47TH TURBOMACHINERY & 34TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 17-20, 2018
GEORGE R. BROWN CONVENTION CENTER

Determination of Operating Conditions and the Impact on Integrally Geared Centrifugal Air Compressor Selection and Performance

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ABSTRACT

Centrifugal compressors are utilized to supply compressed air for a variety of applications in diverse industries ranging from food and beverage processing to petrochemical plants and oil refineries. These machines must cover a wide range of air flow rates and discharge pressures, leading to various levels of power consumption. When considering a compressed air solution, end users and purchasers determine their air flow and pressure requirements as well as their inlet conditions. These conditions, including ambient pressure, ambient temperature, and relative humidity typically depend on the geographic location where the machine will be installed,

as well as characteristics of the installation site itself, such as cooling water capacity. Given that most locations experience a range of ambient conditions over the course of a year and because users might have more than one pressure or flow requirement, there are typically several operating points considered in the selection process. Once these are compiled, they are supplied to compressor manufacturers to determine their best product offering based on the information provided.

This tutorial provides insight into how these provided conditions effect the selection of the compressor aerodynamic hardware and how that selected hardware impacts the compressor's performance in terms of the desired flow, desired pressure, and the corresponding power consumption. The end user desires to have a compressor capable of reaching all of their operating conditions without the need to "blow off"; that is, to discharge excess air flow to the atmosphere and essentially waste energy, while also maintaining the lowest power consumption and cost possible. While it is not always feasible to meet all operating points, a user can greatly improve their ability to acquire a machine that meets what they desire by providing operating and inlet conditions that closely pertain to their needs and are realistic for the region of installation.

This paper will provide several examples of how a user's specified conditions impact the selection of the compressor along with discussions of how providing well-realized conditions can help reduce the compressor's power consumption or enable a user to avoid blowing off excess air during typical operation. The paper will also discuss the flexibility and limitations centrifugal compressor manufacturers have in meeting various operating conditions through the selection of aerodynamic hardware and the inlet control valve including variable inlet guide vanes. Finally, the paper will provide input from an end user of centrifugal air compressors offering their point of view of the conditions provided and the compressed air solutions they need. The overarching goal is to better educate end users and purchasers of centrifugal air compressors so they can make more informed requests and obtain machinery that more efficiently meets their year-round needs.

INTRODUCTION

In general, centrifugal compressors can be used to describe a large variety of machines, which can be used to compress various fluids. The focus of this paper will be on integrally geared, fixed speed atmospheric air compressors but many of the concepts apply to the wider compressor audience. In an integrally geared air compressor a stage typically consists of an inlet casing, an impeller which can be unshrouded/open type or covered, a diffuser which can be vaned or vane less, and a scroll. Heat exchangers are used between each stage for cooling the air, as air is susceptible to large changes in temperature (and therefore density/specific volume) when being compressed. The air flow intake and discharge pressure are typically controlled with an inlet guide vane (IGV) and unloading control valve (UCV). Figure 1 gives an example compressor layout with the typical air flow path shown. Air from atmosphere is drawn into the compressor typically through an inlet filter (not shown), then passes through the IGV and into the first stage where it is compressed as it moves through the impeller and diffuser. The air is then collected by the scroll and discharged from the stage. At this point the hot air moves through the first intercooler where the air is cooled and then moved on to the following stage where the process is repeated.

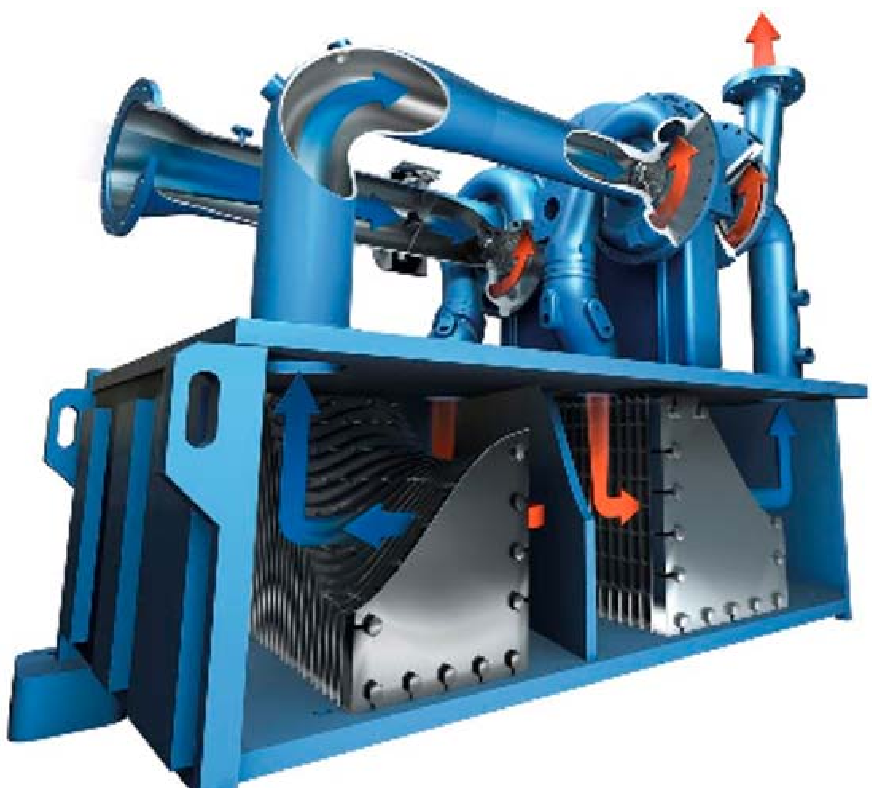

Figure 1. Integrally Geared Air Compressor Cutout

Compressor performance is generally characterized using performance curves. These curves plot compressor discharge pressure versus the flow rate to show the operating characteristics of the compressor and is typically referred to as the operating map. Figure 2 shows an example of one such plot. The operating curves for fixed speed compressors are generated using an inlet control valve, in this case an IGV. Each curve in Figure 2 corresponds to a distinct IGV angle. At a set IGV angle the curve is created by using the UCV to build back pressure; as the pressure increases the flow intake to the machine decreases. The surge point is the lowest possible flow the compressor can intake before flow reversal starts to occur at a set IGV angle and the machine becomes unstable. The choke point is the maximum flow the compressor can intake before choking occurs and the flow will no longer increase for further downstream decreases in pressure at a set IGV angle. Pressure rise to surge is typically given as a percentage and is defined by the ratio in pressure from the design point to surge. The turndown point is the theoretical absolute minimum flow achievable by the machine at a set discharge pressure, and turndown in the compressor refers to decreasing the flow typically at a constant discharge pressure. At the turndown point, the compressor will be in a constant state of surge; therefore this is not a true operating point. A more typical operating range is shown on the figure, where at the minimum actual operating point the compressor still maintains an acceptable pressure rise to surge. The actual operating range being less than the surge line results in the overall control line being parallel but offset from the surge line by some amount of flow. Another common term used in reference to centrifugal air compressors which is not shown on Figure 2 is blow off. Blow off refers to a condition when the required flow is below the turndown point of the compressor. Since the compressor is delivering more flow than needed, the excess air is simply blown off to atmosphere instead of falling into surge.

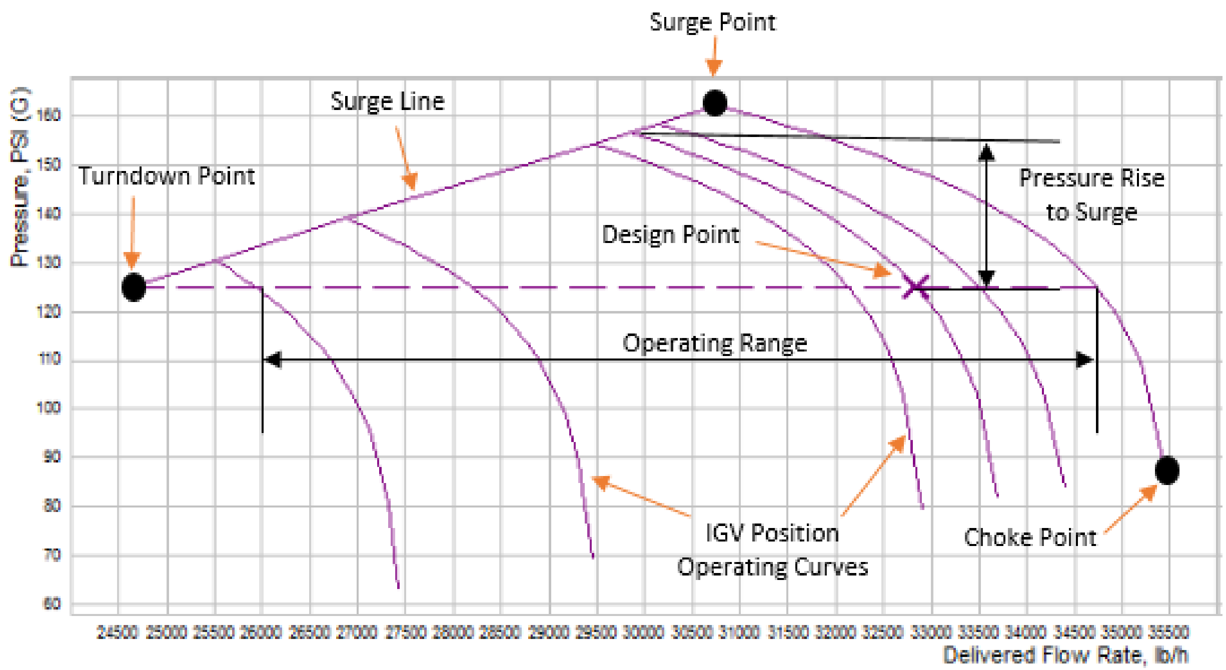

Figure 2. Compressor Operating Map

In addition to the pressure versus flow rate plot, curves are also commonly created that plot the shaft power required to run the compressor versus the flow rate. Figure 3 shows an example of this plot. Similar to Figure 2, each curve on Figure 3 coincides with a set IGV angle. Figure 2 and Figure 3 have the same design point and each of the curves from max capacity to min capacity are for the same IGV set angle on both plots. When viewed together they give an overall picture of the compressor performance over the range of its operating map. It is important to note that these plots are for a single set of site conditions. When the site conditions change, (ambient air temperature for example) the nature of the performance curves will change. This tutorial will discuss how through stage selection, these curves are generated and how the site conditions effect that process and the curves themselves. The tutorial will also discuss how the compressor is able to account for changes in these ambient conditions during operation.

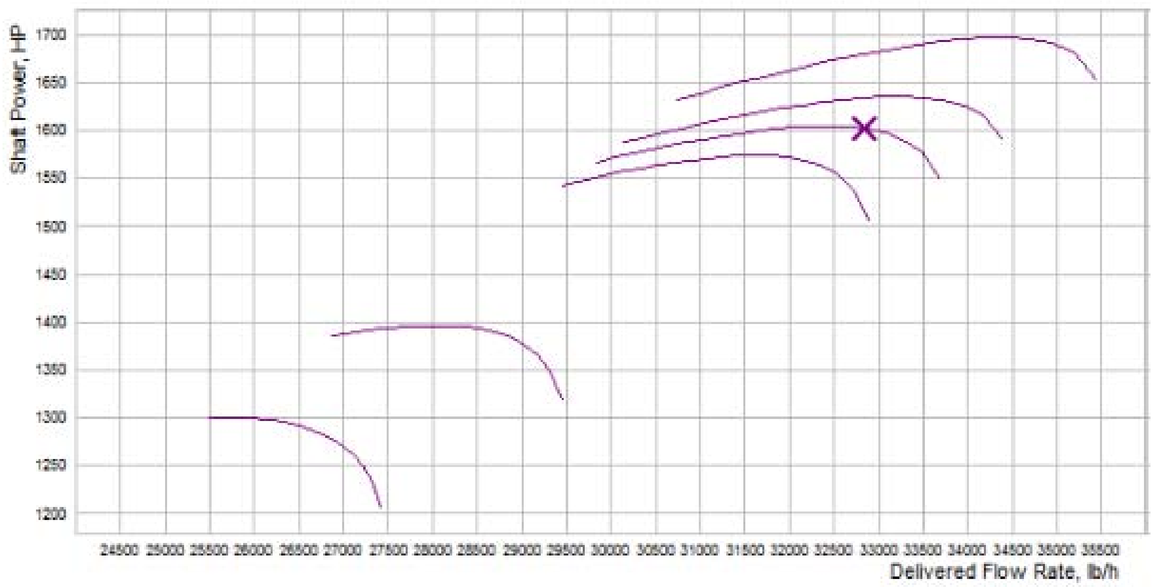

Figure 3. Compressor Power Operating Curves

COMPRESSOR SELECTION PROCESS

In order to perform a compressor selection for a typical application there are a number of parameters about the installation site that need to be known along with the flow and pressure requirements. These parameters include the ambient/inlet pressure, temperature, and relative humidity as well as the motor frequency. For typical machines pulling air from atmosphere, the temperature and relative humidity at the inlet and the ambient condition are considered to be the same. The inlet pressure is the ambient pressure subtracted by the pressure drop through the inlet air filter. The pinion speed is simply the motor speed, (dependent on the motor frequency) multiplied by the gear ratio of the drive gear or bull gear to the pinion. With this information and an inlet volume flow specified, the basic compressor performance can be calculated.

