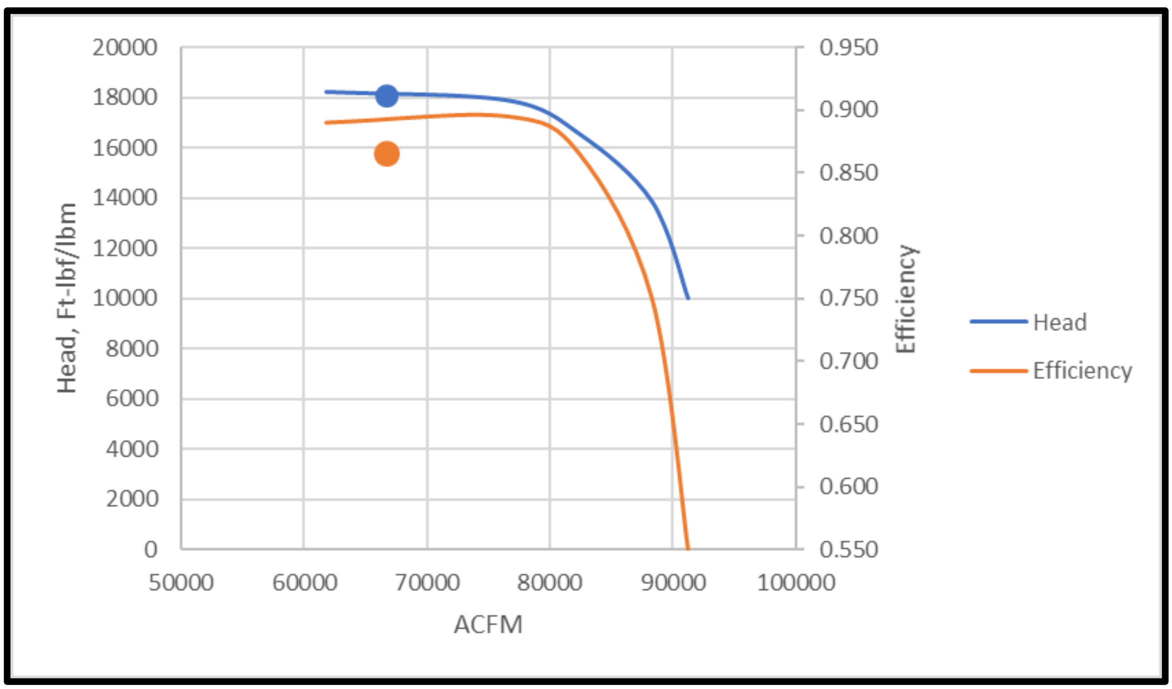

Figure 33: Case Study Three - Second Stage Performance

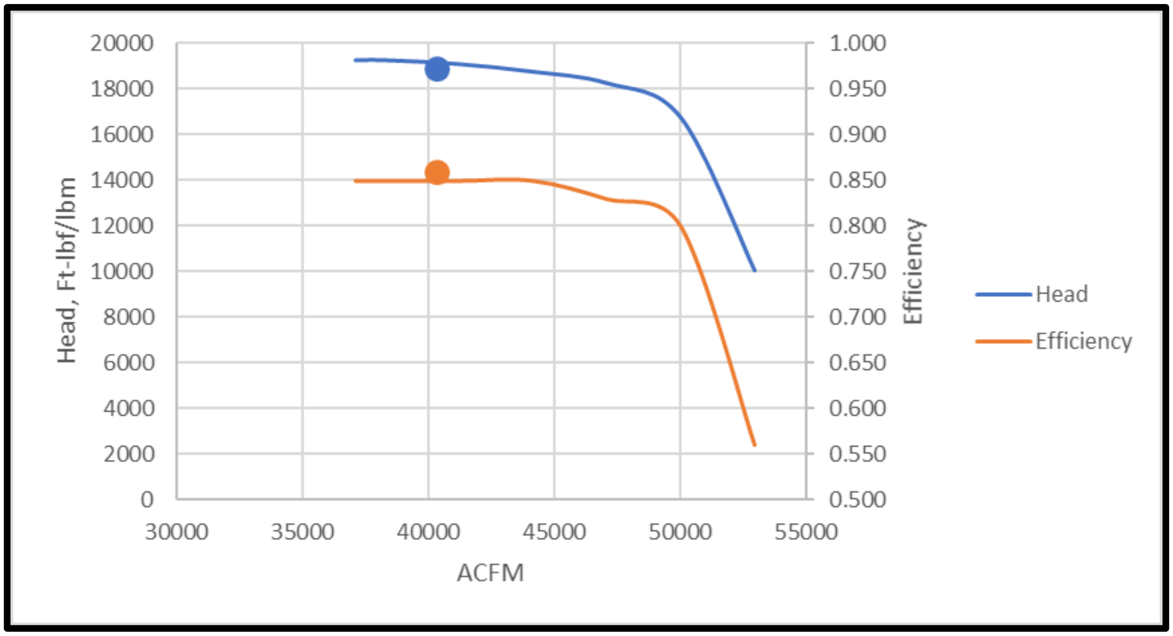

Figure 34: Case Study Three – Third Stage Performance

As with Case Study Two, there were no thermowells in the discharge piping and the discharge temperatures were taken with a hand-held infrared pyrometer used to measure skin temperatures. As discussed previously, inaccuracies in the discharge temperature can have a significant impact on the calculated efficiency. A 10°F increase in discharge temperature for the first stage in this case study results in a seven percent reduction in polytropic efficiency. In a polytropic process, the discharge temperature also affects the calculated head. But, a 10°F increase in discharge temperature for the first stage in this case study results in an increase of 0.75% in the calculated polytropic head. Thus, inaccuracies in the discharge temperature measurements can explain some of the differences in field versus predicted polytropic efficiencies, but only has a small impact in the head.

As shown, the first stage field test point is to the left of the minimum flow point based on the predicted curves. It appears as there is some additional turndown beyond the predicted curve. The head matches the expected head if this curve is extrapolated and the head for the other stages matches well with the predicted heads. There is some plus/minus discrepancies between the field tested stage efficiencies and predicted efficiencies, but this affects power, not the overall head/capacity performance. Based on the field test, there did not appear to be a problem in the compressor that would explain the shortfall in machine capacity.

Looking further at the data, there did appear to be a problem with the intercooler performance. As shown in Table 7, the intercooler approach temperatures were close to design, but the interstage pressure drops were higher.

| **INTERCOOLER** | **1** | **2** |
|---|---|---|
| Design DP, psi | 0.7 | 1.2 |
| Actual DP, psi | 2.05 | 3.1 |
| | | |
| Design Approach, °F | 14.4 | 14.4 |
| Actual Approach, °F | 15.9 | 13.7 |

Table 7: Case Study Three – Intercooler Performance

Although the interstage pressure drops don't appear to be that high, it does have a significant impact on the head/capacity performance as shown in Figure 35. And, as shown, the field tested data point matches fairly well with the predicted performance with the degraded intercooler performance.

