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END USER'S APPROACH TO CALCULATING COMPRESSOR PERFORMANCE

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ABSTRACT

Oil and Gas industry end users have generally approached the topic of centrifugal compressor performance estimation with little consideration to the effect of process gas conditions. This is customary since end users depend mostly on OEM methodologies. This tutorial argues that performance curves are dependent on process gas conditions. The tutorial touches on key gas properties and how they are used in compressor performance calculations. An explanation of the effect of process gas conditions is discussed, and how to develop and make use of invariant performance curves and therefore, estimation results are improved.

Finally, dimensionless analysis laws are applied using an end user developed performance calculation tool. The tool has been successfully implemented in plant conditions with proven results.

INTRODUCTION

Disregarding the effect of changes in process gas and (or) suction conditions can lead end users to misinterpret normal operation as deterioration/improvement in the health of the compressor. Understanding the changes in gas properties and the effect that they have on the compressor performance map is key in predicting the compressor output.

Original Equipment Manufacturer (OEM) performance curves, in general, are provided with sparse mapping. There are many missing operating points that the end user wishes to assess. This has led to the end user to develop its own invariant compressor performance maps, which are derived from the OEM standard maps. Curves are corrected for process gas conditions allowing for a reliable approach to monitoring compressor performance.

A further justification for adopting an in-house performance analysis tool is to facilitate a seamless data connection to the plant historical database. In doing so, enough data points for continuous evaluation over time is possible.

COMMON PERFROMANCE CALCULATION METHODS

Key parameter that concern end users is whether their process compressors are developing the required head at a given flow rate and expected efficiency. This ensures the power supplied by the driver is sufficient to sustain required production. A common practice is to calculate these parameters using actual changes to the gas condition from suction to discharge, as shown in Equations (1) – (3).

Polytropic Head

$$y_p = RZT_{in}\frac{n}{n-1}\left[\left(\frac{P_{out}}{P_{in}}\right)^{\frac{n-1}{n}} - 1\right] \qquad (1)$$

Polytropic Efficiency

$$\eta_p = \left[\frac{ln\left({P_{out}}/{P_{in}}\right)}{ln\left({T_{out}}/{T_{in}}\right)}\right] \times \frac{(k-1)}{k} \qquad (2)$$

Gas Brake Power

$$W_{break} = \frac{\left(y_p\ \dot{Q}_{in}\ \rho_{in}\right)}{\eta_p} \qquad (3)$$

Most industry end users utilise these parameters through different methods, to gauge the health of their process compressors. Some of the common methods and shortcomings that accompany them are described below.

Trending Performance Parameters

One method that has been utilised is to plot a timeline percentage change of actual performance parameters compared to an established baseline. This in effect assumes that the compressor will mostly operate at the same speed and flow, and compress the same gas at the same conditions. Consistent changes in performance parameters are thus attributed to compressor or driver health deterioration.

Fan Laws

Fan Laws are an application of the concept of similitude [1], where flow, head, and power are functions of rotor speed, as shown in Equations (4) – (6):

$$\dot{Q} = f(N) \qquad (4)$$
$$y = f(N^2) \qquad (5)$$
$$W_{brake} = f(N^3) \qquad (6)$$

These equations are used to predict the percent change in performance curves as a percentage change in rotor speed in comparison to the selected design point. The new predicted curve becomes the baseline to compare actual performance against. Fan Laws are indeed applicable to single stage, low pressure ratio impellers, and are used in the test bench of Type 2 ASME PTC-10 test [2]. However, for end users of multistage centrifugal compressors, with access to overall compressor performance curves as a sole reference curve for predicting performance, this method becomes disproportionally inaccurate. Use of the Fan Laws for this kind of application disregards the volume ratio effect [1], as described later.

Head and Flow Coefficient for the Design Curve

Converting the head/flow and efficiency/flow curves provided by the OEM at the design condition and speed is a way of analysing performance non-dimensionally, as described in Equations *(7)* and *(8)*. This method uses the new dimensionless curves as the predicted curves to compare against actual non-dimensional performance parameters. This in effect removes the effect of speed from the analysis. However, it fails to recognise that the single design curve, even though dimensionless, does not cover varied inlet conditions, gas composition and speeds. As a validation, a comparison of dimensionless curves based varied inlet conditions, gas composition and speeds will reveal that there are multiple unequal curves with varying shapes. Similar to the Fan Laws, this method ignores the volume ratio effect.