As mentioned, Figures 2 and 3 show the overall compressor performance. A typical integrally geared air compressor consists of multiple stages with cooling in between each stage. The overall performance map is created by looking at the performance of each individual stage. In order to do this the first thing calculated is the saturated vapor pressure which is accomplished using the following equation.

$$P_s = 0.0886 * 10^{\frac{7.5*T_{in}-240/1.8}{237.3+(T_{in}-32)/_{1.8}}} \quad (1)$$

Where T_{in} is the air temperature into the stage in ˚F and P_s is the saturated vapor pressure in PSIA. Using saturation pressure along with the inlet pressure and relative humidity, the specific humidity can be calculated.

$$SH = 0.622 * \frac{RH*P_s}{P_{in}-(RH*P_s)} \quad (2)$$

Equation 2 is valid for the first stage of the compressor, where humid air is being drawing from atmosphere. Since relative humidity is an ambient condition, for subsequent stages the factor of RH is removed from equation 2 and the specific humidity depends solely on the inlet pressure and saturation pressure of the stage. With the specific humidity known the gas constant of the air/vapor mixture (humid air) is calculated.

$$R_m = \frac{R_v*SH+R_a}{1+SH} \quad (3)$$

With the gas constant of the humid air calculated, the specific volume is calculated. This is done using the ideal gas equation as shown in equation 4.

$$v_{in} = \frac{R_m*(T_{in}+459.67)}{P_{in}*144} \quad (4)$$

In equation 4 the addition of 459.67 is to convert T_{in} to units of ˚R and the factor of 144 is to convert P_{in} to units of psf in order to give v in units of ft^3/lb_m. Next, the flow is calculated and the procedure varies depending on the stage. In this simple example where the inlet volume flow is specified the mass flow is calculated using equation 5. For subsequent stages the mass flow into the stage is reduced by the seal leakage loss of the previous stage and the water knockout through the interstage cooler. The seal leakage losses can vary depending on the size of the compressor and type of seal used (carbon ring, labyrinth, etc.) but is typically small, i.e. less than 1-2 lb_m/min. Next, the volume flow into the stage is calculated using equation 6, which is simply equation 5 rearranged to solve for Q.

$$\dot{m} = \frac{Q}{v_{in}} \tag{5}$$

$$Q = \dot{m} * v_{in} \tag{6}$$

Now that the flow is calculated, the stage non-dimensional performance coefficients can be determined. Since every installation region will have differing site conditions, it is convenient to express the stage performance as non-dimensional parameters that apply regardless of the atmospheric conditions. These coefficients are flow coefficient, head coefficient, work coefficient, and efficiency which are defined by equations 8, 9, 10, and 11 respectively. These coefficients can be defined in a number of ways. An adiabatic process is a reversible constant entropy process for an ideal gas without heat transfer in or out of the system. A polytropic process is a reversible process for an ideal gas with heat transfer and variable entropy, which follows the relationship given by equation 7 where n is the polytropic exponent.

$$Pv^n = constant \tag{7}$$

The coefficients can also be given with respect to an isothermal process which is a constant temperature process. The remaining discussion will be based on adiabatic coefficients.

$$\varphi = \frac{700}{N*D^3}Q \tag{8}$$

$$\psi = \frac{H_{ad}}{U^2/g} \tag{9}$$

$$\gamma = \frac{W}{U^2/g} \tag{10}$$

$$\eta = \frac{\psi}{\gamma} \tag{11}$$

From either empirical data from test(s) or expected data from a design program a polynomial curve fit can be created for the head coefficient and efficiency versus flow coefficient, and then work coefficient can be calculated by rearranging equation 11. In this way the coefficients can be determined for a known flow and therefore flow coefficient and can be used to calculate the stage discharge properties. An example of the three stage curves is shown in Figure 4.

Similar to the overall compressor curves, the low flow point is considered the individual stage surge point and the max flow point is considered the individual stage choke point. In very general terms, the head coefficient is dependent on the stage's impeller diameter and accompanying diffuser as well as the pinion speed. The flow through a stage is primarily determined by the impeller and diffuser blade heights, commonly referred to as stage profiles. As previously stated these polynomial curve fits can be used to determine ψ and η which in turn can be used to calculate the stage discharge properties. The adiabatic head can be calculated by rearranging equation 9 as shown in equation 13 below, which also requires the impeller tip speed be calculated as shown in equation 12.

$$U = \frac{\pi*D*N}{720} \tag{12}$$

$$H_{ad} = \frac{\psi*U^2}{g} \tag{13}$$

With the adiabatic head calculated, the discharge pressure of the stage can be calculated using equation 14.

$$P_{out} = P_{in} * \left(\frac{H_{ad}*(k-1)}{k*P_{in}*v_{in}} + 1\right)^{\frac{k}{k-1}} \tag{14}$$

Equation 14 depends on the value k, the specific heat ratio. Given that k is a function of temperature, it is generally safe to assume that the value of k is 1.4 for air compressors as the temperatures do not get high enough to significantly change this. With the discharge pressure calculated, the stage discharge temperature can be found using equation 15.

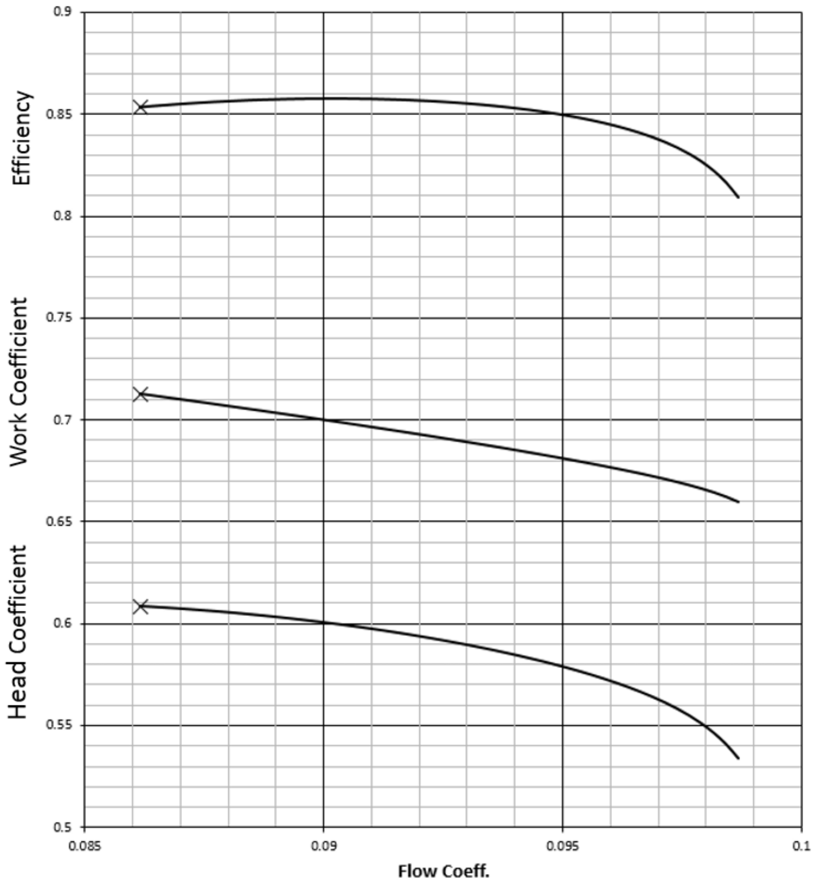

Figure 4. Example Stage Curves

$$T_{out} = \frac{T_{in} * \left[\left({}^{P_{out}}/_{P_{in}}\right)^{\frac{k-1}{k}} - 1\right]}{\eta} + T_{in} \tag{15}$$

In order for equation 15 to be valid T_{in} needs to be converted to ˚R by adding 459.67. This will then give the discharge temperature in ˚R as well, which can be simply converted to ˚F by subtracting 459.67. Finally, the stage gas power, which is a measure of the power input required for the stage to operate at a certain condition, can be calculated as shown in equation 16.

$$GHP = \frac{\dot{m} * H_{ad}}{\eta * 33{,}000} \tag{16}$$

This completes the process for a single stage calculation. When multiple stages are considered, as previously mentioned the inlet mass flow to a subsequent stage is the mass flow of the previous stage minus any water knockout that occurs in the cooler and any seal leakage loss that took place in the previous stage. The inlet pressure into a subsequent stage is the discharge pressure of the previous stage minus the interstage pressure loss of the air as it travels through the interstage piping and interstage cooler, which is typically determined based on empirical test data. The inlet temperature into a subsequent stage is the temperature out of the interstage intercooler. This can be determined by simply using a specified approach temperature, which is defined as the difference between the air discharge temperature and the coolant inlet temperature, or by doing more detailed calculations considering the heat exchanger design. With the inlet pressure, temperature, and flow into the next stage specified, the process outlined above can be repeated for each stage that follows. The mass flow delivered by the compressor will be the mass flow out of the final stage of the compressor, while similarly the compressor discharge pressure will be the discharge pressure out of the last stage. The compressor's total shaft power is the summation of each stage's gas power plus the compressor's mechanical power consumption.

While equations 1-16 were discussed as they relate to a single operating point it can be seen how they are used to create the overall compressor performance maps given previously in Figures 2 and 3. The characteristics of the overall compressor performance are determined by looking at the combined performance of each stage. How each stage performs together is typically referred to as stage matching, since in order to meet a specified overall performance given by the required flow and discharge pressure, the individual stage performances need to match. Given that Figures 2, 3, and 4 all depend on the compressor flow, it can be seen how when considering multiple stages, i.e. multiple figure 4's the overall compressor performance is dependent on the individual stages. For example, the surge point defined in Figure 2 will be determined by the first individual stage to surge in a multi stage machine and similarly the choke point will be determined by the first stage to choke; these stages are typically referred to as controlling surge and choke respectively. Furthermore, it can be seen how the pressure and power curves are generated, by varying the flow the combined stages will have varying pressure and power outputs which when viewed together create the curves in Figures 2 and 3.

When performing a compressor selection, typically several combinations of stages are considered, with each combination giving different power and turndown characteristics for a required flow and discharge pressure at specified site conditions. To further complicate things, typically multiple site conditions are specified in order to determine the compressor performance year round. Additionally, turndown can be specified, so at a single site condition, the compressor will need to be able to achieve multiple flow requirements. In addition to each of the cases to consider, there can be requirements on other parameters. A common request is for the surge margin of the compressor to be at a certain level to ensure that changes to the downstream system will not cause the compressor to surge. Another consideration is on the discharge air temperature of the compressor; this can be required to be minimized in which case an after cooler is used or it can be required to be maximized where the hot air will be utilized for something downstream of the compressor. There can also be requirements surrounding the coolers, where the return temperature of the coolant either needs to be minimized or maximized in order to be utilized elsewhere at the facility, or there can be situations where coolant isn't available and either a closed loop cooling skid needs to be provided or air-to-air heat exchangers can be used. All of these factors effect the selection and performance of the compressor.

EFFECT OF SITE CONDITIONS ON SELECTION AND PERFORMANCE

Flow Units

There are a variety of ways the flow of the compressor can be defined. First, the flow can be specified as volume flow or mass flow which are converted from one to the other with equations 5 and 6 as discussed previously. Typically, stages are designed based on volume flow or the dependent flow coefficient. This is done because while the mass flow can vary with site conditions, the volume flow is relatively the same regardless of the site conditions since it only depends on the air velocity and the area. Mass flow also includes the air density or specific volume. From equation 4 it can be seen the specific volume into the first stage and therefore mass flow depends primarily on the inlet pressure and temperature. In most cases pressure is considered a function of elevation and does not change drastically for most air compressor installation sites, and even when it does change it is relatively constant year round. The ambient temperature and therefore inlet temperature into the compressor however, can vary greatly over the course of a year. From equations 4 and 5 it can be seen that for a reduction in ambient temperature, the specific volume will be reduced, and therefore the mass flow of the compressor will be increased. Additionally, the relative humidity can also effect the mass flow, albeit less than the temperature effect. From equations 2-5 it can be seen that a higher relative humidity, meaning there is more moisture in the air, will lead to a higher specific humidity and a higher gas constant for the humid air which will in turn lead to a higher value of specific volume corresponding to a lower value of mass flow.

In addition to volume flow and regular mass flow, there is also pseudo-mass flow or mass flow based on standard conditions. This type of flow is typically expressed in volumetric flow units but with the addition of Standard or Normal, such as standard cubic feet per minute (SCFM) or normal meters cubed per hour (Nm^3/hr). The Standard or Normal is defined by a reference condition, for example SCFM is typically defined by a pressure of 14.7 psia, a temperature of 60 ˚F, and a relative humidity of 0%. With these parameters specified a specific volume can be calculated in the same way it would normally be calculated using equations 1-4 above. This "standard" value of specific volume is then used to convert the compressor's actual mass flow into a pseudo mass flow with volumetric flow units using equation 6.

To further complicate the discussion of flow in air compressors, the mass flow can be specified as wet or dry. Since the air is humid, some of the mass of the air is water vapor. As the name implies wet mass flow includes the water vapor, while dry mass flow does not consider the moisture. This is accomplished by dividing the wet mass flow by one plus the specific humidity as shown below in equation 17.

$$\dot{m}_{dry} = \frac{\dot{m}_{wet}}{(1+SH)} \tag{17}$$

Typically, dry mass flow delivered or at the outlet of the compressor is desired since it is the useable quantity of air that will not be reduced in the event that there is any downstream condensation. For the following discussion on how the specified conditions effect a selection, the flow will be considered to be regular mass flow dry in units such as lb_m/min or kg/hr.

Temperature Effect

As discussed the temperature can have a large impact on the mass flow. To better illustrate this, Figure 5 shows a typical pressure vs. flow plot with multiple temperature conditions. Each curve and marker corresponds to the max capacity (i.e. IGV full open) of the compressor at a corresponding ambient temperature keeping all other parameters the same. As expected, the max capacity mass flow out of the compressor increases as the ambient temperature decreases. Along with the flow increase, the surge point at max capacity also increases in pressure. This is in part due to the fact that as discussed the decrease in ambient temperature leads to a decrease in inlet specific volume, and from equation 14 it can be seen that when the specific volume decreases the discharge pressure of the first stage will increase. Since the first stage typically has the largest pressure ratio this leads to an increase in surge pressure as the ambient temperature decreases. Also, while the max capacity increases, the turndown point and therefore the minimum capacity remains the same. This is due to each curve having a common surge line; although as the temperature decreases the mass flow increases, the inlet volume flow remains the same. In order to turn down the compressor, the inlet volume flow into the first stage is reduced using the IGV. In this way, if for example the 115 deg day flow and pressure was the required design point, the compressor can be turned down to operate there even as the ambient temperature decreases. However, this is not always the case; for example, looking at Figure 5 if the discharge pressure is reduced to 100 psig the compressor would be able to operate there when the ambient temperature is 115 ˚F but as the temperature decreases the compressor will be operating closer and closer to choke. Similarly, if the discharge pressure is increased to 140 psig and the required design flow is 510 lb_m/min, the compressor would be able to operate there when the ambient temperature is 95 ˚F and lower, but on a 115 ˚F day the compressor would not be able to operate there, either the flow or pressure would be less.

