FEATURE

Centrifugal Compressor Performance Estimation

by Cheah Cang To

This article first provides some introductory materials on compressors and presents the method developed by the author specifically for multi-stage compressor performance estimation based on data obtained from selected applications in Oil & Gas industries.

Nomenclature

K	=	Isentropic exponent or ratio of specific heat	V	=	Volume
Cp	=	Specific heat capacity at constant pressure	n	=	Polytropic exponent
Cv	=	Specific heat capacity at constant volume	C	=	Constant
T	=	Absolute temperature	Hp	=	Polytropic head
v	=	Specific volume	nv	=	Polytropic volume exponent
Hp	=	Polytropic head	nt	=	Polytropic temperature exponent
ugc	=	Universal gas constant	ηp	=	Polytropic efficiency
R	=	Specific gas constant	ueq	=	Equivalent impeller tip speed
T	=	Temperature	Deq	=	Equivalent impeller diameter
Tc	=	Critical temperature	s	=	Work input factor
Tb	=	Boiling temperature	ψp	=	Polytropic head coefficient
Tbr	=	Reduced boiling temperature	$\dot{Q}$	=	Inlet volume flow rate
Tr	=	Reduced temperature			
P	=	Pressure	Cp	=	Heat capacity at constant pressure
Pc	=	Critical temperature	Cv	=	Heat capacity at constant volume
Pr	=	Reduced pressure	$\dot{m}$	=	mass flow rate
ω	=	Pitzer acentric factor			
vr	=	Reduced volume	φ	=	Flow coefficient
ηp	=	Polytropic efficiency			
Z	=	Compressibility factor			

Note : subscripts 1 and 2 used in the content denote conditions at suction and discharge flanges, respectively.

SECTION 1: INTRODUCTION TO COMPRESSOR

A compressor is rotating machinery that increases the pressure of compressible fluid by reducing its volume. Compressors exist in numerous forms. The exact configuration depends on the specific application. In general, the different types of compressors can be categorised into two main groups, namely:

a. Positive displacement: Positive displacement machines work by mechanically reducing the volume of the working fluid. These machines can be divided into two sub-groups: Rotary and Reciprocating.

b. Dynamic: Dynamic machines work by mechanically changing the velocity of working fluid, which can be divided into Axial and Centrifugal.

Fig. 1 summarises the compressor family in term of compression ratio and actual inlet volume flow rate. Flow and compression ratio limits are indicated in Fig. 1, it can be seen that positive displacement machines are not suitable for handling inlet volume flow higher than 1700 m^3/hr (approximately).

It should be noted that the left end of dynamic machine does not reach zero flow. The threshold of minimum inlet volume flow rate is defined by the surge limit of dynamic machine.

This article focuses on the application of multi-stage centrifugal compression system, which is commonly used on offshore platforms transporting natural gas from offshore facilities to onshore plants. A typical cross-sectional view of multi-stage centrifugal compressor can be seen in Fig. 2. A centrifugal stage consists of 3 basic components (Fig. 3): Rotor assembly (impeller), diffuser and return channel. Gas flows along shaft circumference towards impeller inlet eye, and kinetic energy is mechanically exerted by impeller blades. The gas then leaves the impeller and enters the

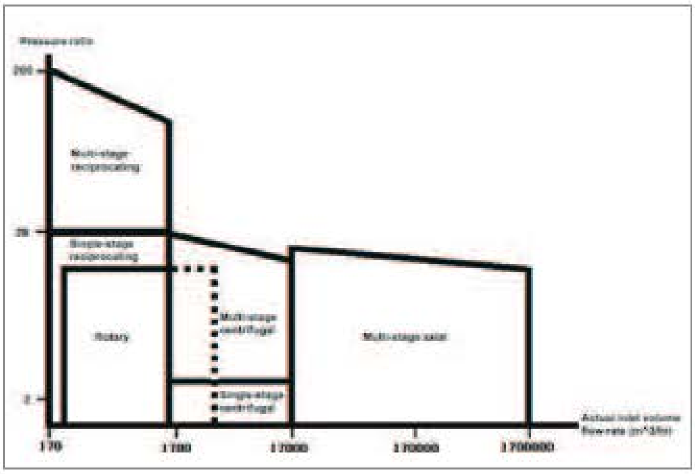

Fig. 1: Typical application ranges of compressor types in term of compression ratio with respect to inlet volume flowrate [6]