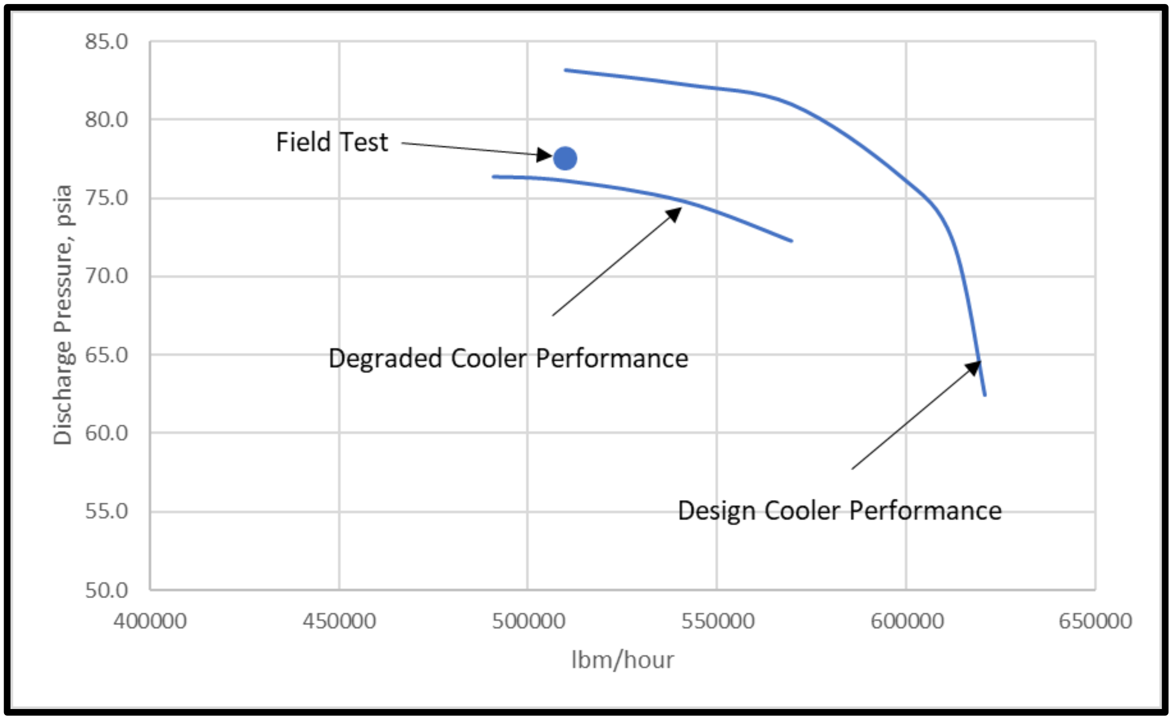


Figure 35: Case Study 3 – Predicted Performance with Design and Degraded Intercoolers

Based on this information, it appeared that there was a problem with the intercoolers. To determine the cause, an inspection was performed. As shown in Figure 36, the coating on the inside of the cooler shell was coming off and there was a considerable amount of rust that was flaking off.

Figure 36: Case Study Three As Found Intercooler Condition

The debris from the coating and rust plugged the intercooler fins caused the increase in interstage pressure drops. This tightly packed fins acted as a filter. See Figure 37. An attempt to clean the fins was unsuccessful. The intercooler bundles were eventually replaced, and the compressor performance was restored.

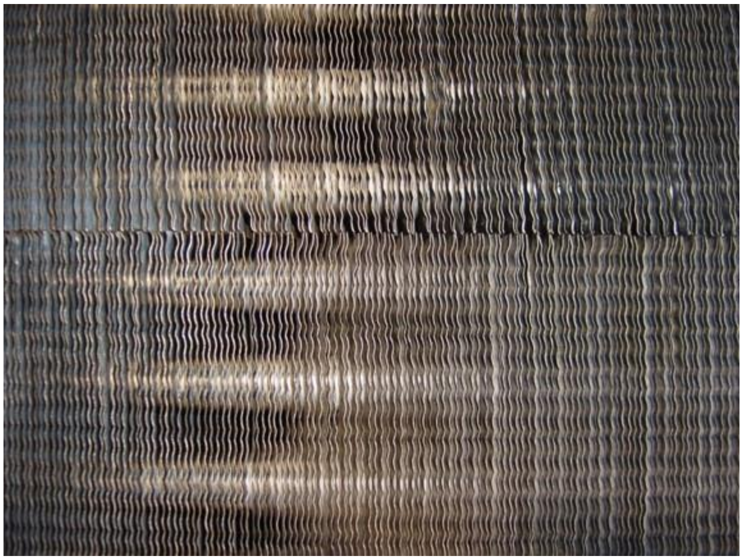
Figure 37: Case Study Three Intercooler Fins

To address the corrosion issue, atmospheric tested revealed moderate amounts of Sulfur in the air. This plant is located in a heavy industrialized area. When the cooler bundles were replaced, the intercooler shells and interstage piping were sand blasted and re-coated with a coating designed to resist Sulfur compounds. In addition, the MAC inlet filter was modified to incorporate HEPA elements to eliminate particulates from the atmosphere from fouling the intercoolers. These actions prevented future intercooler degradation.

CASE STUDY THREE LESSONS LEARNED
As with Case Study Two, the performance problem was confirmed with stage testing and the problem area was identified making the physical inspection and corrective action much easier. At first glance, the interstage pressure drops do not appear to be that high. However, as shown, this had a significant impact on the overall compressor performance.

**REAL GAS COMPRESSORS**

Up to this point, the compressors that were analyzed could be analyzed using ideal gas laws. However, there are many compressor applications where the gases deviate significantly from an ideal gas and alternate calculation procedures are needed. The process of analyzing stage performance is the same. Overall and interstage data is taken so that stage volumetric flow, head and efficiency can be determined and compared with supplier shop test or predicted invariant stage curves. The only difference is that the equations used to determine the stage performance are different. The thermophysical properties of the gas have to be calculated at the measured conditions and this can introduce some additional inaccuracy. As described by Sandberg (2017), "…it is assumed that the thermophysical properties derived from these measurements are equally as accurate. Unfortunately, this is not always the case." As described by Sanders (2017), what is this is coming about with more severe compressor involving discharge pressures and more complex compositions of the handled gases. This tutorial will not address any inaccuracies in the derived thermophysical properties, but interested readers can consult the paper by Sandberg (2017) for more information.