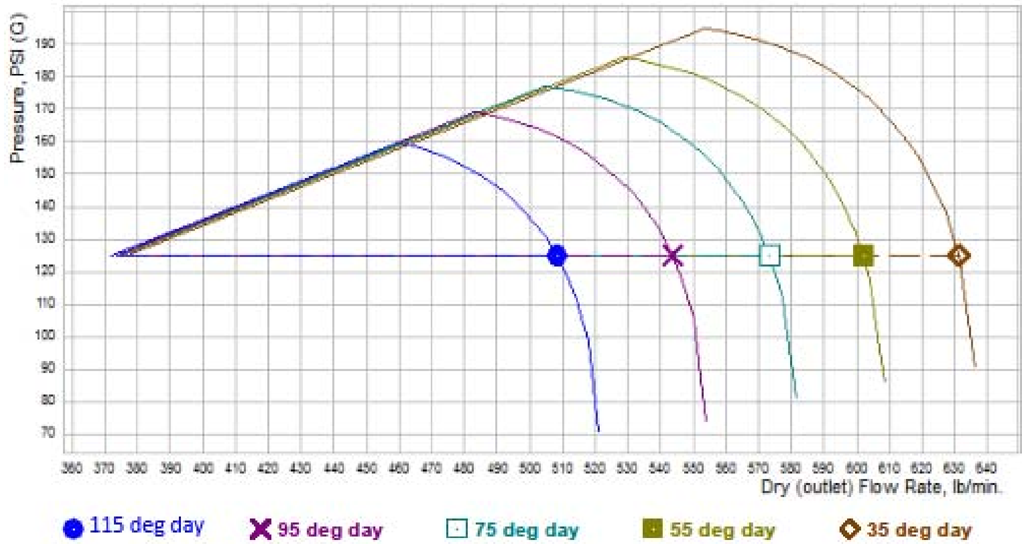

Figure 5. Pressure vs. Flow at Various Ambient Air Temperatures

Given that the ambient temperature has such an effect on the mass flow, it is important to consider when selecting a compressor. For example, when there are multiple conditions given if there is a high ambient condition that is required to meet the design flow and pressure, this condition must be used to size the compressor, otherwise as was discussed above the compressor may not be able to meet the requirements. Sizing the compressor based on a high ambient temperature will increase the size of the compressor and can lead to the compressor having a higher power consumption, a larger motor, and larger coolers. Conversely, if there is a minimum or low ambient condition that is required to meet a turned down flow condition, the compressor needs to be selected with adequate turndown sometimes at the expense of power consumption.

Relative Humidity Effect

In addition to temperature, the relative humidity will also effect the mass flow as discussed above. Figure 5 shows the effect relative humidity has on a pressure vs. flow graph. Similar to Figure 5, each marker on Figure 6 corresponds to the max capacity of the compressor at various relative humidity values with all other parameters kept constant. From this it can be seen that for higher values of specific humidity, the mass flow is decreased as expected based on equations 2-5. Although the relative humidity does not have as much of an effect as temperature, it is also effects the selection of the compressor. Similarly, to a high ambient temperature, a high relative humidity value needs to be considered to size the compressor, since for example in Figure 6 if 540 lb_m/min is required, the below compressor would not be able to operate there when the relative humidity is 80% or greater.

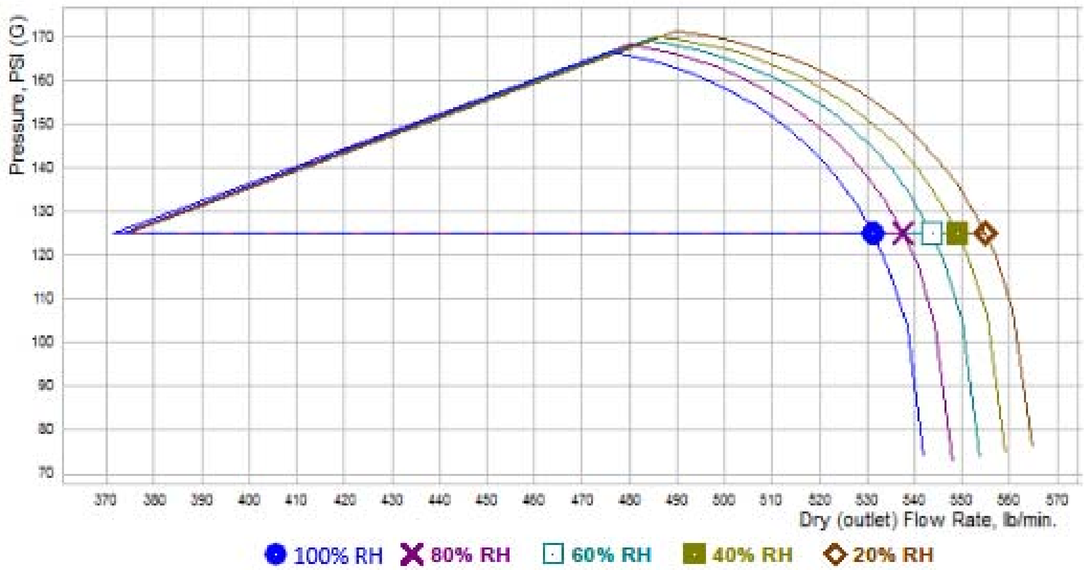

Figure 6. Pressure vs. Flow at Various Relative Humidity Values

Coolant Supply Temperature Effect

Another key consideration for the selection of the compressor is the coolant supply temperature. As mentioned previously the intercooler performance can be given by approach temperature which is the difference of the outlet air temperature and the inlet coolant temperature. For a cooler with a known approach temperature value, if the coolant is supplied at a higher temperature the air discharged from the cooler and into the subsequent stage will be hotter. Similarly to a higher ambient air temperature, as the coolant supply temperature increases the mass flow capacity of the compressor decreases. Figure 7 shows the effect coolant supply temperature has on a pressure vs. flow graph for various coolant supply temperatures.

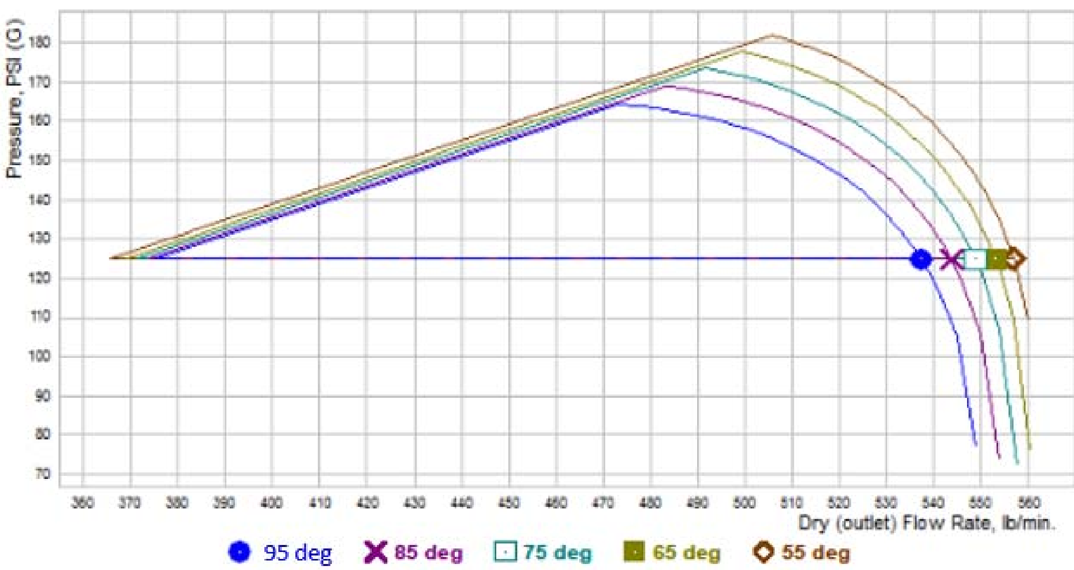

Figure 7. Pressure vs. Flow at Various Coolant Supply Temperatures

Case Studies

Given the influence the ambient conditions have on the selection and sizing of centrifugal air compressors, it is important to use information that accurately depicts the site requirements and ambient conditions. It is obviously desired that the compressor be able to meet the requirements year round, therefore the compressor needs to be sized such that when it is hot and humid the compressor will be able to operate at the required condition. The compressor also needs to have adequate turndown capability so that it can meet any low flow requirements when it is colder. Misstating the values of ambient conditions can result in the compressor being over or undersized for the desired application. The following examples illustrate how specified ambient conditions can effect a compressor selection, and how more well realized conditions can lead to the selection of a better suited machine. While coolant supply temperature can also effect the compressor performance as shown in Figure 7 it will not be included in this case study since the effect is so similar to ambient temperature and relative humidity.

Case 1: Baseline Performance

The first case that will be looked at is the selection of a compressor based on fairly well-realized conditions to serve as a baseline for the subsequent cases. For this selection a total of 6 conditions were considered, three of which require that a specified flow be met at a discharge pressure of 154 psig. Table 1 below gives a summary of the conditions specified, as well as the basic ambient conditions considered.

Table 1. Case 1 Condition Summary

Condition	Description	Pin, psia	Tin, ˚F	RH, %	Flow, lb_m/min
1	Nom Amb, Power Guarantee	14.5	56	65%	**1,566**
2	Nom Amb, Turndown	14.5	56	65%	**1,356**
3	Nom Amb, Max Flow	14.5	56	65%	1,848
4	Max Amb, Req'd Flow	14.5	90	45%	**1,628**
5	Max Flow, Min Amb.	14.5	14	50%	2,000
6	Min Flow, Min Amb.	14.5	14	50%	1,404

In Table 1 the required flows are designated in bold, while the other flow values are determined by the compressor selection. In order to meet these flows at the required discharge pressure the individual stages need to be selected accordingly as discussed previously. Stages with appropriate profiles and diameters are chosen so that the overall compressor performance meets the flow and pressure at the conditions specified. Given the previous discussion that a required flow at a high ambient temperature will size the compressor, condition 4 is the first condition considered. Since at the nominal ambient condition there is also a turndown condition specified, this also needs to be considered in the initial selection of the compressor the ensure this operating point can be achieved. In order to satisfy the max ambient required flow, stages with appropriate profiles are chosen such that the required flow is achievable with impeller diameters that can meet the required discharge pressure of 154 psig. Then to satisfy the turndown case, the impeller diameters selected are increased to increase the pressure rise to surge at maximum capacity of the machine so that there is more turndown available. With these factors considered a selection is made, resulting in a four stage compressor. The compressor operating map for condition 4 in Table 1 can be found below in Figure 8 while the compressor operating map for conditions 1-3 can be found in Figure 9.