Flow Coefficient

$$\varphi = \frac{\dot{Q}_{in}}{\pi\left({D^2}/{4}\right)u_t} \qquad (7)$$

Polytropic Head Coefficient

$$\psi_p = \frac{y_p}{{u_t^2}/{2}} \qquad (8)$$

OEM PERFORMANCE MAPS

For variable speed centrifugal compressors, API 617 [3] states that any specified operating points shall be noted within the envelope of the performance curve predicted. It also states that the effect of specified inlet pressures, temperatures, and molecular weights shall be indicated. As a result, OEMs provide several performance curves to meet the project expected operation points. These curves typically come, for varying flow rates and speed, in the form of:

- Polytropic head
- Polytropic efficiency
- Pressure ratio
- Power
- Discharge temperature
- Discharge pressure

Effect of Inlet Conditions

The effect of suction pressures, temperatures, and molecular weight being referred to by API 617, is the possible variation in polytropic head for a given volumetric flow rate and speed. This manifests as a physical shift of the constant speed curves and alteration of the allowable operating window. In turn, this highlights that the predicted OEM performance map at the selected operating conditions is not applicable for other operating conditions. The changes in the operating window for a selected multistage compressor is shown in Figure 1 – Figure 3, as an effect of changing suction temperature, molecular weight, and suction pressure, respectively. This creates an advantage for end users to have performance curves that are invariant to process conditions.

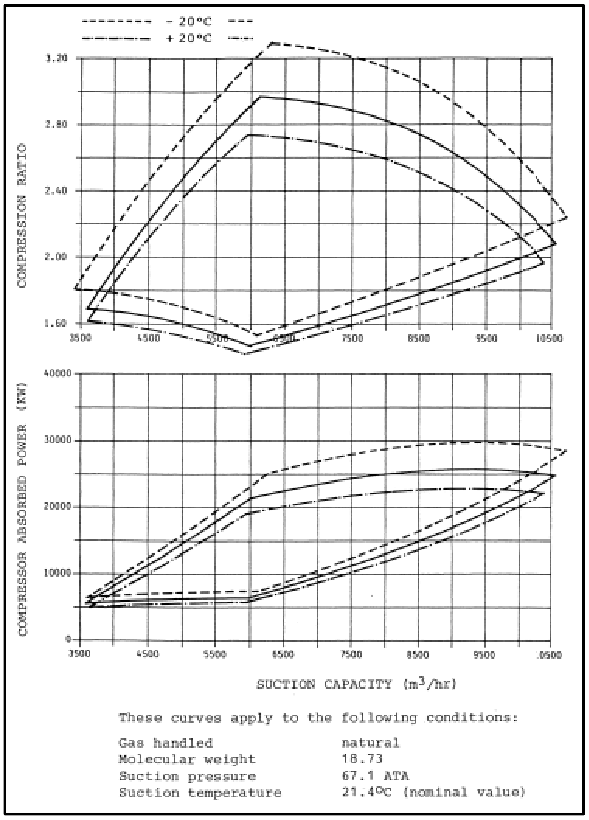

Figure 1 – Effect of changing suction temperature on the predicted compressor operating window. (Courtesy GE)

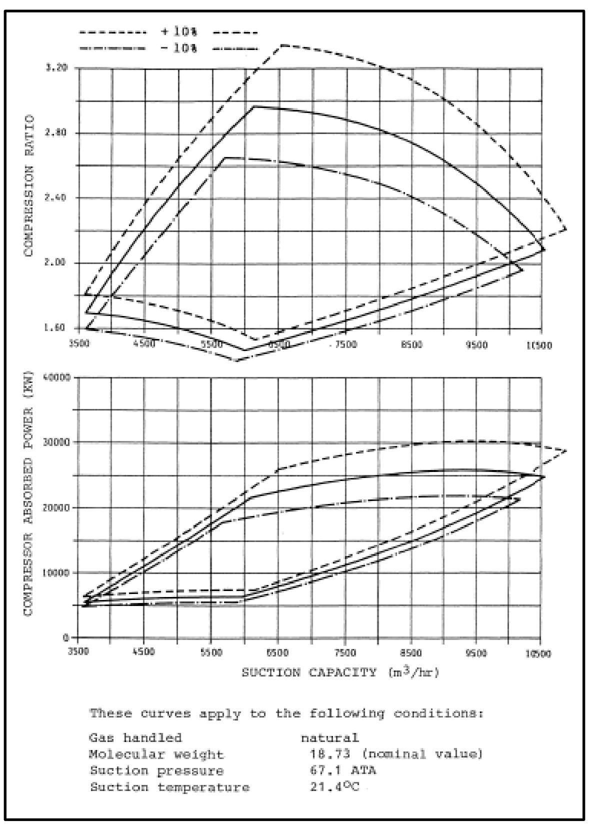

Figure 2 – Effect of changing molecular weight on the predicted compressor operating window. (Courtesy GE)