diffuser (static part of compressor), diffuser is used to reduce gas velocity (after leaving impeller) and increase the static pressure. Compressed gas from the diffuser is routed into the subsequent impeller via a return channel.

Fig. 2: Typical cross-sectional view of centrifugal compressor [6]

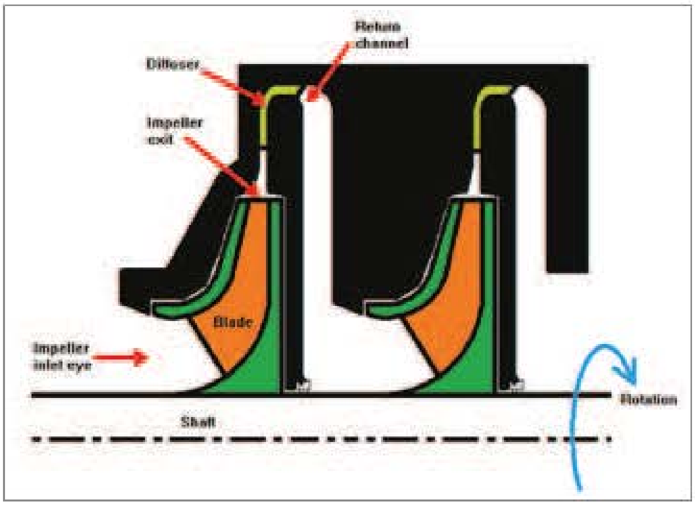

Fig. 3: Sketch of compressor stage (impeller, diffuser and return channel)

The compressor performance curve consist of plots of compressor discharge pressure (or polytropic head) at various constant rotational speed conditions (Fig. 5). Minimum and maximum allowable inlet volume flow rates at constant rotational speed are called "Surge" and "Choke" limits, respectively.

Centrifugal compressors are volumetric machinery, so when there is insufficient gas to replace what is being pushed forward (i.e. towards the left end of performance curve), discharge gas will flow backwards through the compressor towards the suction side. The gas flowing backwards increases the volume on the suction side and the compressor picks up and begins to push the gas forward again. This phenomenon is called "Surge". This occurs very quickly and the backward and forward gas flow causes rapid fluctuations in the flow, pressure and temperature of the compression system. Prolonged operation in surge mode can seriously damage the compressor as well as associated upstream and downstream equipment due to high vibrations in the compression loop. Surge can be prevented by implementing anti-surge system and control valve, which recycles discharged gas back to the suction flange in order to maintain minimum volume flow rate to the compressor.

Operating compressors in the high volume flow region (towards the right end of performance curve) is often referred to as "Choke". Choke occurs when the internal gas flow velocities are close to Mach 1, and generates a rapid increase in pressure losses. Prolonged operation in the choke limit should be avoided, because efficiency of compressor is very low.

Efficiency islands (of constant efficiency) are contours ranging from 74% to 82% in example provided in Fig. 5. The addition of efficiency islands on performance curves effectively provides a 3-dimensional topology to this 2-dimensional curve. With inlet conditions specified (e.g. suction pressure, suction temperature, inlet volume flow rate and gas compositions), compression power can therefore be estimated.

Compressors are often required to operate across a range of flows and pressures; the design point is typically landed at the peak efficiency region of performance curves.

Integrity of impellers may be lost (due to excessive shear forces between impeller shroud and blades) if maximum allowable tip speed is exceeded when the compressor operates above the maximum continuous speed.