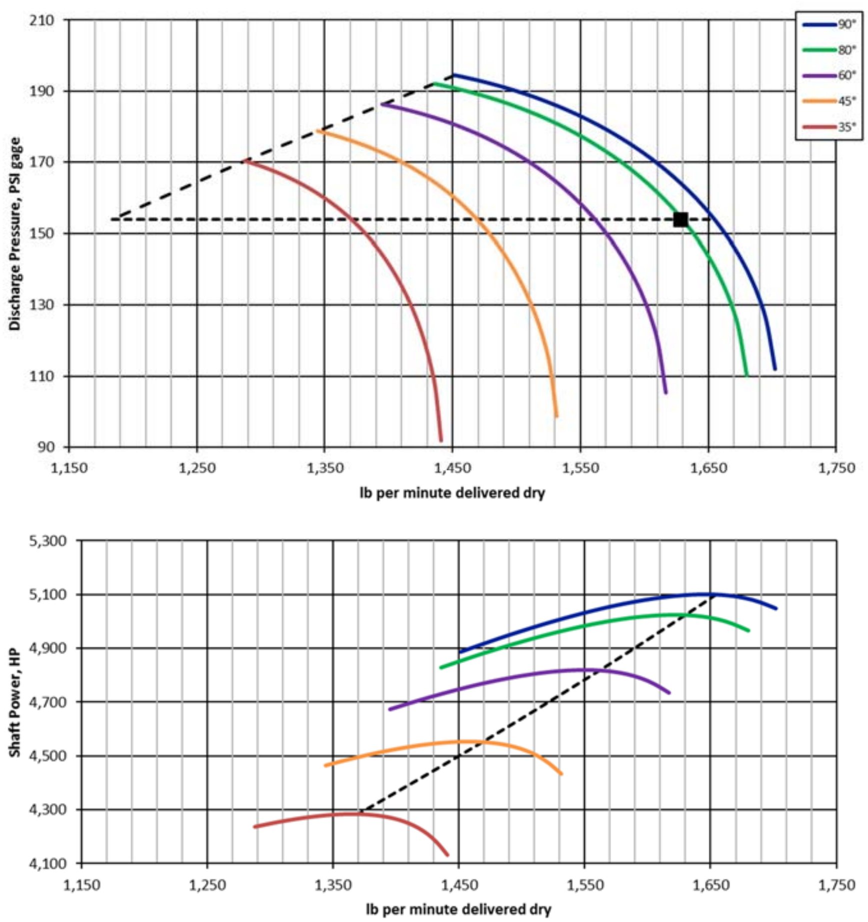

Figure 8. Case 1 Max Ambient Compressor Operating Map

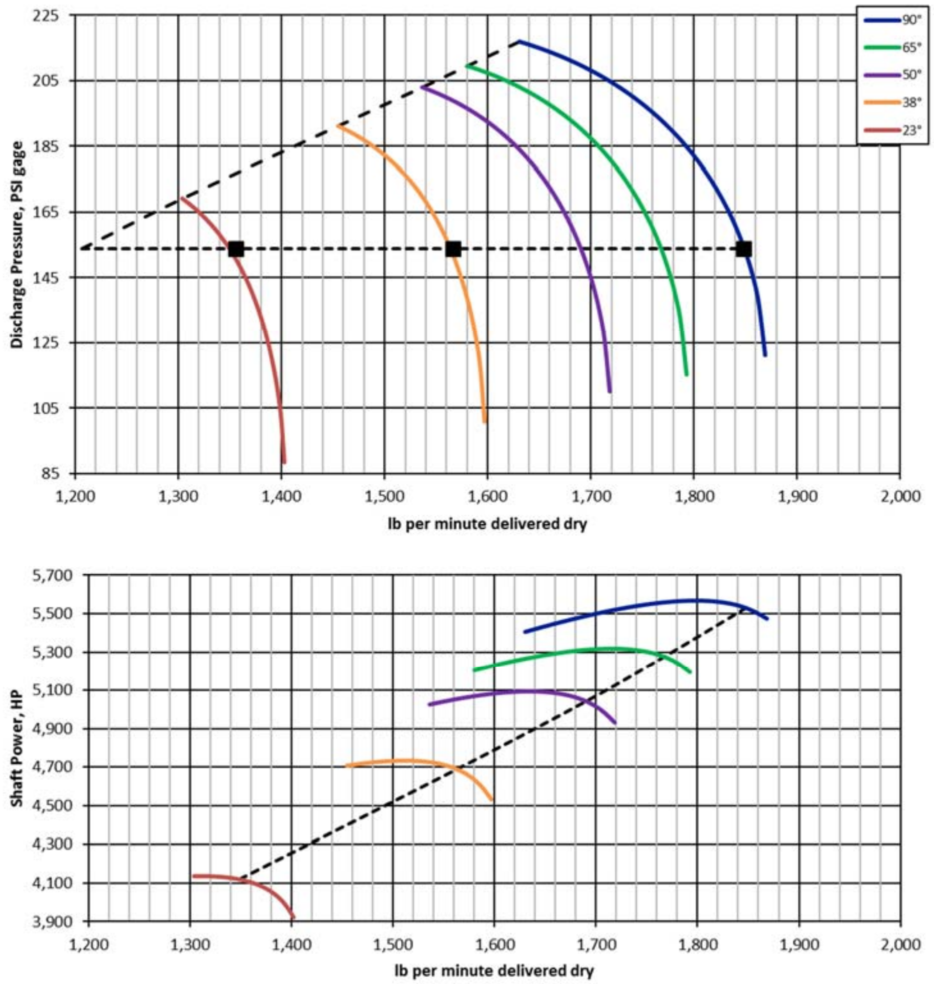

Figure 9. Case 1 Nominal Ambient Compressor Operating Map

From Figures 8 and 9, it can be seen that at the max ambient condition the compressor is operating very close to the maximum capacity, while at the nominal ambient turndown condition, the compressor is operating almost at the minimum capacity as expected. With the compressor size and turndown determined from these two conditions the other conditions can be plotted. Figure 9 also shows the nominal ambient power guarantee condition which can easily be achieved based on the compressor being sized for the max ambient case. Figure 9 also shows the max flow at the nominal ambient condition. In addition to the max ambient and nominal ambient conditions, there are also minimum ambient conditions specified. As discussed previously, a decrease in the ambient temperature leads to an increase in the mass flow through the compressor and therefore the power consumption. In this example the maximum flow at the minimum ambient condition is specified as a required operating condition; therefore, this will be the basis for the motor sizing since it will be the maximum power required to run the compressor. Having the motor sized for the compressor's unthrottled performance at a lower ambient condition is a requirement that can be specified as part of API 672 Special Duty and in other situations is specially requested. Figure 10 below shows the compressor operating map at the minimum ambient condition with the maximum and minimum flows identified.

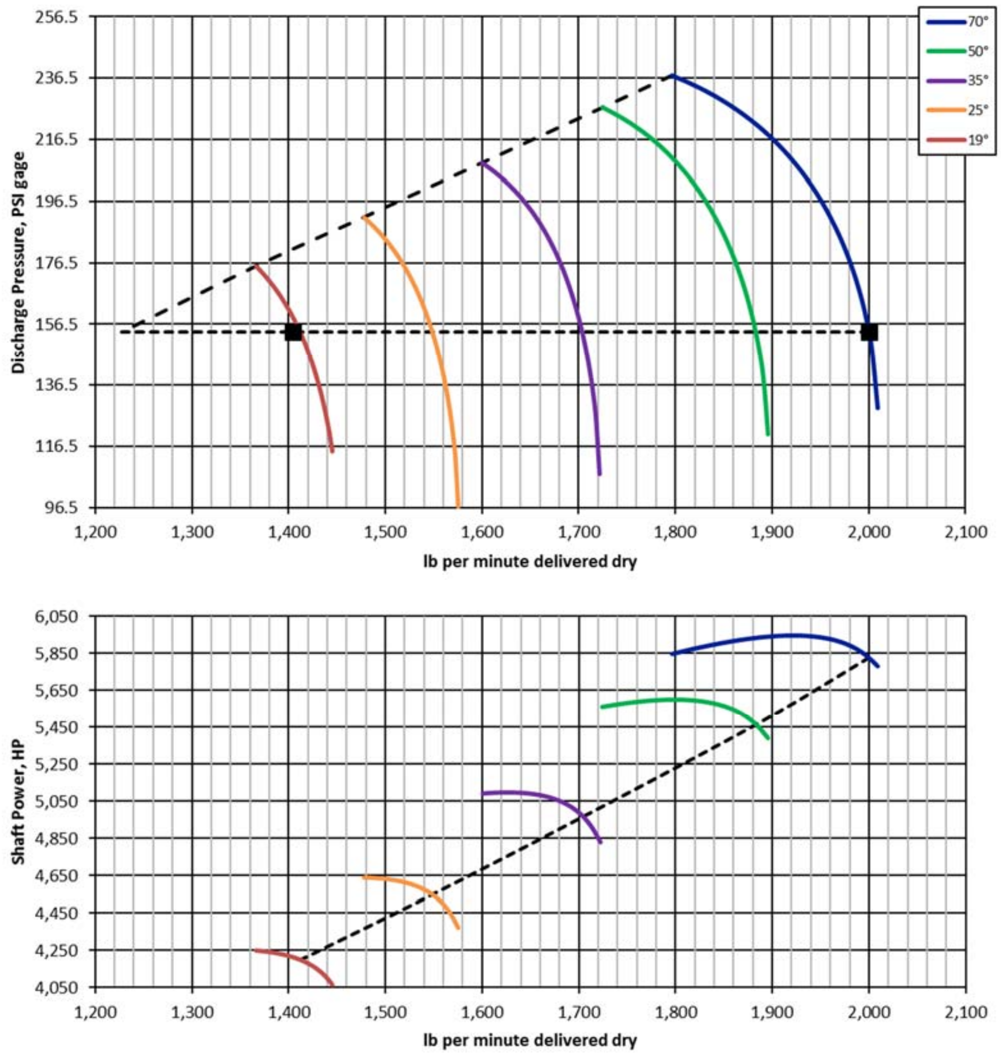

Figure 10. Case 1 Minimum Ambient Compressor Operating Map

Table 2. Case 1 Power Consumption

Condition	Description	Pin, psia	Tin, ˚F	RH, %	Flow, lb_m/min	Power, HP
1	Nom Amb, Power Guarantee	14.5	56	65%	**1,566**	4,695
2	Nom Amb, Turndown	14.5	56	65%	**1,356**	4,140
3	Nom Amb, Max Flow	14.5	56	65%	1,848	5,530
4	Max Amb, Req'd Flow	14.5	90	45%	**1,628**	5,024
5	Max Flow, Min Amb.	14.5	14	50%	2,000	5,827
6	Min Flow, Min Amb.	14.5	14	50%	1,404	4,175

Table 2 provides the same summary of the conditions given in Table 1, but now includes the power consumption in horsepower at each condition. As discussed the maximum flow at the minimum ambient condition will control the motor size, which in this case results in a 6000 HP motor. These power values will serve as the baseline for the cases to follow.

Case 2: Reduced Flow at Maximum Ambient Condition

Given that a required flow at a maximum ambient condition will size the compressor it is common to reduce the required flow at this condition or make it not a requirement and instead accept the flow available when the compressor is sized for some other condition. This keeps the compressor from being oversized when the ambient conditions are more typical of values encountered for a majority of the year. While not always possible as it can be critical that the compressor never operate undersized, reducing the flow at a maximum ambient temperature condition can lead to lower power values during more common ambient conditions. With this in mind, this case will look at the same ambient conditions as Case 1 but the required flow at the maximum ambient condition is reduced 10%. Table 3 below gives a summary of the conditions considered for Case 2. The required discharge pressure is again 154 psig.

Table 3. Case 2 Condition Summary

Case	Condition	Pin, psia	Tin, ˚F	RH, %	Flow, lb_m/min
1	Nom Amb, Power Guarantee	14.5	56	65%	**1,566**
2	Nom Amb, Turndown	14.5	56	65%	**1,356**
3	Nom Amb, Max Flow	14.5	56	65%	1,650
4	Max Amb, Req'd Flow	14.5	90	45%	**1,465**
5	Max Flow, Min Amb.	14.5	14	50%	1,848
6	Min Flow, Min Amb.	14.5	14	50%	1,404

Based on the maximum ambient condition required flow being reduced, the compressor can now be sized smaller than it was for Case 1. This is accomplished by selecting stages with smaller profiles in order to match at the reduced flow. The compressor selected for this example consists of four stages, and the reduction in maximum ambient required flow leads to all four stages being selected with smaller profiles than was selected for Case 1. In addition to the reduced maximum flow corresponding to smaller stage profiles, the reduction in max flow also leads to a reduction in the amount of turndown required to meet the turndown condition. This leads to the third stage being selected with a smaller diameter, since not as much head rise is required to achieve the turndown. Since the compressor is sized smaller than it was for Case 1 the corresponding maximum flows at the nominal ambient and minimum ambient conditions are also reduced. The required nominal ambient power guarantee and turndown conditions are maintained and achieved based on the compressor selection. Figures 11, 12, and 13 show the compressor operating maps for the maximum ambient, nominal ambient, and minimum ambient conditions respectively. Based on the reduction in flow and the compressor being sized smaller it is expected the power will decrease, which is the case and is summarized below in Table 4. From this information it can be seen that at the nominal ambient, power guarantee condition which typically corresponds to the most common ambient conditions, the power is reduced 95 HP which equates to a 2.02% power reduction compared to Case 1 to go along with a 95 HP (2.35%) reduction in power at the required turndown flow. In addition to these power savings, the max flow and therefore power for the minimum ambient condition is also reduced. Since this condition sizes the motor, the reduction in flow allows for a smaller, cheaper motor to be utilized with this compressor. While not always possible it can be seen how accepting lower flows at high ambient conditions that may only be encountered a few times in a year, can improve the performance of the compressor during more typical operation.

Table 4. Case 2 Power Summary

Case	Condition	Pin, psia	Tin, ˚F	RH, %	Flow, lb_m/min	Power, HP	Flow Δ, lbm/min	Power Δ, HP	Power % Diff
1	Nom Amb, Power Guarantee	14.5	56	65%	**1,566**	4,600	0	-95	-2.02%
2	Nom Amb, Turndown	14.5	56	65%	**1,356**	4,043	0	-98	-2.35%
3	Nom Amb, Max Flow	14.5	56	65%	1,650	4,846	-198	-684	-12.37%
4	Max Amb, Req'd Flow	14.5	90	45%	**1,465**	4,455	-163	-569	-11.33%
5	Max Flow, Min Amb.	14.5	14	50%	1,848	5,274	-152	-553	-9.49%
6	Min Flow, Min Amb.	14.5	14	50%	1,404	4,073	0	-102	-2.45%