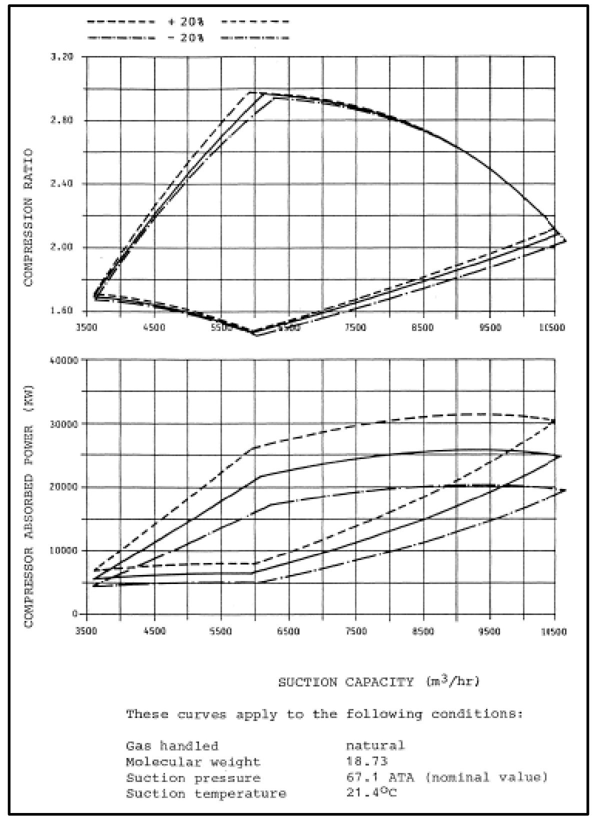

Figure 3 – Effect of changing suction pressure on the predicted compressor operating window. (Courtesy GE)

INVARIANT COMPRESSOR PERFORMANCE MAPS

Ludtke [4] offers a practical insight to the physical mechanism which causes the performance curves to vary with changes in suction conditions and (or) gas composition. He describes it as inter-stage mismatching where each subsequent compressor stage is operating further away from its design point, from suction to discharge. This is a result of the changing volume ratio at each compressor stage, compared to design. This effect becomes pronounced for a compressible gas, which generally corresponds to impeller tip-speed Mach numbers (machine Mach number) above 0.4 and more than two stages. Under these conditions, any change in the parameters which can affect the machine Mach number (suction temperature and gas composition – see Equation *(9)*) will result in a change in polytropic head for a given volumetric flow rate and shaft speed.

$$M_{ut} = \frac{u_t}{a_{in}} = \frac{u_t}{\sqrt{kZ_{in}RT_{in}}} \qquad (9)$$

Non Dimensional Analysis

As described earlier, the objective is to derive non-dimensional parameters which are practically invariant to the effects of compressibility. Batson [5] showed that for a real compressible gas the dependent variables, Polytropic head, y_p, and Polytropic efficiency, η_p, can be expressed as a function of six independent variables to adequately describe the performance of a centrifugal compressor as follows:

$$y_p, \eta_p = f\{D, N, \dot{Q}, a_{in}, \rho_{in}, \mu\} \qquad (10)$$

where a_{in} and ρ_{in} are chosen at the compressor suction because they vary through the machine.

Application of the Buckingham Π theorem states that there will be total of four dimensionless groups, excluding η_p.

$$\Pi_1, \eta_p = f\{\Pi_2, \Pi_3, \Pi_4\} \quad (11)$$

Selecting D, N and ρ_{in} as the repeating variables, the dimensionless groups can be written as:

Mach number corrected head factor:

$$\Pi_1 = \frac{y_p}{a_{in}^2} \quad (12)$$

Machine Mach Number (M_{ut}):

$$\Pi_2 = \frac{ND}{a_{in}} \quad (13)$$

Mach number corrected flow factor:

$$\Pi_3 = \frac{\dot{Q}}{a_{in}D^2} \quad (14)$$

Machine Reynolds number:

$$\Pi_4 = \frac{\rho_{in}ND^2}{\mu} \quad (15)$$

Assuming the change in machine Reynolds number, Π_4, is small from operating point to operating point, it can be neglected to yield the final non-dimensional relationship as follows:

$$\Pi_1, \eta_p = f\{\Pi_2, \Pi_3\} \quad (16)$$

A new performance map can therefore, be readily derived from the OEM provided maps. Furthermore, its range can be maximised by applying the analysis to the full set of standard performance curves provided, for all suction conditions and gases. Since the three non-dimensional groups are corrected for suction speed of sound, the Mach number effects on compressibility are adequately captured. The derived performance map is invariant to changes in suction conditions and gas composition, within the range of the constant machine Mach number, Π_2, characteristics. Real gas behavior is captured by applying the appropriate equation of state in the analysis. An example of the derived non-dimensional performance map is shown in Figure 4.