Combining first and second law of thermodynamics as well as few dozens of algebraic steps, the theoretical compressor work (or compressor head) is derived as follow.

$$H_p = Z_1 R T_1 \frac{n_v}{n_v - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n_v - 1}{n_v}} - 1\right] \Rightarrow J/kg \qquad \text{Eq. 1 [1]}$$

Centrifugal compressors do not develop constant compression ratio at a given inlet volume flow rate; rather they develop a constant polytropic head at a defined inlet volume flow rate.

Re-arranging Eq. 1, compressor discharge pressure (P2) can be calculated.

$$P_2 = \exp\left[Ln\left(\frac{H_p(n_v-1)}{Z_1 R T_1 n_v}+1\right)\times\left(\frac{n_v}{n_v-1}\right)\right]\times P_1 \Rightarrow N/m^2 \quad \text{Eq. 2}$$

Compressibility factor (Z) is a measure of deviation of real gas behaviour from ideal gas behaviour. Gas behaved differently at a given temperature and pressure. However they behave similarly at temperatures and pressures normalized with respect to their critical temperatures and pressures, namely reduced temperature and reduced pressure, respectively.

$$T_r = \frac{T}{T_c} \quad \text{Eq. 3} \qquad\qquad P_r = \frac{P}{P_c} \quad \text{Eq. 4}$$

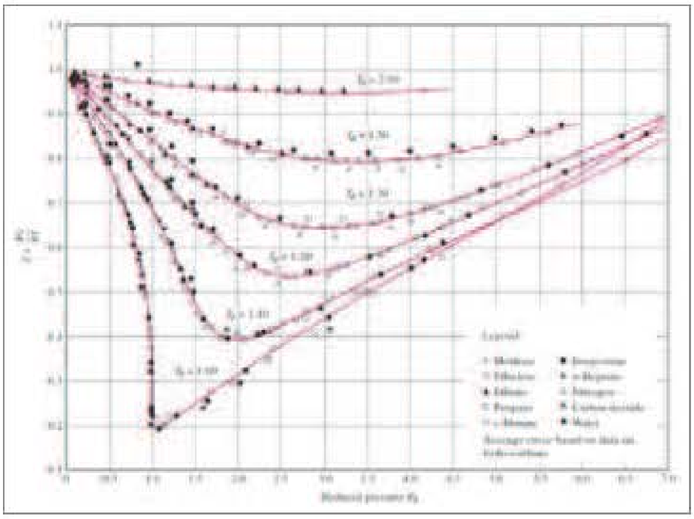

Fig. 4: Generalised compressibility factor chart derived from experimental data [5]

Compressibility factor (Z) is defined as the ratio of the actual volume of real gas to the volume estimated by the ideal gas law. Compressibility factor of real gas used in Eq. 1 and Eq. 2 can be obtained from generalised compressibility factor chart (Fig. 4).

Compressor discharge temperature can be derived from the following polytropic compression expression: $PV^n = C$. Where: P = pressure, V = volume, n = polytropic exponent and C = constant

$$P_1V_1^n = P_2V_2^n \Rightarrow \frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n \Rightarrow \frac{P_1}{P_2} = \left(\frac{RT_2}{P_2}\times\frac{P_1}{RT_1}\right)^n \Rightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \Rightarrow T_2 = T_1\times\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \quad \text{Eq. 5}$$

Thermodynamic characteristics of real gas (i.e. polytropic exponents) can be estimated by real gas equations of state, Lee-Kelser-Plocker (LKP) equations of state are used in this article. It would be impractical to produce complete steps of LKP in this article; only key parameters pertaining to the derivation of polytropic exponents will be presented. For instance, details derivation of reduced volume (V_r) will not be given in this article as it requires extensive calculations with iterative procedures. Reader can refer to literature dedicated for this topic. [2]