Case Study Four will be used to illustrate an example of a performance problem with a compressor in a real gas application.

**CASE STUDY FOUR**

This case study pertains to an eight stage integrally geared centrifugal compressor driven by an 1800 RPM, 17,500 HP synchronous motor. The gearbox consists of a bullgear and four rotors. The pinion and stage locations are shown in Figure 38. The first stage impeller is a semi-open type and all other impellers are closed types, meaning that there are shrouds on both the back and front walls around the vanes that rotate with the assembly.

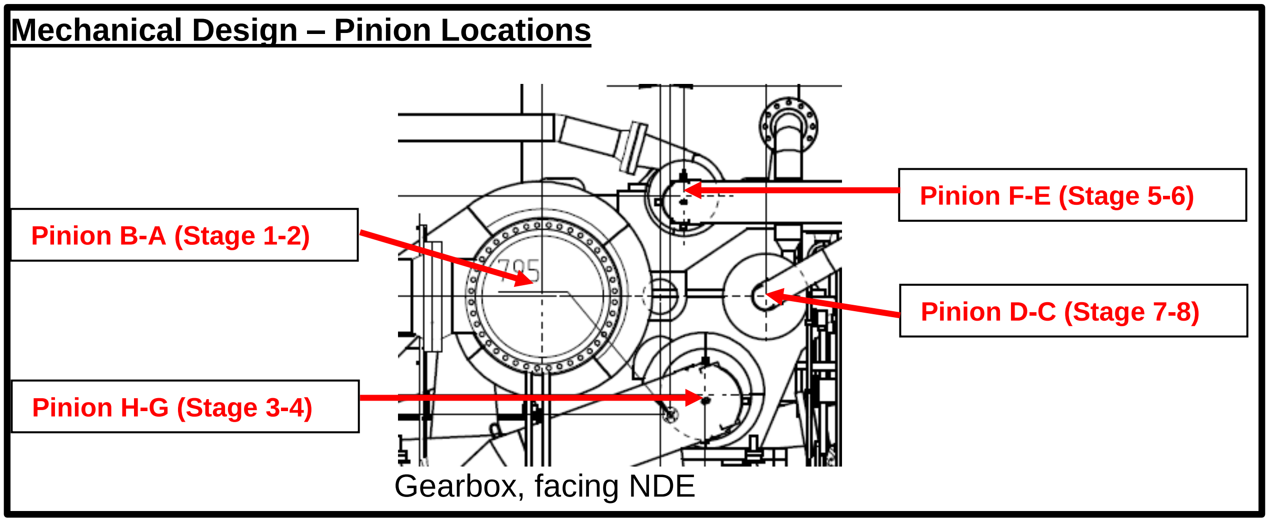


Figure 38: Case Study Four Pinion and Stage Locations

This compressor compresses saturated CO2 from 17.6 psia to 664.7 psia in stages one to five. The CO2 is then sent to a TEG contactor where it is dried and then is compressed to 2284.7 psia in stages six to eight. A simplified P&ID is shown in Figure 39. The P&ID does not show all the instruments, but enough is shown to understand the basic process control. There are two sections and each section is independently controlled with an IGV on the first stage of each section and a recycle valve.