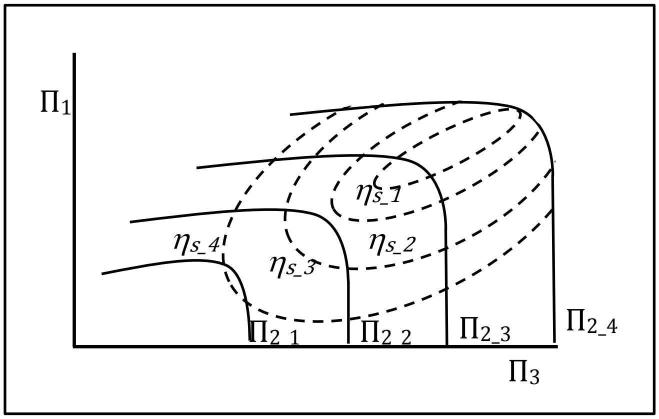

Figure 4 – Typical non-dimensional invariant compressor map

GAS PROPERTIES

The preceding sections highlighted the need for a clear understanding of gas properties and how they influence compressor performance.

Industrial gas processes are based on different molecular compositions that may change over time. The thermodynamic properties of these gas mixtures change as they go from one stage of processing to the next. The key elements in polytropic compression that concern machinery are: compressibility factor, Z, and polytropic exponent, n. Equations of state, (EOS) are typically used to determine these properties. Pseudo-critical properties of the gas mixture and gas specific heat become instrumental in many of these calculations.

Polytropic process

An actual compression process is commonly analyzed using a polytropic process [6]. This is described as a succession of infinitesimal isentropic process steps, with each step separated by a frictional heat loss h_f. All frictional heat loss is excluded from the polytropic head definition which actually makes it lossless. However, it is not reversible due to the entropy rise encountered from the total frictional heat loss when compressing between states 1 and 2 in Figure 5. One can explain the enthalpy change in the actual and polytropic compression processes from P_1 to P_2 in Figure 5, using Equations *(17)* and *(18)* respectively, keeping in mind that a polytropic process follows Equation *(19)* for a perfect gas.

$$\Delta h = h_2 - h_1 \quad (17)$$

$$y_p = \lim_{x\to\infty} \sum_{i=1}^{x} y_{si} \quad (18)$$

$$PV^n = Constant \quad (19)$$

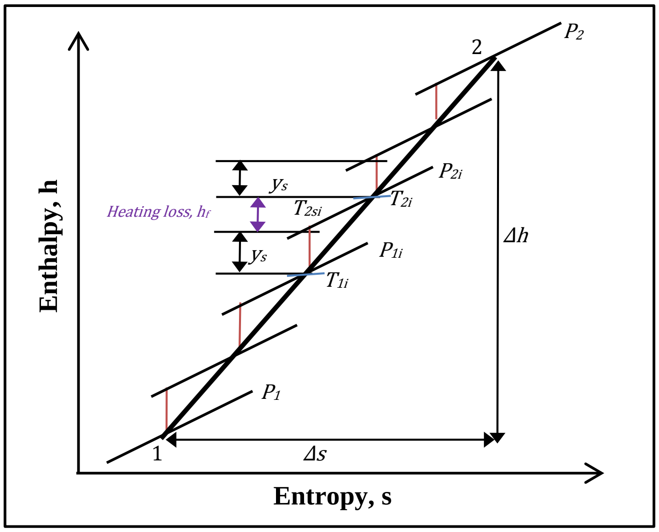

Figure 5 – Explaining the actual compression process: changes in Enthalpy, h, and Entropy, s during compression

Equations of State (EOS)

Realising the differences between real gases and ideal gases puts forward the need to calculate the compressibility factor and polytropic exponent to be able to calculate the Polytropic head and efficiency across a compressor. The challenging element is that these parameters are functions of temperature, pressure, and gas composition.

Equations of State (EOS) have been developed through the years to reveal the changes in gas properties at different conditions. Among the properties estimated from an EOS are:

- Densities (vapor and liquid),
- Vapor pressures,
- Critical pressures and temperatures,
- Vapor-Liquid equilibrium
- Thermodynamic properties (Δh, Δs, ΔG, ΔA).

It is through an EOS that one can accurately predict the required gas properties. However, each EOS has its own limitations and choosing the one that fits the gas conditions becomes critical. Sandberg [7] highlights the effect of selecting the EOS in calculating compressor performance, namely the differences in Z, as shown in Figure 6. Since all EOS correlations were derived from experimental data, it should be expected that different fluid properties will be calculated from each EOS for identical input conditions. Appendix A summarises the calculation of Z for some of the most common EOSs. A review of these equations shows how constants have been added from Vander Waals to Benedict-Webb-Rubin-Starling (BWRS) to increase the accuracy of matching the behaviour of the gas.