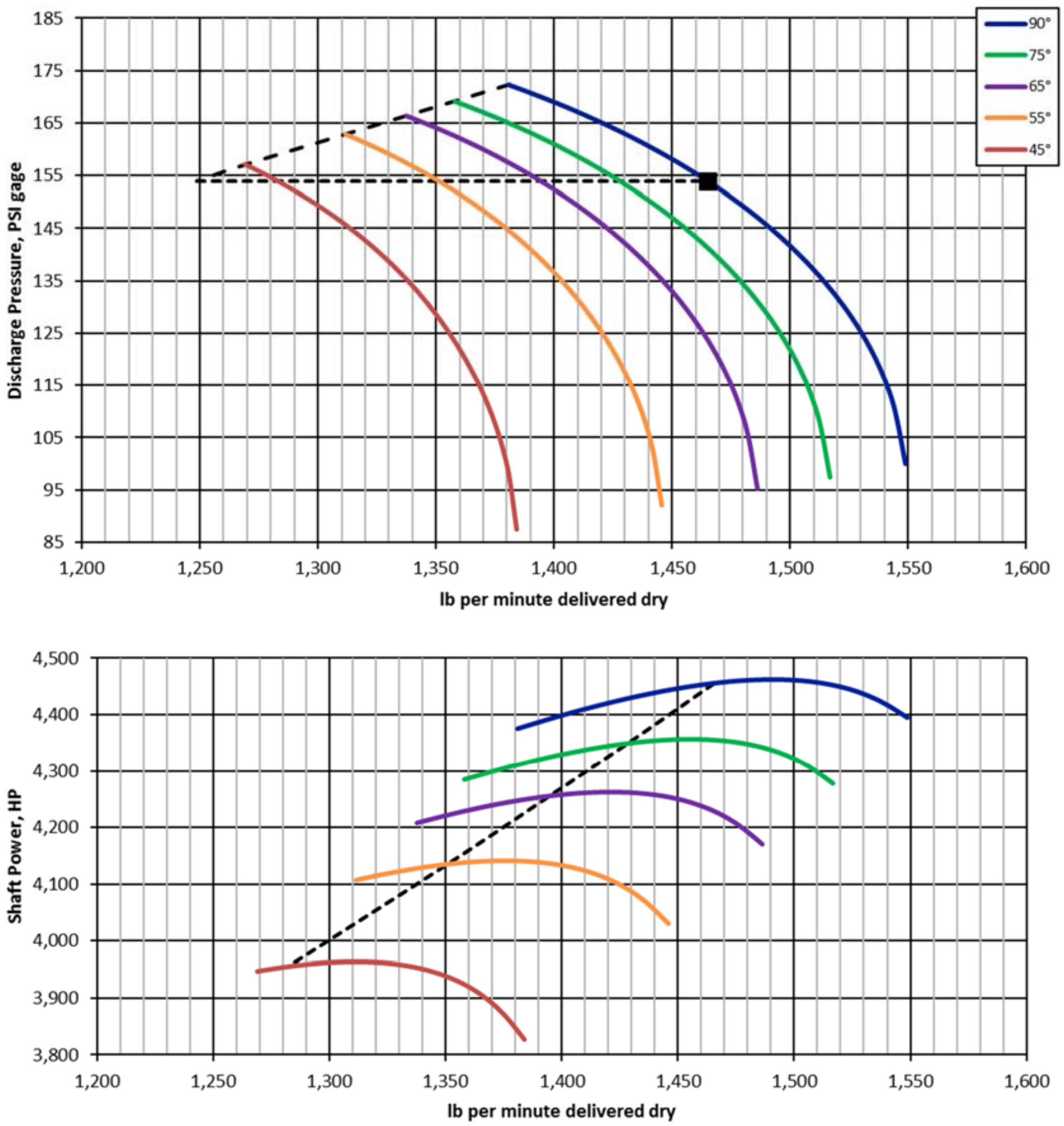

Figure 11. Case 2 Maximum Ambient Compressor Operating Map

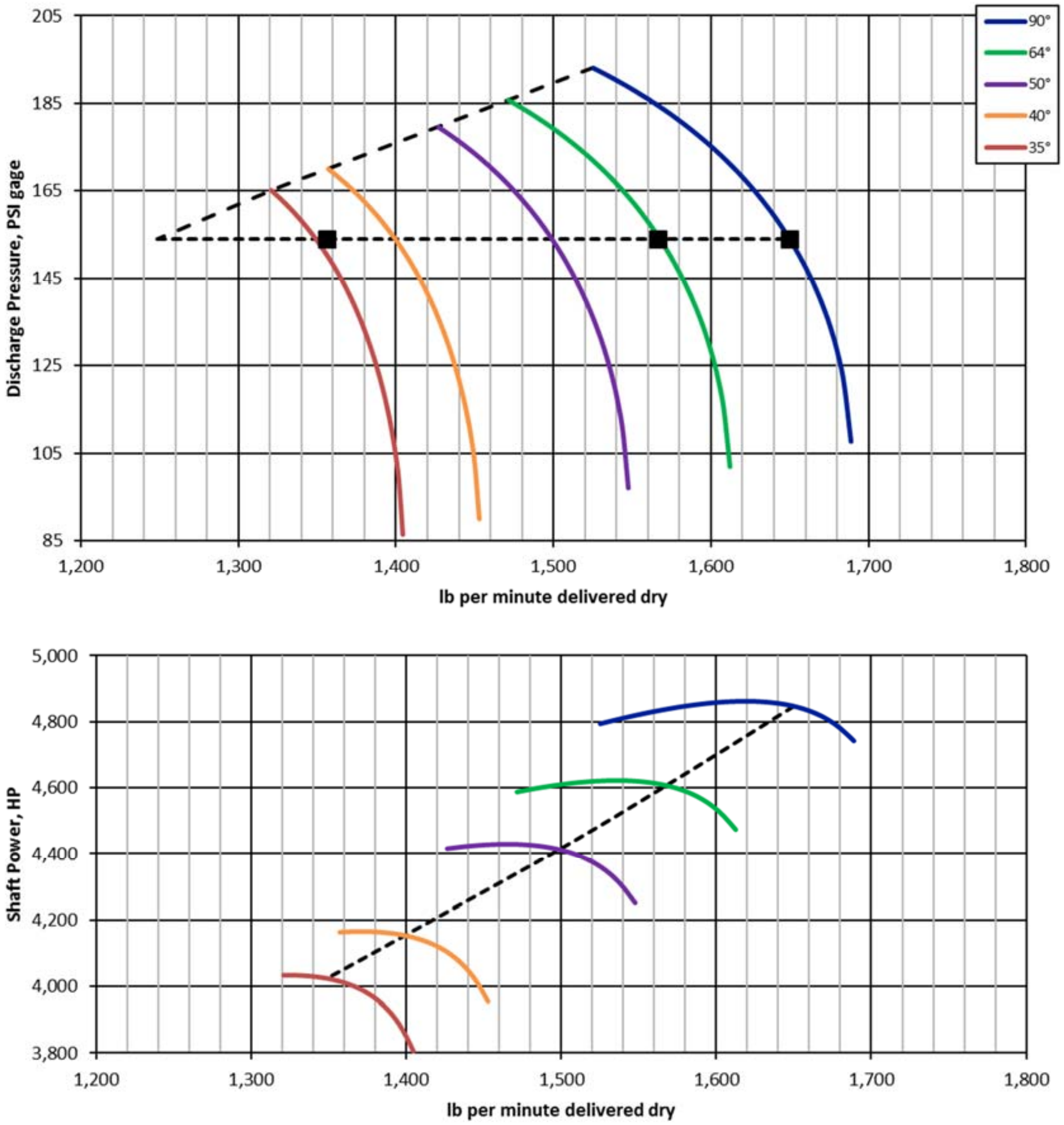

Figure 12. Case 2 Nominal Ambient Compressor Operating Map

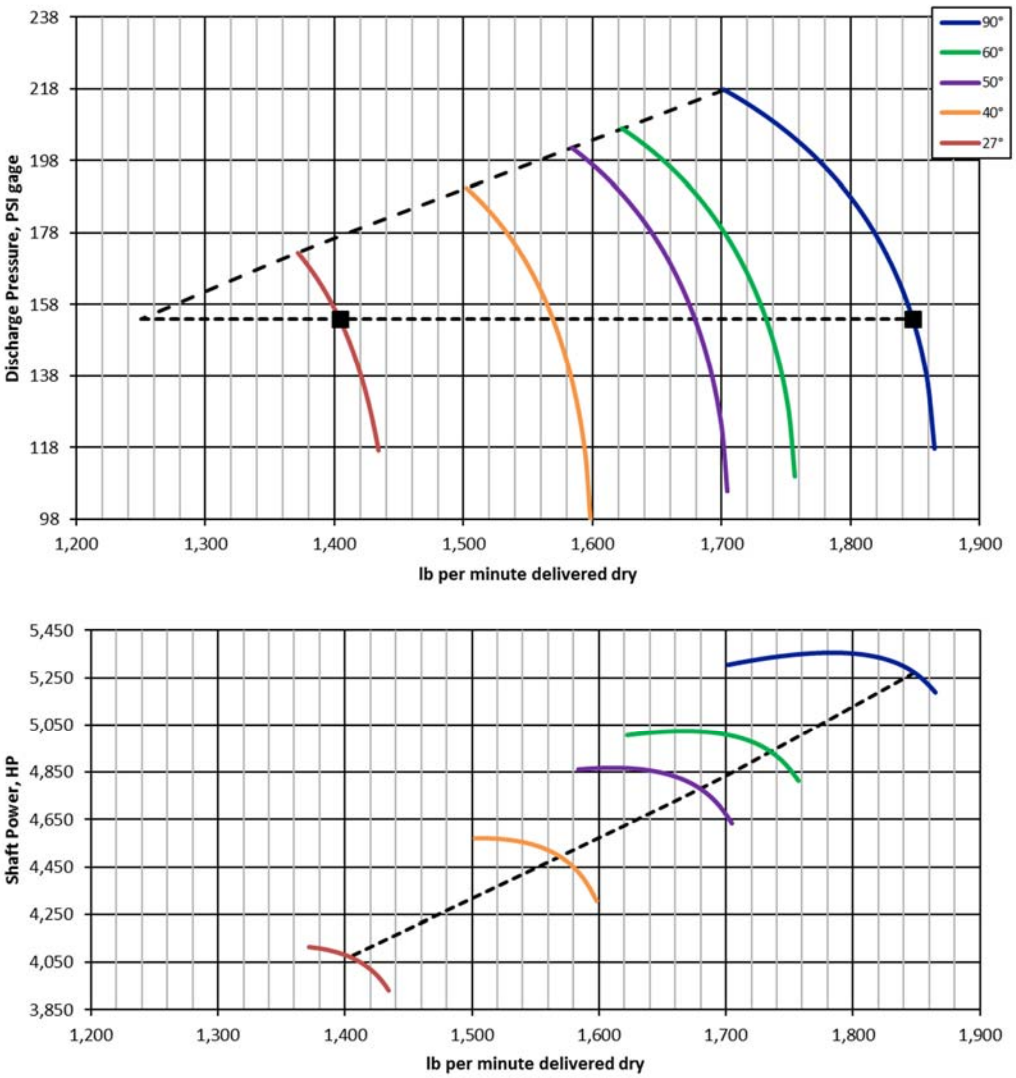

Figure 13. Case 2 Minimum Ambient Compressor Operating Map

Case 3: Extreme Ambient Conditions

When selecting centrifugal compressors, it is fairly common for maximum ambient conditions to be supplied in extreme combinations. For example, high ambient temperatures are given along with a relative humidity value of 100%. A relative humidity of 100% occurs when the air temperature known as the dry bulb temperature is equal to the dew point temperature, which is the temperature when water vapor starts to condensate out of the air. Having a high ambient air temperature (>100 ˚F) at 100% relative humidity is something that does not naturally occur. Dew point temperature higher than 92 ˚F seldom occurs anywhere in the world. As seen from Figures 5 and 6 a high ambient temperature and a high relative humidity will reduce the flow of the compressor. When the flow is given as a requirement at such a condition, it can lead to the compressor being oversized and inefficient at more typical ambient conditions. This case will again look at similar conditions to Case 1, except the ambient temperature of the max ambient condition is increased to 110 ˚F with a relative humidity of 100%. In order to compare the two cases one to one, the flow at each condition is matched between the two and the discharge pressure required is maintained at 154 psig. A summary of Case 3 is given below in Table 5.

Table 5. Case 3 Condition Summary

Case	Condition	Pin, psia	Tin, ˚F	RH, %	Flow, lb_m/min
1	Nom Amb, Power Guarantee	14.5	56	65%	**1,566**
2	Nom Amb, Turndown	14.5	56	65%	**1,356**
3	Nom Amb, Max Flow	14.5	56	65%	1,848
4	Max Amb, Req'd Flow	14.5	110	100%	**1,628**
5	Max Flow, Min Amb.	14.5	14	50%	2,000
6	Min Flow, Min Amb.	14.5	14	50%	1,404

With these conditions in mind, the max ambient condition again is used to size the compressor while the turndown at nominal ambient is also considered. Since the higher ambient temperature and relative humidity will lower the flow capacity, the compressor needs to be sized larger than it was in Case 1. The compressor again consists of four stages, but compared to Case 1 each of the stages is now required to be selected with a larger profile in order to meet the required flow at the max ambient conditions. In addition to this, in order to maintain the turndown level required at the nominal ambient condition the second and fourth stages are selected at larger diameters in order to have the head rise necessary to achieve the turndown. Table 6 shows the increases in power that are encountered by sizing the compressor for this unrealistic max ambient condition. It can be seen that with the two cases matched in flow, the power in Case 3 is higher for every condition as expected. For case 1 which corresponds would be the most typical operating condition of the compressor, the power is increased 150HP (3.19%) by sizing the compressor this way. Had the coolant supply temperature been similarly given at an unrealistically high value, the issue would be further compounded and the power increase would be even greater. In addition to the power increase the power at the minimum ambient max flow condition has exceeded 6000 HP, and the actual max flow capacity leads to an even higher power which would trigger the use of a higher horsepower, more expensive motor than was used for Case 1. Figure 14, 15, and 16 show the compressor operating maps for Case 3.

Table 6. Case 3 Power Summary

Case	Condition	Pin, psia	Tin, ˚F	RH, %	Flow, lb_m/min	Power, HP	Flow Δ, lbm/min	Power Δ, HP	Power % Diff
1	Nom Amb, Power Guarantee	14.5	56	65%	**1,566**	4,845	0	150	3.19%
2	Nom Amb, Turndown	14.5	56	65%	**1,356**	4,288	0	147	3.56%
3	Nom Amb, Max Flow	14.5	56	65%	1,848	5,659	0	129	2.33%
4	Max Amb, Req'd Flow	14.5	110	100%	**1,628**	5,298	0	274	5.46%
5	Max Flow, Min Amb.	14.5	14	50%	2,000	6,005	0	178	3.05%
6	Min Flow, Min Amb.	14.5	14	50%	1,404	4,366	0	191	4.57%