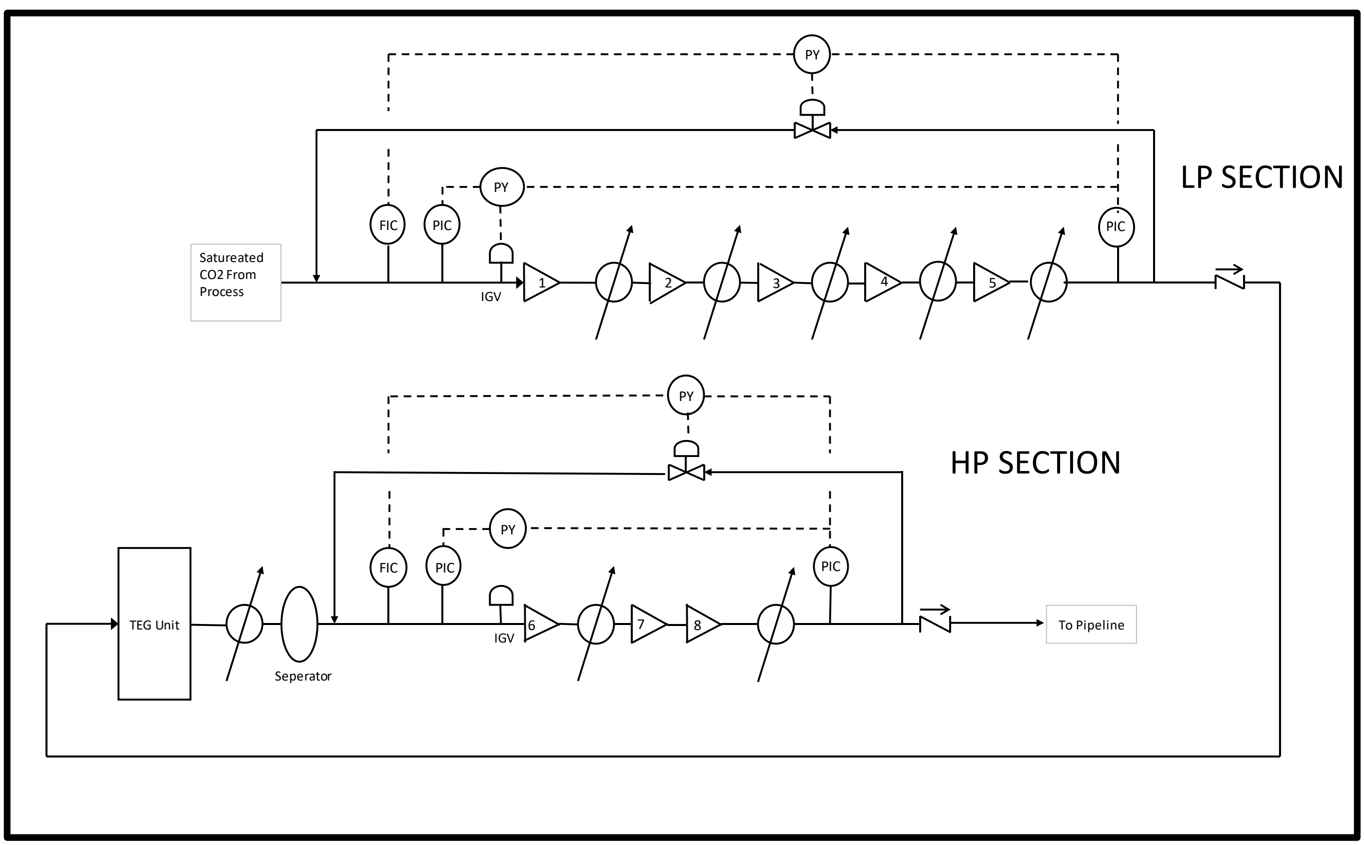


Figure 39: Case Study Four Simplified P&ID

The compressor was in operation for a few years before a performance problem was identiled with the high pressure (HP) section. The residual flow indicated a loss in capacity. See Figure 40. The plant operators also noticed that the HP section IGVs were 100% open whereas the position would vary during the cooler months as compressor capacity was greater than plant production.