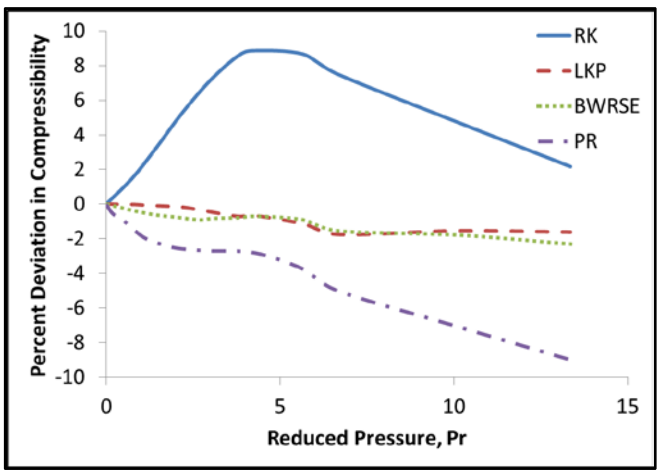

Figure 6 – Changes in compressibility factor for several EOS at Reduced Temperature of 1.63 and Accentricitity factor of 0.036 (reproduced by permission from the Turbomachinery Symposium) **[7]**

Compressibility Factor Correlations

Alternative to the EOSs discussed, several simpler correlations have been experimentally developed to calculate Z. Generalised compressibility diagrams, such as Nelson-Obert [8] diagram, were developed for pure gases as functions of pressures and temperatures.

A more accurate correlation is Standing and Katz [9], which is widely used by petroleum engineers. It has been proven to accurately predict hydrocarbon gas behaviors [10], with certain limitations when compared to other EOS's. Similar to EOSs, knowledge of critical pressure and temperature of the compressed gas is required to be able to compute *Z*. The correlation is a function of reduced pressure, P_r, and reduced temperature, T_r, where pseudocritical properties can be used for hydrocarbon gas mixtures.

Several equations have been developed to mathematically model these correlations, to name a few: Hall-Yarborough, Beggs and Brill, and Dranchuk and Abou-Kassem Equation of State [8].

Specific Heat Capacity

Knowledge of specific heat of the compressed gas mixtures is needed for the purposes of using Equation *(1)*, in estimating the Polytropic exponent, as detailed in Equations *(20)* and *(21)*.

$$k = \frac{C_p^0}{C_v^0} = \frac{MC_p}{MC_p - R} \quad (20)$$

$$\frac{n-1}{n} = \frac{k-1}{k\eta_p} \quad (21)$$

Once again, this parameter is a function of pressure, temperature, and gas composition. The Shomate Equation, shown below, is one of the most accurate methods in estimating the heat capacity of a gas [11]. Coefficients used in the equation are available for a wide range of components through the NIST Web Book [11].

$$C_p^0 = A + BT + CT^2 + DT^3 + \frac{E}{T^2} \quad (22)$$

It should be stressed, however, that Equation *(21)* defines the polytropic exponent for a perfect gas, and is used in this tutorial as an estimate. For a real gas, the polytropic volume exponent n_v, would need to be defined at a constant efficiency, as described in Equation *(23)*.

$$n = n_v = -\frac{v}{P}\left(\frac{\partial P}{\partial v}\right)_\eta \quad (23)$$

USER'S METHODOLOGY IN CALCULATING PERFORMANCE

Appreciating the concepts presented earlier helps end-users in developing their own compressor performance monitoring strategy. This section details a practical methodology that is used in building a performance calculation tool. The approach looks at defining a predictive curve to meet the operating condition of the unit at the calculated machine Mach number by interpolating between OEM provided curves. This is similar in concept to ASME PTC-10 Type 2 [12] test, where the OEM tries to match an available gas to meet the specified gas predicted characteristic curves using an allowable deviation in machine Mach number.

To be able to proceed with the methodology, the following parameters are required to be readily available, using the concepts discussed earlier.

Site Collected Parameters:	T_{in}, P_{in}, T_{out}, P_{out}, $\dot{Q}$, M
Calculated Parameters:	T_r, P_r, Z, C_p^0, κ, ρ_{in}

In the interest of usability with a spreadsheet software package directly connected to the site's historian, a Standing and Katz correlation based equation is applied to estimate *Z*. The Shomate Equations are used to estimate C_p. Calculation errors for the performance parameters, polytropic head and efficiency, are in the range of 3% when compared with other EOS based engineering software for the range of process gases at site. This error range can be different for other end users, depending on the processed gas composition and conditions.

Actual Performance

Performance parameters are calculated from Equations *(1)* and *(2)*. This then becomes the actual performance of a compressor that needs to be compared against a predicted performance. These parameters are calculated directly from site conditions at suction and discharge of the compressor.

Predicted Performance

As explained earlier, different process inlet conditions can affect the predicted performance of the compressor. To reflect the changes, OEMs provide end users with sets of compressor overall performance curves at agreed inlet conditions and speeds. However, on most occasions, end users operate the compressors at conditions different from these curves.