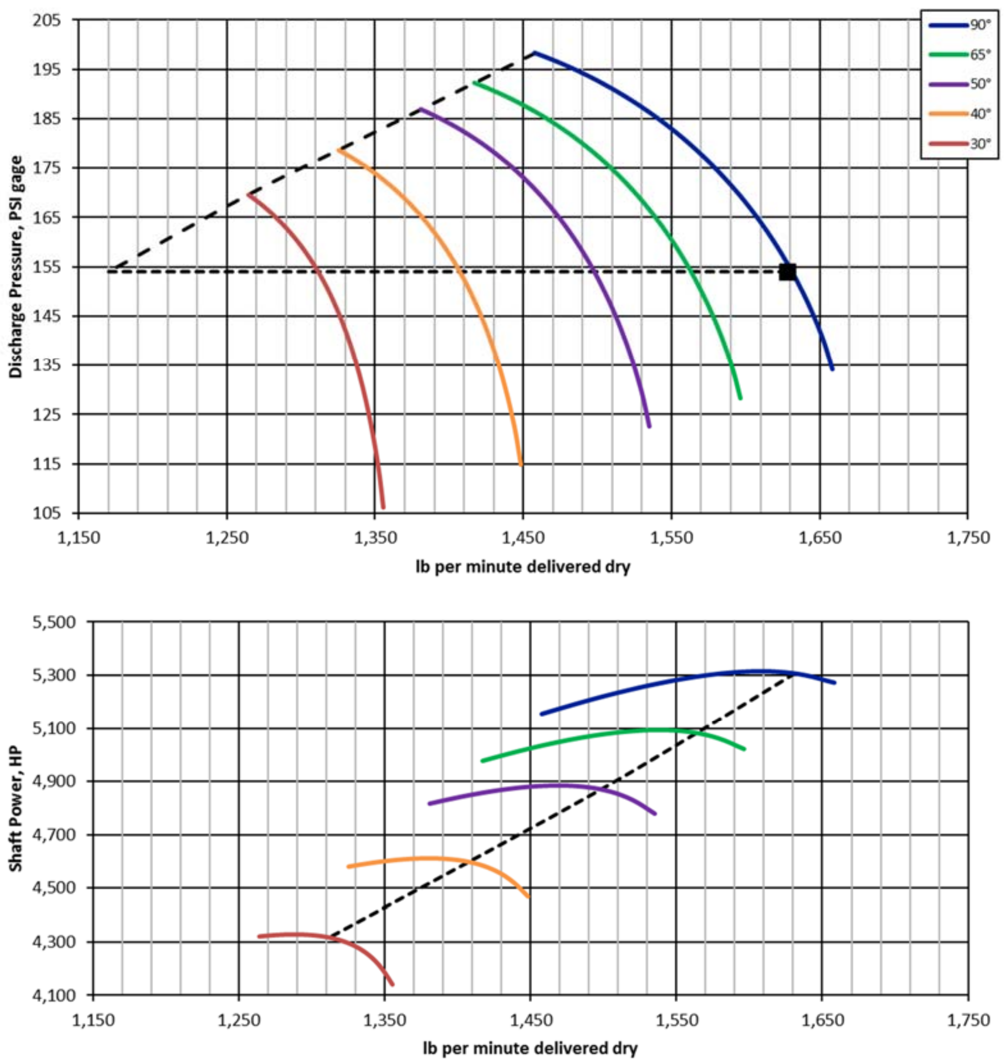

Figure 14. Case 3 Maximum Ambient Compressor Operating Map

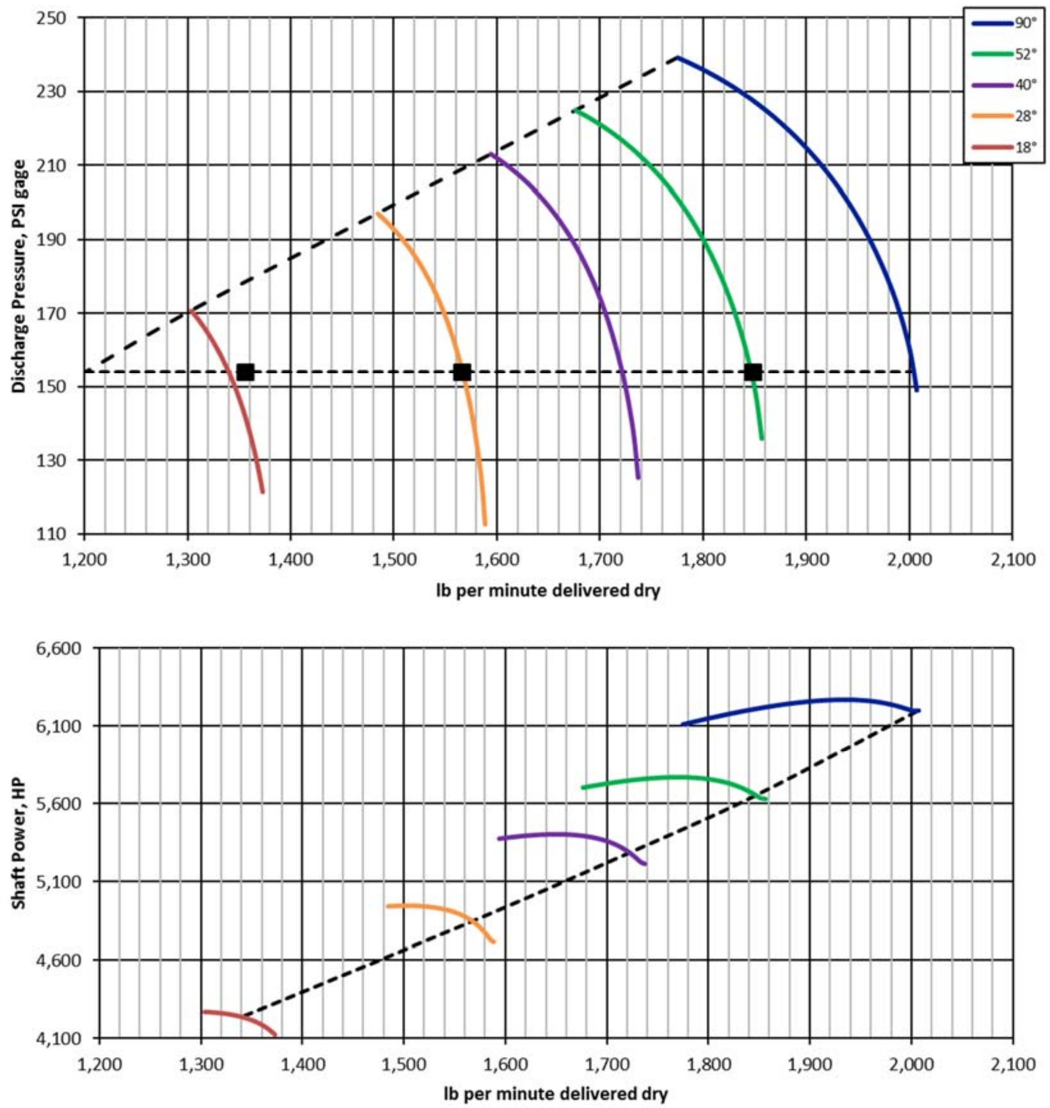

Figure 15. Case 3 Nominal Ambient Compressor Operating Map

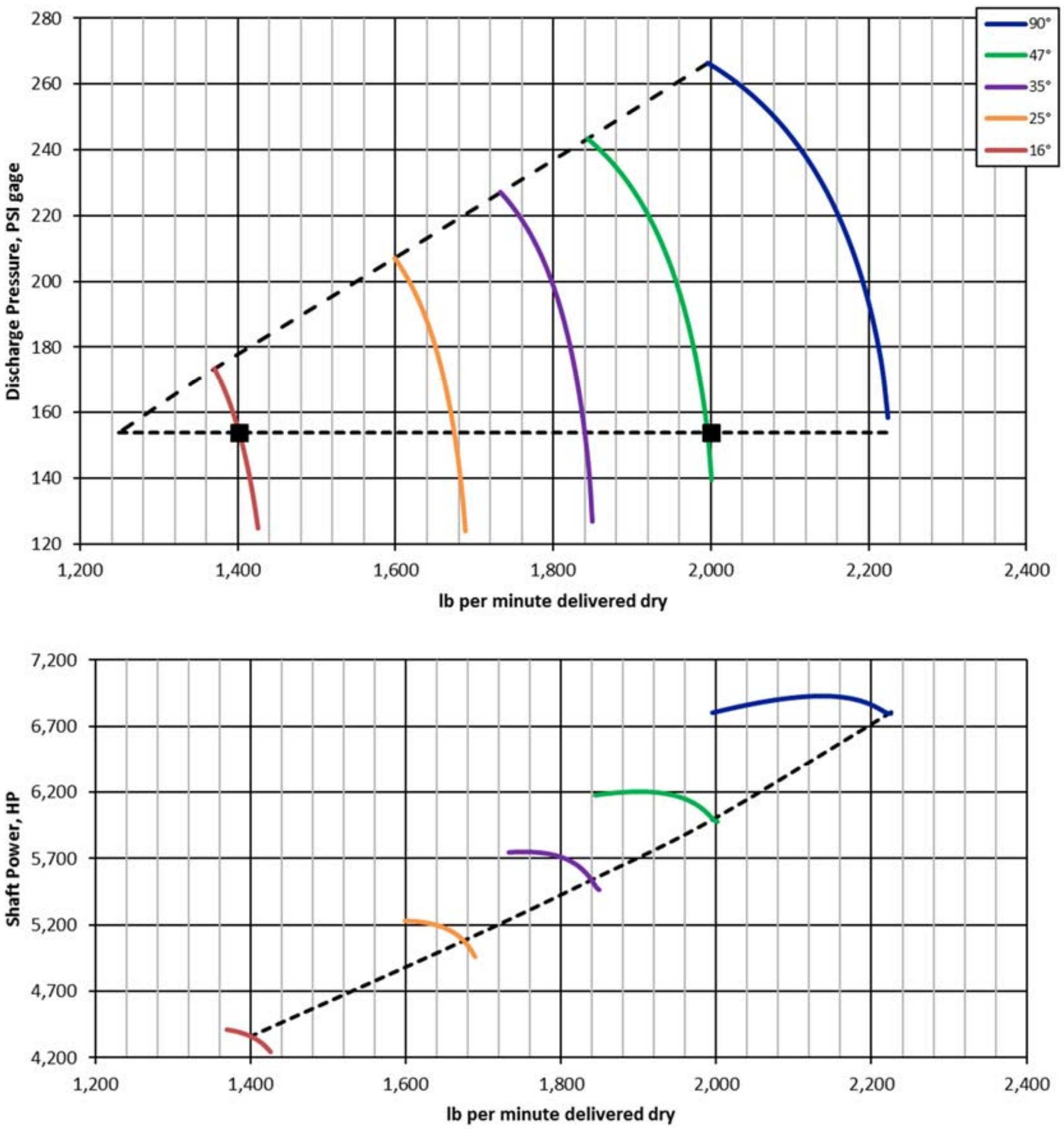

Figure 16. Case 3 Minimum Ambient Compressor Operating Map

IGV OPERATION AND COMPRESSOR CONTROL

IGV Performance

As discussed, the compressor is controlled with the unloading control valve and typically an inlet guide vane. These are two key components that enable to compressor to meet multiple operating conditions and function properly as the ambient conditions change. As the UCV closes, the compressor builds back pressure, and therefore the individual stages operate at higher pressure ratios and lower flows. In order to maintain the discharge pressure but alter the flow, the inlet guide vane is utilized. In order to reduce the flow at a set pressure the UCV actuates as needed to maintain the pressure while the IGV closes, lowering the flow of the compressor. How this works is more complicated than the UCV, while closing the IGV does simply correlate to a change in flow, it does so because it is altering the performance of the first stage of the compressor.

With an IGV fully open the incoming flow to the first stage impeller is said to be axial, it is directed straight into the inlet of the impeller. The easiest way to depict this is through the use of a velocity triangles as shown in Figure 17. A velocity triangle can be used to determine how well aligned the incoming air flow is with the impeller blades. This is typically discussed in terms of relative velocities. In Figure 17 the impeller would be rotating from left to right. The axial velocity is the incoming air velocity in the inlet piping. The relative velocity is the velocity relative to the rotating impeller; it is essentially what the incoming velocity looks like if you were to be sitting on the rotating impeller. The angle between the axial velocity and the relative velocity is known as the relative flow angle. This is then compared to the inlet blade angle of the impeller and in general at the design point, the relative velocity should be well aligned with the impeller blade. For constant ambient conditions, as the inlet flow is reduced this will correspond to a reduction in volume flow, and since the area is fixed, the inlet axial velocity will decrease as shown on the right in Figure 17. This reduction in axial velocity will in turn lead to the relative velocity no longer being well aligned with the impeller blades and can eventually cause surge.

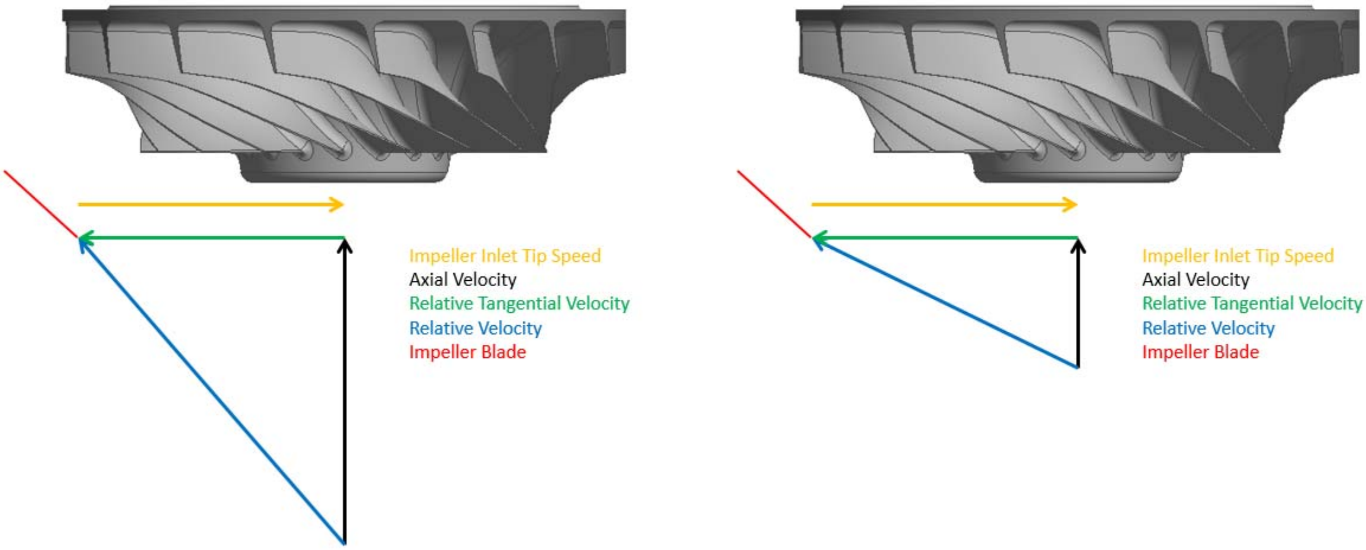

Figure 17. Axial Flow Velocity Triangles

As the IGV closes it imparts what is known as preswirl onto the air flow; that is that the air flow is already swirling in the direction of rotation of the impeller. This causes the inlet flow to the impeller, also known as the absolute velocity to no longer be entirely axial but to instead be at an angle known as the absolute flow angle and introduces an absolute tangential velocity similar to the relative tangential velocity. The sum of the absolute values of these tangential velocities is equal to the inlet tip speed. As can be seen from Figure 18, the introduction of preswirl allows the relative velocity to be better aligned with the blade for the same axial velocity that is shown on the right in Figure 17. This allows the compressor to still operate efficiently at reduced flow.