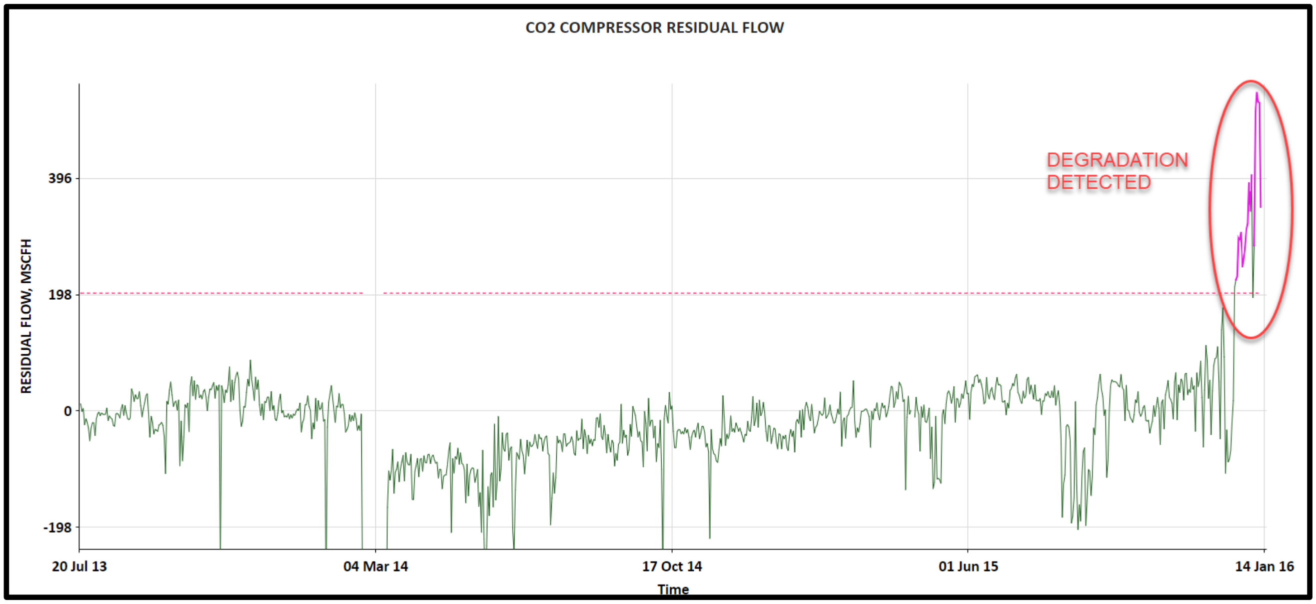


Figure 40: Case Study Four Residual Flow Trends

There were no interstage pressure transmitters in the HP section and the machine could not be shut down because of customer demand. So, there was no opportunity to do any stage by stage testing. This machine is fed from two different CO2 plants and the problem seemed to coincide after an outage at one of the plants. This was a four week outage and the recycle valves were open because of the lower feed rates. So, it was speculated that the continuous recycle may have eroded or damaged the recycle valve seat and/or assembly during this period. An outage opportunity became available and the recycle valve was removed and inspected, which revealed the valve plug was not on the seat; it was sitting about 0.125 inches off the seat in the full closed position. The valve was rebuilt and the compressor performance improved, but was still not right and it gradually degraded further.

Based on this an internal inspection was performed and severe fouling was found in stage six as shown in Figure 41.

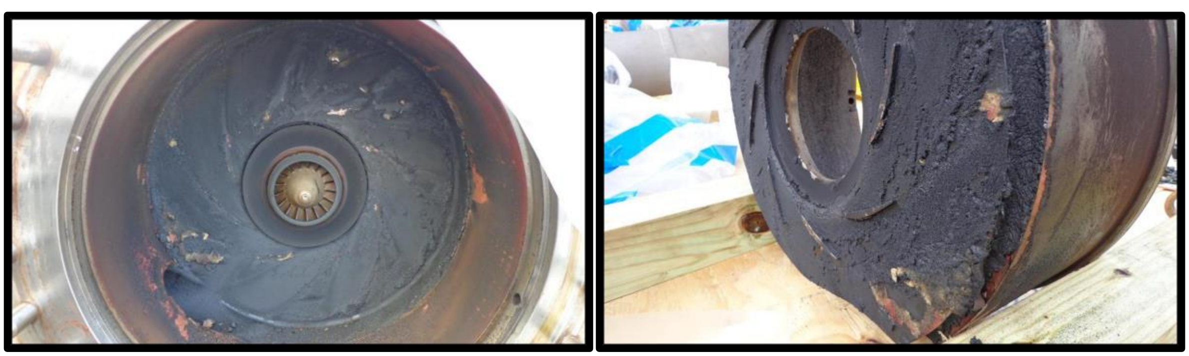

Figure 41: Case Study Four Stage Six Fouling

The fouling was due to TEG from the upstream drier system. After a lengthy investigation, a process issue was discovered that caused the TEG to oxidize and form a polyester polymer that fouled the sixth stage.