From the earlier discussion on non-dimensional analysis, a set of invariant performance curves are developed in the form of Equation 16. Machine Mach number, M_{ut}, is calculated for each of the conditions and speeds provided by the OEM. A number of performance curves are selected to represent a wide range of machine Mach numbers. Invariant non-dimensional curves are derived from the performance curves, as shown in Figure 7. This becomes the background database that is used for deducing the predicted performance curve at site conditions.

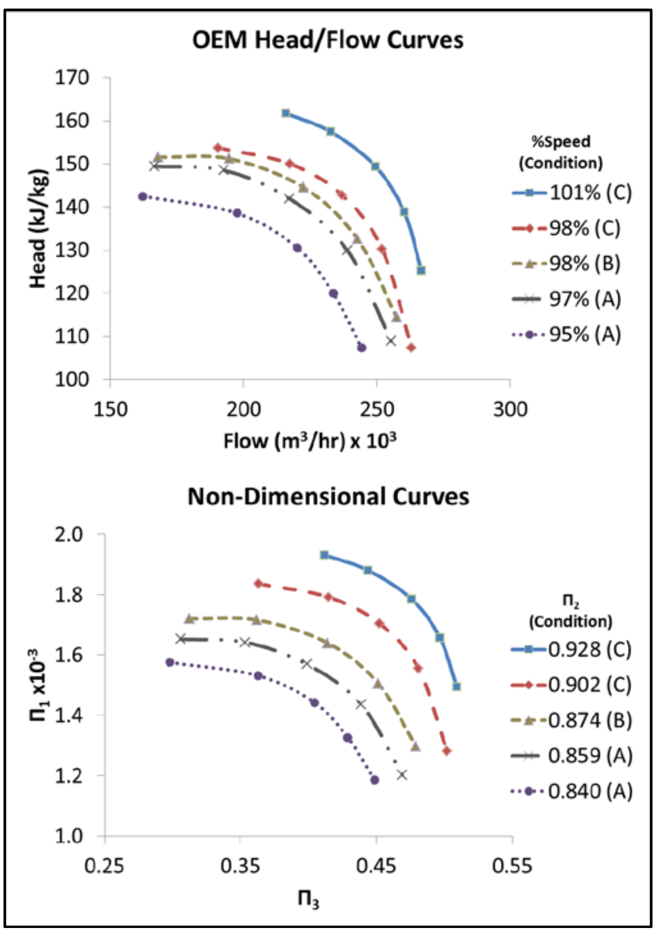

Figure 7 – Converting OEM head/flow curves at different speeds and conditions to dimensionless curves at represented Mach number

Using process gas conditions, machine Mach number is calculated for the existing condition, and the appropriate non-dimensional curve can be obtained through interpolation from the defined database.

To be able to calculate the predicted polytropic head, actual field volumetric flow rate is used to interpolate across the non-dimensional site corrected curve, and the predicted head factor is calculated. Predicted polytropic head is then calculated from the non-dimensional parameter (Equation *(12)*)

Relative Performance

Once predicted and actual performance parameters are calculated, the relative difference between the two parameters is then used as the key health indicator. Accordingly, process seasonality effects and changes in load requirements are factored out, as shown in Figure 8. The only effect to cause a relative change between actual and predicted would be a change in compressor health.

As estimated gas parameters from the EOS are likely to be different from those used by OEM's prediction, performance parameters will typically have a baseline deviation and will not exactly coincide with the predicted curves. Hence, the use of relative performance will insure end users to ignore the effect of this baseline deviation and focus on changes in this deviation as an indicator of performance changes. Nevertheless, this error can be reduced by developing new sets of characteristic curves based on the end user's EOS and parameter correlations.