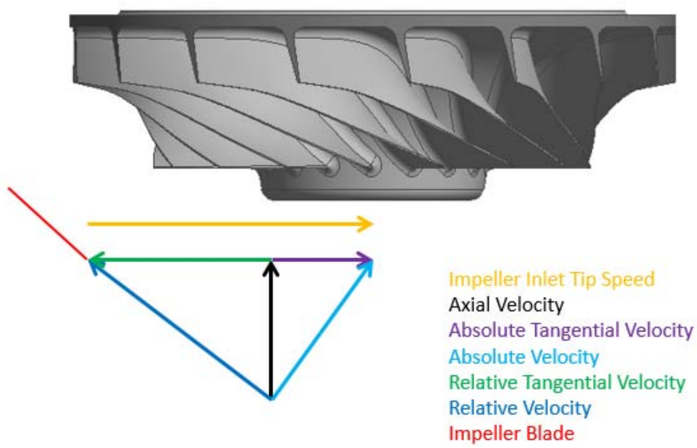

Figure 18. Preswirl Flow Velocity Triangle

This introduction of preswirl changes how the impeller operates. Rather than having a single set of non-dimensional coefficient curves as discussed previously and shown in Figure 4, a first stage with an upstream inlet guide vane will now have a range of curves it can be operated over, an example of which is shown in Figure 19. On the figure 0 deg. IGV angle corresponds to the IGV being fully open and becoming increasingly closed as the IGV angle increases.

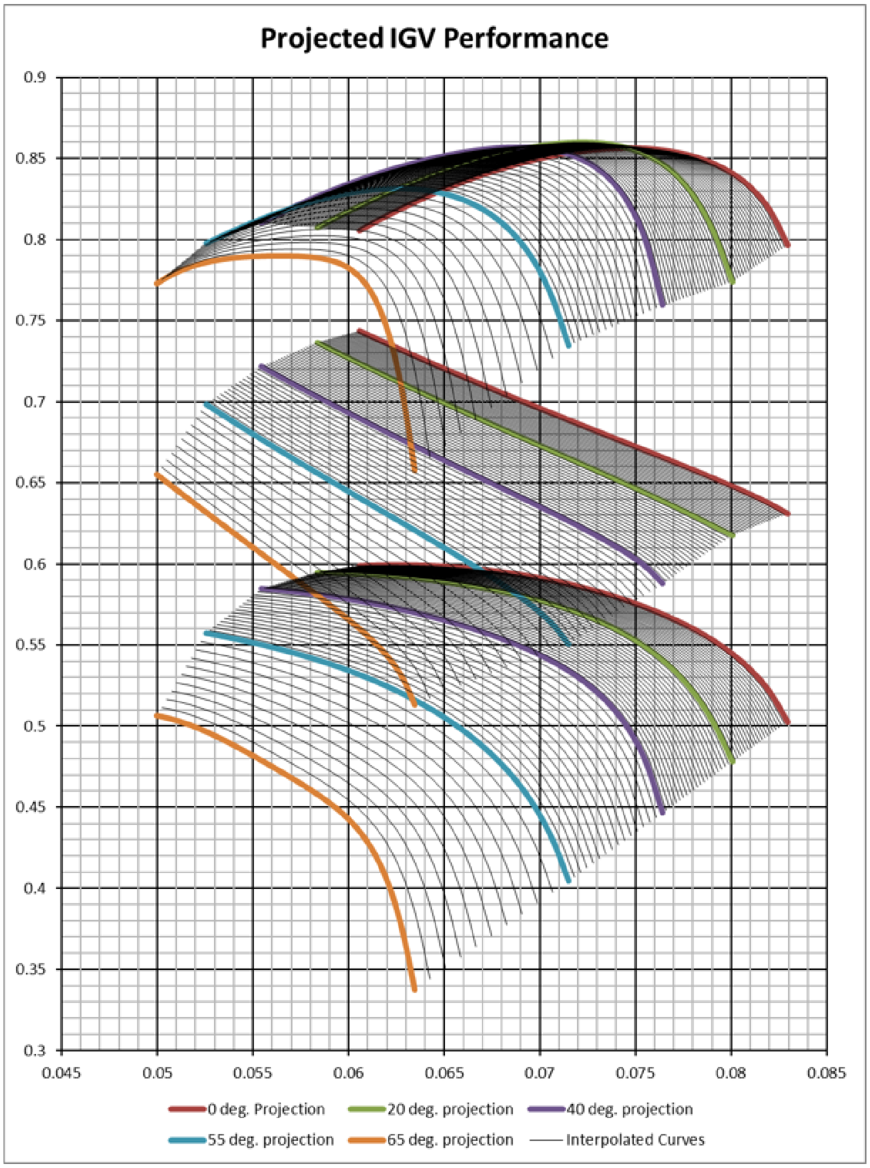

Figure 19. Typical IGV Operating Map

For set ambient conditions, the IGV is used to vary the volume flow. Even though the ambient conditions (and therefore the inlet specific volume) are constant, the mass flow through the first stage will be reduced as shown in equation 5. Since the volume flow is being reduced, the flow coefficient will be reduced, and it can be seen from Figure 19 that the first stage can still operate at similar values of head coefficient even as the volume flow is reduced, meaning the machine is able to maintain discharge pressure. Since the mass flow through the first stage is reduced, the mass flow through the rest of the machine will be similarly reduced, however the volume flow into subsequent stages will be in line with the typical values they are designed for since the flow velocities will be similar and the areas will be the same. This allows the compressor to be operated over a wide range of flow at a constant pressure. IGVs can also be installed on subsequent stage to further improve the turndown capability.

Figure 19 also shows a common limitation of turning down with an IGV. As the IGV angle is reduced the head coefficient curve's slope increases and the curve becomes increasingly vertical. As this happens, relatively small fluctuations in volume flow can lead to large changes in head and therefore pressure. When operating at higher IGV angles (meaning the IGV is more closed), the compressor can become more susceptible to surge. This trend is also seen in the previous compressor operating curves that have been shown that as the flow is reduced the rise to surge is also reduced.

Compressor Control

As previously mentioned the compressor is controlled through the use of an upstream IGV and a UCV at the compressor discharge. Figure 20 shows an example P&ID diagram of a three stage compressor showing the location of the IGV and UCV.

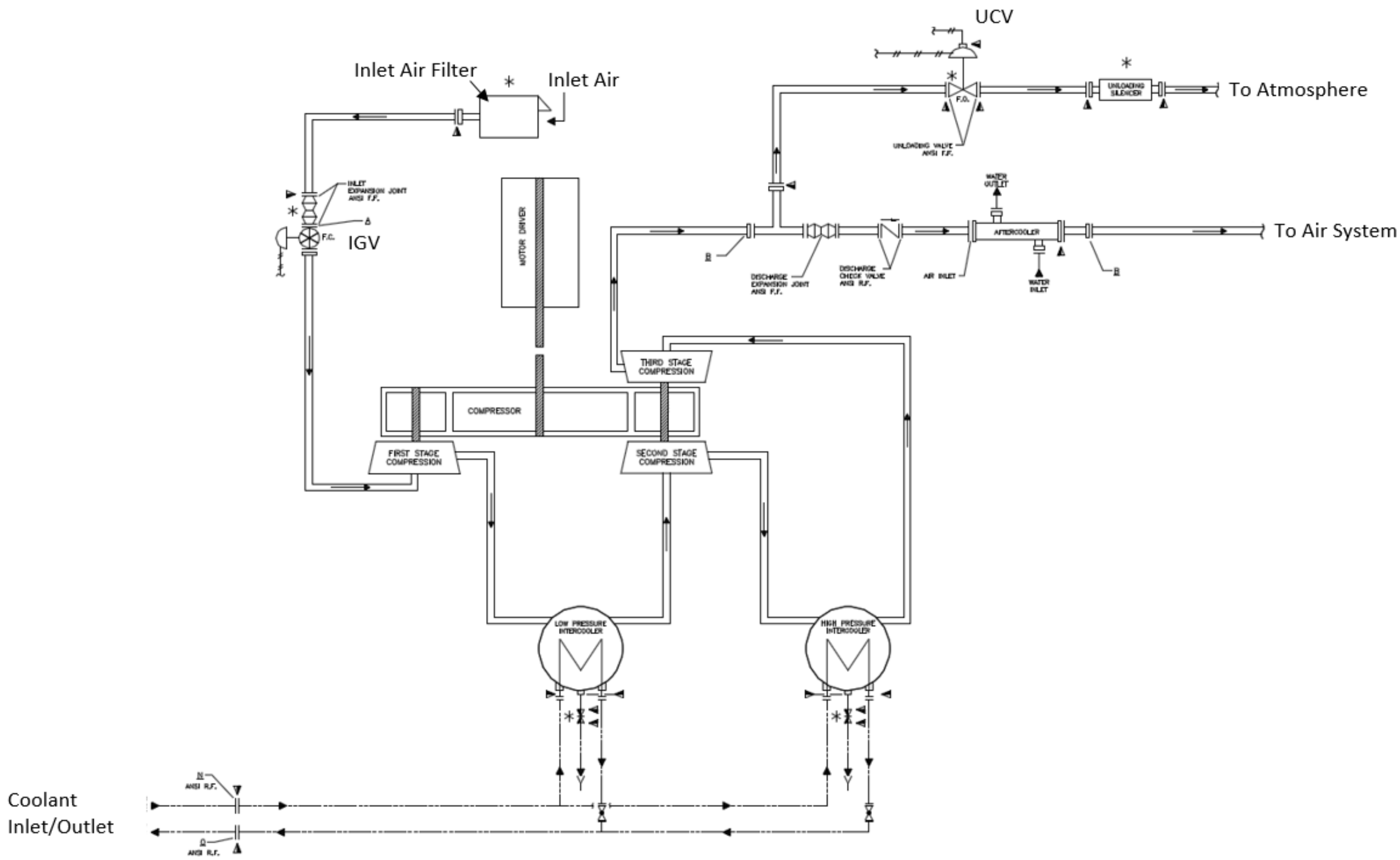

Figure 20. Example Compressor P&ID

Without the IGV and UCV the compressor would operate uncontrolled, and if the site conditions, coolant supply/temperature, motor speed, and several other factors match what the compressor was sized for it would operate as intended. However, if the downstream flow demand were to change the compressor would move up or down its operating curve and the discharge pressure (which can be measured by a pressure transmitter at the compressor discharge) would also change, which could lead to surge or choke of the compressor. With the UCV installed the discharge pressure can be maintained by opening or closing to relieve or build back pressure as necessary. The UCV will maintain the pressure but the flow is still uncontrolled. As the ambient temperature changes, the mass flow of the compressor will also change, so if for example the ambient temperature decreases the mass flow if the compressor would increase while the UCV maintains the discharge pressure, essentially what is shown above in Figure 5. Given that the mass flow is increasing the power draw of the main drive motor will also increase as shown by equation 16, and if the motor was not sized for this lower ambient condition, it could become overloaded. To avoid this and enable control over the compressor's flow, the IGV is installed. From an aerodynamic point of view, the IGV is controlling the flow of the compressor but given the relationship between flow and gas power and therefore motor power, from a controls perspective the IGV is controlling the motor power. This can be done by monitoring the motor amps using a motor amp transmitter, which is related to the motor horsepower by equation 18 for a known motor voltage, power factor, and motor efficiency.

$$Amps = \frac{431*MHP}{Volts*PF*Eff} \tag{18}$$

In this way with the design horsepower known at the required flow and pressure, the compressor flow can be maintained using the IGV while the UCV maintains the discharge pressure. There are other ways to control the compressor such as directly measuring the flow and actuating the IGV based on the measurement, which works in essentially the same way as motor amp control. The remaining discussion will be based on motor amp control. An example of how the compressor is controlled is shown in Figure 21. Point A corresponds to the required flow and pressure where the compressor will operate at a determined shaft power. The IGV and UCV will operate together to maintain the compressor at this operating point. The type of operation is typically known as base mode.

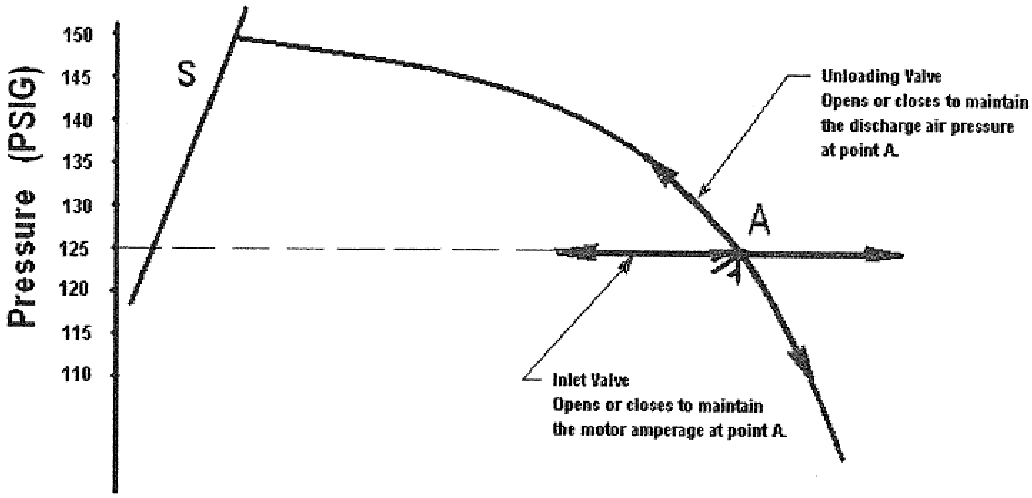

Figure 21. Base Mode Compressor Control Curve

Figure 21 shows how a single operating point is maintained, but as discussed previously it is typical for turndown to be required so that when the demand for air decreases the compressor can operate at a reduced flow while still maintaining the discharge pressure. This is accomplished by determining the min and max motor amp set points. As discussed previously and shown on Figure 2 the compressor has a useable operating range. The max motor amp set point typically corresponds to the maximum flow the compressor was sized for while the minimum amp set point corresponds to the minimum flow achievable by the compressor while maintaining an acceptable pressure rise to surge. The min amp set point can be predicted based on the expected performance of the compressor, but it is ultimately determined by suction surging the compressor at the installation site. A suction surge occurs when the compressor's flow is reduced to the turndown point at the required discharge pressure as shown on Figure 2. The minimum amp set point is then determined by offsetting the surge amp point by an adequate value. The compressor can then operate between the minimum and maximum amp set points depending on the plant demand. This is illustrated below in Figure 22 which is showing what is known as suction throttle mode. In this mode as long as the required discharge pressure is being met, the inlet guide vane will close until the compressor is operating at the minimum amps set point, thereby reducing the flow and accompanying power consumption of the machine. While operating at the minimum amps set point if the discharge pressure is no longer being achieved and the unloading valve is fully closed, the inlet guide vane will then start to open, increasing the flow of the compressor and meeting the required discharge pressure.