Had a stage by stage performance analysis been done in the field, it is likely the problem with stage six would have been discovered sooner and/or led to a more targeted inspection and repair that would have been quicker and less intrusive. However, these would have needed to be performed based on real gas equations. Performing stage by stage calculations using ideal gas equations would lead to erroneous results. Let’s look at an example based on stage six design. The design inlet and outlet pressures and temperatures, flowrate and gas composition is shown in the table 8.

| DESIGN POINT - STAGE SIX | |
|---|---|
| Inlet Pressure, psia | 655.4 |
| Outlet Pressire, psia | 1222.5 |
| Inlet Temperature, F | 102.8 |
| Outlet Temperature F | 205.7 |
| | |
| Mass Flow, lbm/hour | 248459 |
| | |
| DRY GAS COMPOSITION | |
| CO2 | 97.0% |
| CO | 0.7% |
| H2 | 0.7% |
| CH4 | 1.6% |
| | |
| Relative Humidty | 1.70% |

Table 8: Case study Four Stage Six Design Point

Polytropic calculations based on ideal gas equations and real gas equations are compared and shown in Table 9 and also on the predicted stage curve shown in Figure 42. Equations (26) to (28) are used for the real gas calculations. Some of the gas properties were calculated using a Computer-Aided Physicochemical Property (CAPP) System that was developed and is maintained by Air Products. The Schultz correction factors discussed in ASME PTC-10 were not used. When performing real gas calculations, several isentropic calculations have to be performed. This is represented by the ‘ in the formula. CAPP was also used for these calculations.

| IDEAL GAS CALCUATIONS | |
|---|---|
| ACFM | 867 |
| Polytropic Head, ft-lbf/lbm | 13422 |
| Polytropic Efficiency | 0.508 |
| | |
| **REAL GAS CALCULATIONS** | |
| ACFM | 692 |
| Polytropic Head, ft-lbf/lbm | 10745 |
| Polytropic Efficiency | 0.798 |

Table 9: Case Study Four ideal gas Versus Real Gas Calculations for Stage 6

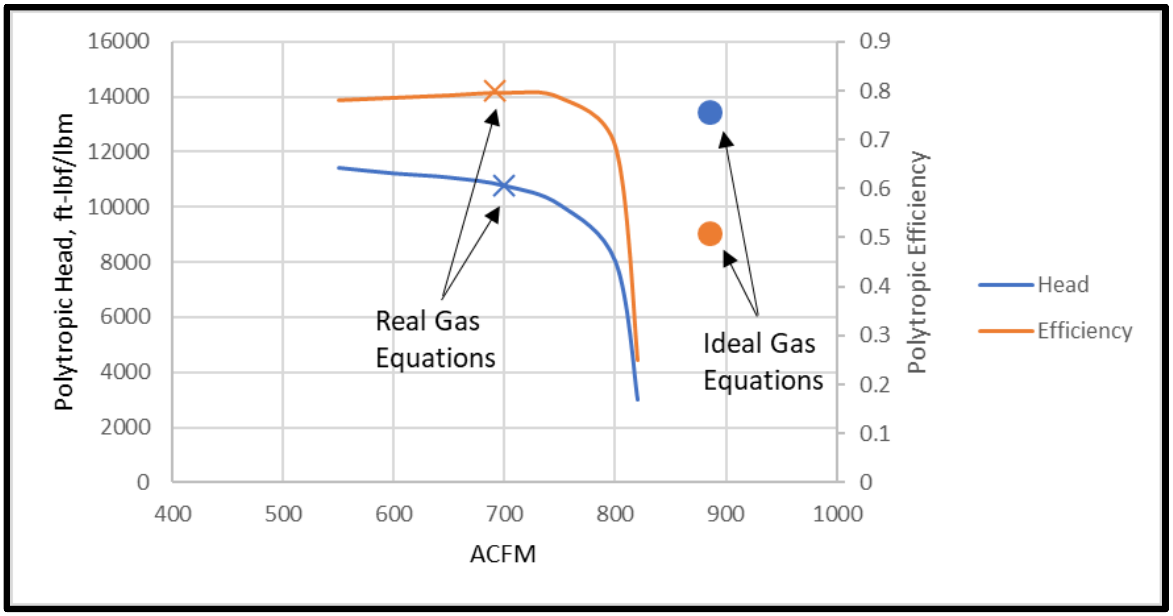


Figure 42: Case Study Four, Stage 6 Ideal Gas Versus Real Gas Comparison

As shown, using ideal gas equations for a real gas gives erroneous results. ASME PTC-10 provides guidance on when ideal gas equations can be used and when real gas equations have to be used. Although this is in regard to shop performance testing, it equally applies to field testing.