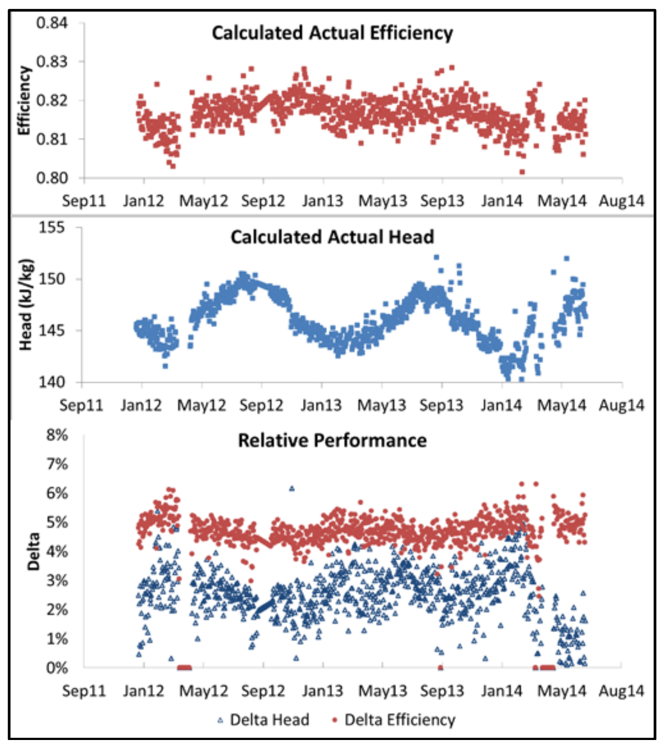

Figure 8 – Seasonality effect reduced through relative performance

Polytropic head and polytropic efficiency versus volumetric flow rate curves can be reversed calculated using the OEM provided pressure and temperature rise versus volumetric flow rate diagrams, if available. Equations *(1)* and *(2)* are used to calculate polytropic head and polytropic efficiency for each point. The same process explained above can be followed in converting the user developed characteristic curves to dimensionless curves based on machine Mach number.

Verification Methodology

As discussed earlier, machine Mach number can be used as the single selector component to define the reference dimensionless performance curve. Figures 9 and 10 show the application of this theory on a selected compressor in natural gas service. The dimensionless curves of two different conditions, A and B, having similar machine Mach numbers, lie on top of each other. Conditions A and B are shown in Table 1.

Table 1 - Process conditions for the curves displayed in Figures 9&10

Condition	A	B
Suction Pressure(Bar A)	9.3	8.8
Suction Temperature (C)	58	38
Speed (%)	88%	100%
Molecular Weight	37.6	29.3
Machine Mach Number (Π_2)	0.689	0.69

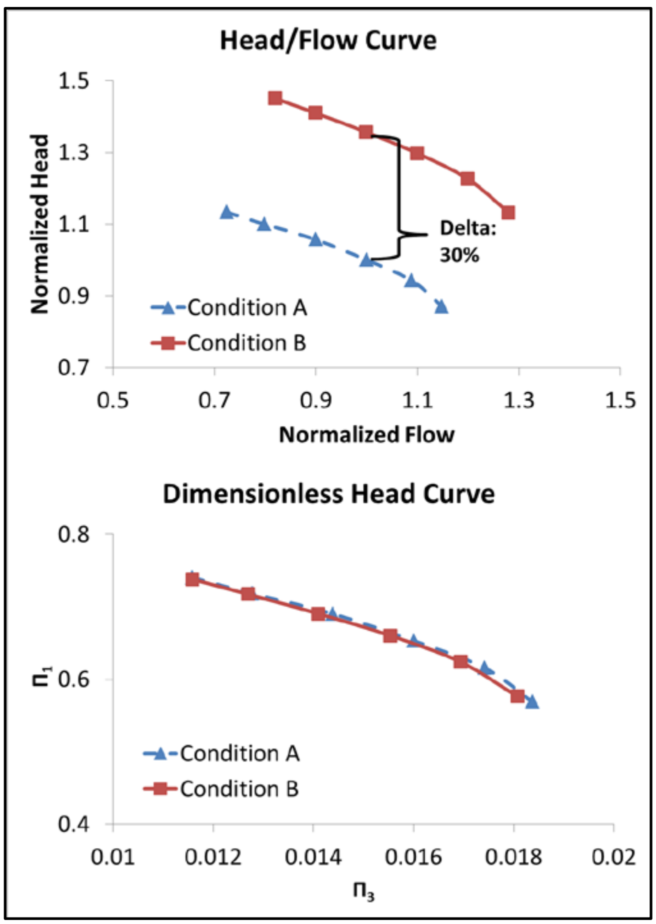

Figure 9 – Comparing dimensionless performance curves of two different conditions having the same machine Mach number (Head)

CONCLUSION

This tutorial sets out to demonstrate the strong effect of process gas conditions upon the performance of centrifugal compressors. Other end users are encouraged to take note of this effect and account for it through proper performance modelling. One such method, which has been successfully applied by an end user is presented herein.