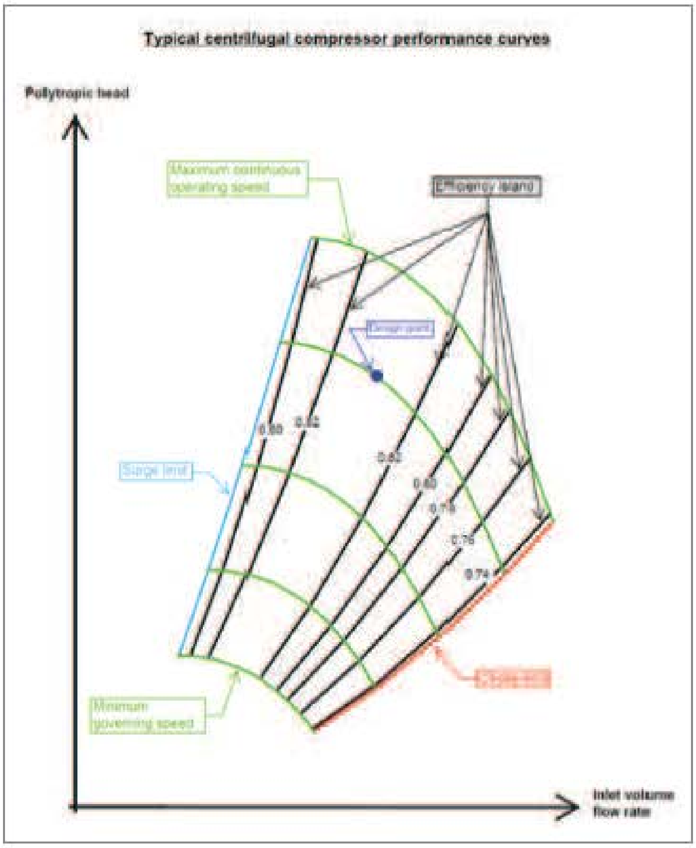

Fig. 5: Typical centrifugal compressor performance curves

Isentropic volume exponent, Kv is defined as:

$$K_v = -\frac{v}{P}\frac{C_p}{C_v}\left(\frac{\partial P}{\partial v}\right)_T$$

Where:

$$\left(\frac{\partial P}{\partial v}\right)_T = \left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}\frac{P_c^2}{ugc\times T_c}$$

For LKP equation of state, the partial derivative $\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}$ is defined as follows:

$$\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r} = -\frac{T_r}{v_r^2}\left[1+\frac{2B}{v_r}+\frac{3C}{v_r^2}+\frac{6D}{v_r^5}+\frac{c_4}{T_r^3 v_r^2}\left\{3\beta+\left(5-2\left(\beta+\frac{\gamma}{v_r^2}\right)\right)\frac{\gamma}{v_r^2}\right\}\exp\left(-\frac{\gamma}{v_r^2}\right)\right]$$

Where:

$$B = b_1 - \frac{b_2}{T_r} - \frac{b_3}{T_r^2} - \frac{b_4}{T_r^3}$$

$$C = c_1 - \frac{c_2}{T_r} + \frac{c_3}{T_r^3}$$

$$D = d_1 + \frac{d_2}{T_r}$$

(Continued on page 25)

Constant	Simple Fluid	Reference
b1	0.1181193	0.2026579
b2	0.265728	0.331511
b3	0.154790	0.027655
b4	0.030323	0.203488
c1	0.0236744	0.0313385
c2	0.0186984	0.0503618
c3	0.0	0.016901
c4	0.042724	0.041577
d1	0.0000155488	0.000048736
d3	0.0000623689	0.00000740336
β	0.65392	1.226
γ	0.060167	0.03754

Two values of $\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}$ will be calculated,

namely $\left.\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}\right|^{simple}$ and $\left.\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}\right|^{reference}$

$\left.\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}\right|^{simple}$ shall be evaluated with constants of "simple fluid".

$\left.\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}\right|^{reference}$ shall be evaluated with constants of "reference fluid".