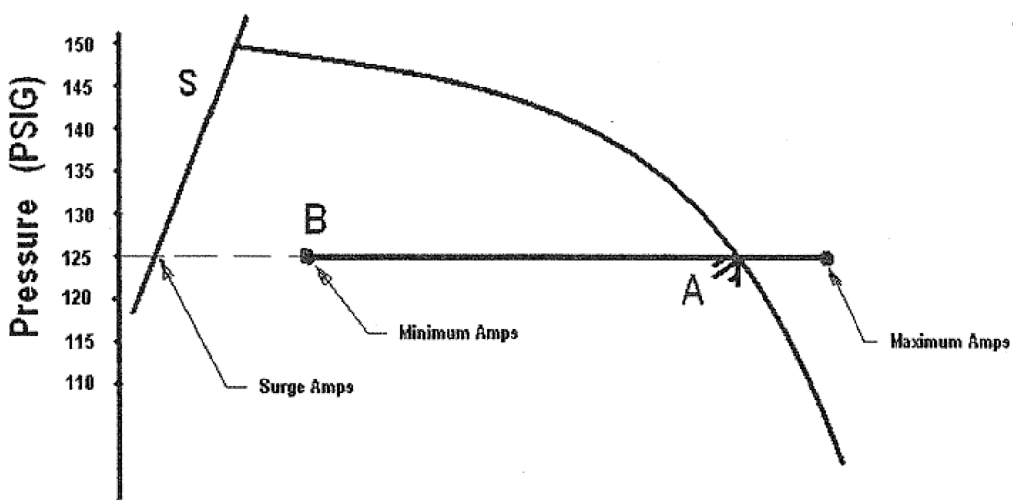

Figure 22. Suction Throttle Mode Compressor Control Curve

While base mode and suction throttle, can meet the high demands of air and suction throttle can meet a range, neither control mode accounts for when the demand for air is very low or nothing. In this case intermittent mode can be used. This control mode is shown below in Figure 23. When the demand for air from the compressor is high, it will essentially operate in base mode, maintaining the discharge pressure. As the demand decreases, the unloading valve will open to maintain the discharge pressure set point. Once the unloading valve is at a certain % open where most of the air is being vented to atmosphere, the control panel will start a timer, typically two minutes. Once this timer expires if the unloading valve is still venting most of the air, the UCV will open fully and the IGV will close. In this state the compressor will continue to run unloaded as shown by point D on Figure 23, significantly decreasing the horsepower. The control panel will monitor the system air pressure and if it decreases below a low pressure set point, shown by point E on Figure 23, the inlet valve will open and the unloading valve will close in order to meet the demand.

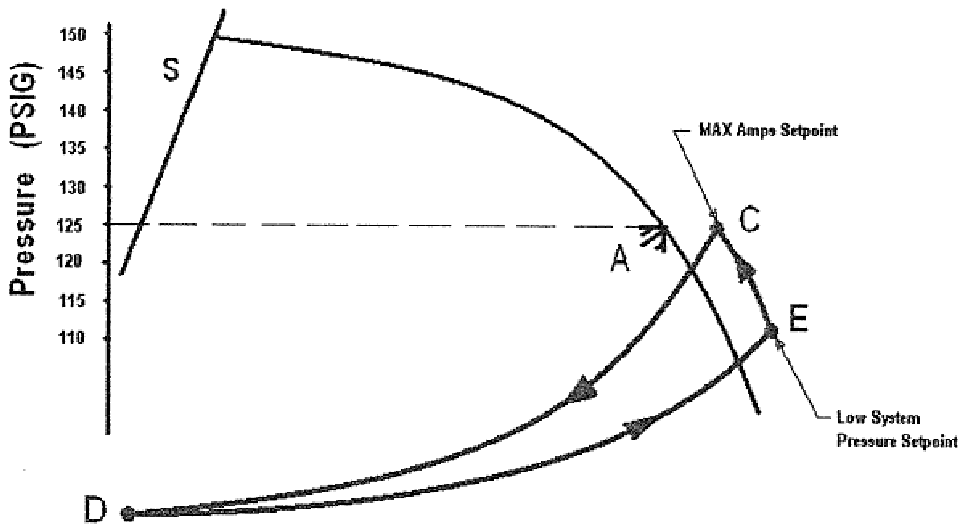

Figure 23. Intermittent Mode Compressor Control Curve

Given that intermittent mode only covers high plant demand, there is an additional control mode known as auto dual control. This mode is a combination of suction throttle and intermittent mode. Similar to suction throttle as the plant demand decreases and the UCV is maintaining the discharge pressure, the IGV will start to gradually close and reduce the flow and therefore power until the compressor is operating at the min amp set point. At this condition if the demand is further decreased and the unloading valve reaches a certain % open, the compressor will operate as it does in intermittent mode. A control curve for auto dual control is shown below in Figure 24.

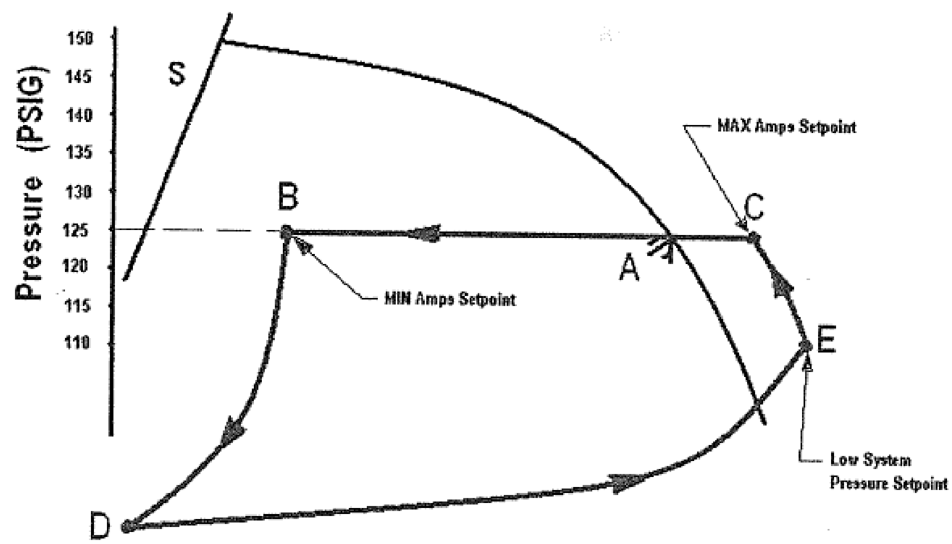

Figure 24. Auto Dual Compressor Control Curve

A PURCHASER'S PERSPECTIVE

Centrifugal compressors are commonly used to supply compressed feed air for cryogenic nitrogen separation systems. In these systems, it is desired to minimize overall capital cost and power consumption. Compressor selection and power consumption are influenced by the following:

- The desired nitrogen flow and flow pattern
- Plant operating pressure
- Nitrogen purity
- Ambient conditions (pressure, temperature, relative humidity)
- Coolant temperature

The air flow and pressure required from the compressor are determined from the nitrogen flow, pressure, and purity required by the user. For example, if the user requires a higher nitrogen pressure, the air pressure required from the compressor may be higher as well. If higher nitrogen purity is required, the air flow required will also be higher.

Typically, the nitrogen pressure and purity required are fixed. However, the nitrogen user may require variable flow. Further, there may be small variations in air flow due to the air separation process. For example, unit operations that involved packed bed or heat exchanger pass switching will produce air flow and pressure variations during transitions. If the air compressor flow rate is fixed, the average power consumption will be less than optimal when the excess flow is vented versus being used by the separation process. A hypothetical nitrogen use pattern is shown in Figure 25. In this example, if the air compressor is designed to track with the nitrogen flow demand, the power consumption will be approximately 85 percent of the power that would be consumed if the air flow were fixed.

Ambient conditions will also vary for a given nitrogen user. Ambient pressure will be relatively fixed, but ambient temperature and humidity will vary from night to day as well as seasonally. For example, it is not unusual for a location to have an average low temperature of 0°F and an average high temperature of 95°F with the humidity level varying from 0 percent to 100 percent. Compressors designed for higher ambient temperature and/or humidity level will consume more power than necessary at lower ambient temperatures and humidity levels. The compressor will have to produce pressurized air at the desired flow and pressure for at least a portion of the observed ambient conditions. Typically, the air compressor will be specified with emphasis on design conditions, which are based on user requirements. Design conditions will vary from user to user.

It may also be desirable for a given compressor to be used for a wide range of nitrogen users. In that instance, the nitrogen pressure and purity required will vary from user to user. Further, ambient pressure will vary from user to user in addition to ambient temperature and humidity. Some locations may be at high elevation with low ambient pressures (for example, ambient pressure at sea level will typically be 14.7 psia, while ambient pressure at 5000 feet will be 12.2 psia). Compressors designed for a higher elevation will have lower capacity/higher power consumption than compressors designed for lower elevation.

Compressor cooling system (intercooler, aftercooler, oil cooler) design will impact the compressor selection and power. It must consider the following:

- Coolant temperature
- Coolant type
- Required air temperature from the aftercooler

Single use coolants, such as cooling water, are typically cooler than recycled coolants, such as glycol/water. However, single use coolants may contain high levels of contaminants (for example, calcium or silt). For such systems, the compressor coolers must be designed to consider the potential for a high degree of fouling. Recycled coolants, such as glycol/water mixtures, typically contain very low contaminant levels.

Maximum air temperature at the compressor discharge is typically limited by downstream processes. The aftercooler must be designed with an approach to the coolant temperature that produces an acceptable discharge temperature.

Coolant selection must be examined on a case-by-case basis to determine whether single use coolants or recycled coolants are best able to meet the downstream process limitations at minimized power.

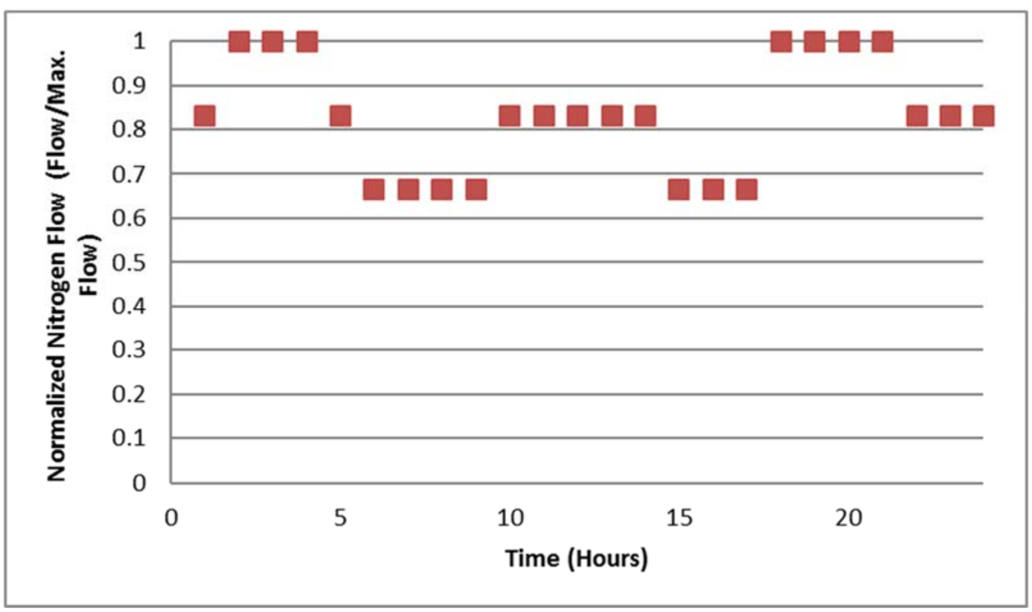

Figure 25. Hypothetical Nitrogen Use Pattern

CONCLUSIONS

Centrifugal air compressors are well suited for constant pressure applications that require a range of flow rates be covered in order to meet variable system demand. As this tutorial has shown, a compressor's sizing and selection in order to meet this demand is heavily dependent on the requirements and site ambient conditions specified. Based on this it is important that these conditions are specified in a realistic manner to ensure that the compressor can be selected to be the most efficient possible at the actual operating conditions it will be utilized at. With these conditions well-realized the compressor can be selected appropriately and through the use of the IGV and UCV coupled with the correct compressor control mode, the air compressor will be able to cover the full operating envelope desired.

NOMENCLATURE

D	= Impeller Diameter	(in)
Eff	= Motor Efficiency	(%)
g	= Earth Gravitational Acceleration	(ft^2/s)
GHP	= Stage Gas Power	(HP)
H_{ad}	= Adiabatic Head	(ft)
k	= Specific Heat Ratio	(k)
$\dot{m}$	= Mass Flow	(lb_m/min)
MHP	= Motor Horsepower	(HP)
N	= Impeller Rotational Speed	(RPM)
n	= Polytropic Exponent	(-)
P_{in}	= Stage Inlet Pressure	(psia)
P_s	= Saturated Vapor Pressure	(psia)
PF	= Motor Power Factor	(-)
Q	= Stage Inlet Volume Flow	(ICFM)
R_a	= Specific Gas Constant, Air	(ft-lb_f/lb_m-˚R)
R_m	= Specific Gas Constant, Mixture	(ft-lb_f/lb_m-˚R)
R_v	= Specific Gas Constant, Vapor	(ft-lb_f/lb_m-˚R)
RH	= Relative Humidity	(%)
SH	= Specific Humidity	(-)
T_{in}	= Stage Inlet Temperature	(˚F)
U	= Impeller Tip Speed	(ft/s)
v_{in}	= Stage Inlet Specific Volume	(ft^3/lb_m)
γ	= Work Coefficient	(-)
η	= Adiabatic Efficiency	(-)
φ	= Flow Coefficient	(-)
ψ	= Head/Pressure Coefficient	(-)
IGV	= Inlet Guide Vane	
UCV	= Unloading Control Valve	

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