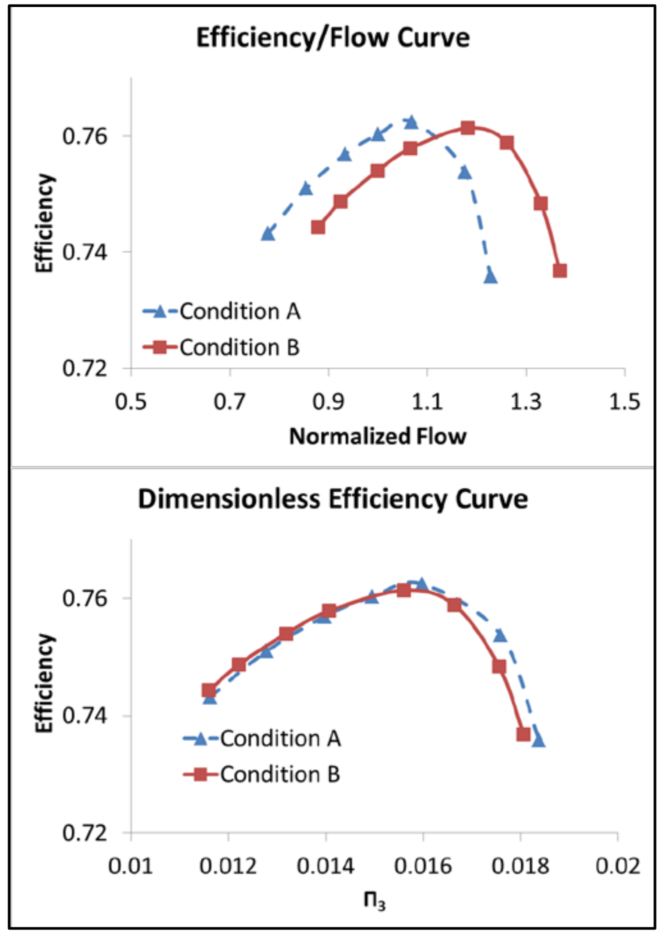

Figure 10 – Comparing dimensionless performance curves of two different conditions having the same machine Mach number (Efficiency)

NOMENCLATURE

a	Stagnation sonic velocity	(m/s)
C_p	Specific heat capacity at constant pressure	(J/molK)
D	First stage impeller exit diameter	(m)
h	Stagnation enthalpy	(kJ/kg)
M	Molecular weight	mol/g
M_{ut}	Machine Mach number	
n	Polytropic exponent	
N	Shaft speed	RPM
P	Stagnation pressure	(bar a)
P_c	Critical pressure	(bar a)
P_r	Reduced pressure (P/P_c)	
$\dot{Q}$	Volumetric flow rate	(m^3/s)
R	Specific gas constant	(kJ/kgK)
T	Stagnation temperature	(K)
T_c	Critical temperature	(K)
T_r	Reduced temperature (T/T_c)	
u_t	Tangential velocity at 1st stage exit diameter	(m/s)
W	Power	(kW)
y	Head	(kJ/kg)
Z	Compressibility factor	
η	Efficiency	

κ	Isentropic exponent	
μ	Dynamic viscosity	(cP)
v	Specific volume	(m^3/kg)
Π	Buckingham PI non-dimensional group	
ρ	Stagnation density	(kg/m^3)
φ	Flow coefficient	
ψ	Head coefficient	

Subscripts

in	Compressor suction
out	Compressor discharge
p	Polytropic
s	Isentropic

APPENDIX A

Table 2 summarises the calculation of Z for some of the most common EOS for pure substances. It should be noted that mixing rules are different for each EOS for gas mixtures.

Table 2 - Common EOS for calculating Z

EOS	Pure Component Coefficients
Van der Waals $\mathbf{Z} = \frac{\boldsymbol{v}}{\boldsymbol{v}-\boldsymbol{b}} - \frac{\boldsymbol{a}}{\boldsymbol{RTv}}$	$a = \frac{27(RT_c)^2}{64P_c}$ $b = \frac{RT_c}{8P_c}$
Redlich Kwong Soave $\boldsymbol{Z} = \frac{\boldsymbol{v}}{\boldsymbol{v}-\boldsymbol{b}} - \frac{\boldsymbol{a}}{\boldsymbol{RT}(\boldsymbol{v}+\boldsymbol{b})}$	$a = \frac{0.42747\alpha R^2 T_c^2}{P_c}$ $b = \frac{0.08664RT_c}{P_c}$ $\alpha = \left[1 + m\left(1 - \sqrt{T/T_c}\right)\right]^2$ $m = 0.48 + 1.574\omega - 0.176\omega^2$
Peng Robinson $\boldsymbol{Z} = \frac{\boldsymbol{v}}{\boldsymbol{v}-\boldsymbol{b}} - \frac{\boldsymbol{a}(\boldsymbol{v}/\boldsymbol{RT})}{\boldsymbol{v}(\boldsymbol{v}+\boldsymbol{b}) + \boldsymbol{b}(\boldsymbol{v}-\boldsymbol{b})}$	$a = \frac{0.4572R^2T_C^2\alpha}{P_c}$ $b = \frac{0.07780RT_c}{P_c}$ $\alpha = \left[1 + \chi(1 - \sqrt{T/T_c}\right]^2$ $\chi = 0.37464 + 1.54226\omega - 0.26992\omega^2$
Benedict Webb Rubin Starling (BWRS) $Z = 1 + \frac{B}{v} + \frac{C}{v^2} + \frac{D}{v^5} + \left(\frac{C'}{v^2}\right)\left(1 + \frac{\gamma}{v^2}\right)e^{\left(-\gamma/v^2\right)}$	

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