What happens if real gas equations are used for an application that can be analyzed as an ideal gas. Calculations were done for Case Study Three Stage One, and as shown in Table 10, the results are effectively the same, even though the calculation methods are quite different.

| IDEAL GAS CALCUATIONS | |
|---|---|
| ACFM | 114894 |
| Polytropic Head, ft-lbf/lbm | 20538 |
| Polytropic Efficiency | 0.919 |
| | |
| **REAL GAS CALCULATIONS** | |
| ACFM | 114850 |
| Polytropic Head, ft-lbf/lbm | 20540 |
| Polytropic Efficiency | 0.922 |

Table 10: Case Study Three Stage One; Ideal Gas Versus Real Gas Results

LESSONS LEARNED – CASE STUDY FOUR

The same techniques for modelling and evaluating overall compressor performance for ideal gas compressors can be applied to real gas compressors. As shown in this case study, the ECM tool was able to identify degradation in machine performance, even at a relatively early stage.

When evaluating stage performance in a compressor in a real gas application, using ideal gas equations and derived thermophysical properties will lead to erroneous results. To evaluate stage performance, it is necessary to use real gas equations and the thermophysical properties. Although this was not illustrated in this example, had stage by stage testing been done, it is very likely the problem with the sixth stage would have been identified and would have led to a less intrusive, less costly and faster inspection and repair.

## CONCLUSIONS

As discussed by Verderame and Metha, "With the advent of the IIot (Industrial Internet of Things), manufacturing companies now are amassing large datasets. Sensors on all manner of equipment such as compressors, pumps, motors, expansion turbines, blowers, cooling towers and heat exchangers are streaming data to process historians at high rates. Advanced data analytics enable the transformation of these enormous datasets into reduced dimensional, composite variables that take into account the inherent characteristics of the current operating condition, capturing the heartbeat of the process." As shown in the case studies, the use of this concept has can be used to detect even the early stages of centrifugal compressor thermodynamic performance degradation.

As presented in this tutorial stage by stage performance testing is a technique that can be used to identify the degraded area(s) of the machine. To do this, either shop tested stage data or predicted data is needed to compare to be able to compare actual versus shop tested versus shop tested or predicted performance. As shown in the case studies, this technique was used to successfully identify the area of the machine that had degraded. At this point, inspections are generally necessary to determine the cause of the degradation. Identifying the area results in a less intrusive inspection and repair, saving time and money.

As discussed in this tutorial, care must be taken when deciding to analyze stage data based on ideal gas equations or real gas equations. As shown, treating something that behaves as a real gas as an ideal gas can lead to erroneous results and conclusions.

## BASIC SYMOLS AND UNITS

*Variables*

| | |
|---|---|
| f | = Polytropic work factor (dimensionless) |
| g | = Gravitational constant (32.174 lbm ft/lbf . sec$^2$) |
| h | = Enthalpy (BTU/lbm) |
| H | = Head (BTU/lbm) |
| J | = Mechanical equivalent of heat (778.17 ft Ibf/Btu) |
| k | = Ratio of specific heats, cp/cv (dimensionless) |
| m | = Mass (lbm) |
| MW | = Molecular Weight |
| n | = Polytropic exponent for a path on the p-v diagram |
| N | = Speed (RPM) |
| P | = Pressure (psia) |
| $r_p$ | = Pressure ratio (dimensionless) |
| Q | = Volume flow (ft$^3$ / minute) |
| $Q_h$ | = Heat losses, BTU/min |
| T | = Temperature (°R) |
| R | = Gas Constant (ft * lbf / lbm* °R) |
| U | = Blade Tip Speed (feet/second) |
| V | = Fluid velocity (feet/second) |
| v | = Specific volume (ft$^3$ / lbm) |
| w | = Mass flow (lbm / minute) |
| W | = Work per unit mass (BTU/min) |
| z | = Elevation |
| Z | = Compressibility factor |
| | |
| φ | = Flow coefficient (dimensionless) |
| η | = Efficiency |
| ρ | = Density (lbm / ft$^3$) |
| µ | = Head coefficient (dimensionless) |

Subscripts

| | |
|---|---|
| i | = Inlet |
| iso | = Isothermal |
| d | = Discharge |
| p | = Polytropic |

s = Isentropic

Superscripts

' = Result based on an isentropic process

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