Isentropic temperature exponent, Kt is defined as:

$$K_t = \frac{C_p}{C_p + P\frac{\left(\frac{\partial P}{\partial T}\right)_v}{\left(\frac{\partial P}{\partial v}\right)_T}}$$

Where:

$$\left(\frac{\partial P}{\partial T}\right)_v = \left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}\frac{P_c}{T_c}$$

For LKP equation of state, the partial derivative $\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}$ is defined as follows:

$$\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r} = \frac{1}{v_r}\left[1 + \frac{b_1 + \frac{b_3}{T_r^2} + \frac{2b_4}{T_r^3}}{v_r} + \frac{c_1 - \frac{2c_3}{T_r^3}}{v_r^2} + \right.$$

$$\left.\frac{d_1}{v_r^5} - \frac{2c_4}{T_r^3 v_r^2}\left\{\left(\beta + \frac{\gamma}{v_r^2}\right)\exp\left(-\frac{\gamma}{v_r^2}\right)\right\}\right]$$

Where:

Constant	Simple Fluid	Reference
b1	0.1181193	0.2026579
b3	0.154790	0.027655
b4	0.030323	0.203488
c1	0.0236744	0.0313385
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c4	0.042724	0.041577
d1	0.0000155488	0.000048736
β	0.65392	1.226
γ	0.060167	0.03754

Two values of $\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}$ will be calculated,

namely $\left.\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}\right|^{simple}$ and $\left.\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}\right|^{reference}$

$\left.\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}\right|^{simple}$ shall be evaluated with constants of "simple fluid".

$\left.\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}\right|^{reference}$ shall be evaluated with constants of "reference fluid".

Both $\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}$ and $\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}$ shall be interpolated by Pitzer acentric factor, ω with the following formula:

$$X = X^{simple} + \frac{\omega}{0.3978}\left(X^{reference} - X^{simple}\right)$$

Where:

$$X = \left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}, \left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}$$ and

$$\omega = \frac{\ln\left[\frac{101325}{P_c}\right] - 5.92714 + \frac{6.09648}{T_{br}} + 1.28862\ln T_{br} - 0.169347T_{br}^6}{15.2518 - \frac{15.6875}{T_{br}} - 13.4721\ln T_{br} + 0.43577T_{br}^6}$$

Substituting the values of $\left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}$ and $\left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}$ into

$\left(\frac{\partial P}{\partial v}\right)_T = \left(\frac{\partial P_r}{\partial v_r}\right)_{T_r}\frac{P_c^2}{ugc \times T_c}$ and $\left(\frac{\partial P}{\partial T}\right)_v = \left(\frac{\partial P_r}{\partial T_r}\right)_{v_r}\frac{P_c}{T_c}$ respectively.

Both Kv and Kt can now be calculated.

Where:

$$K_v = -\frac{v}{P}\frac{C_p}{C_v}\left(\frac{\partial P}{\partial v}\right)_T \qquad K_t = \frac{C_p}{C_p + P\frac{\left(\frac{\partial P}{\partial T}\right)_v}{\left(\frac{\partial P}{\partial v}\right)_T}}$$

Gas compression undergoes polytropic process, therefore both isentropic exponent Kv and Kt must be converted into polytropic exponents, namely:

1. Polytropic volume exponent,

$$n_v = \frac{K_v}{1+\left(1-\frac{1}{\eta_p}\right)\frac{K_t-1}{K_t}K_v} \quad \text{Eq. 6}$$

2. Polytropic temperature exponent,

$$n_t = \frac{1}{1-\frac{Z\times ugc}{C_p}\left(\frac{1}{\eta_p}-1\right)-\frac{K_t-1}{K_t}} \quad \text{Eq. 7}$$

These polytropic exponents (Eq. 6 and Eq. 7) are used for calculating polytropic head and compressor discharge temperature.

1. Polytropic head, $H_p = Z_1RT_1\frac{n_v}{n_v-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n_v-1}{n_v}}-1\right]$

2. Discharge temperature, $T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{n_t-1}{n_t}}$

SECTION 2: CENTRIFUGAL COMPRESSOR PERFORMANCE ESTIMATION

In oil and gas projects involving API 617 process hydrocarbon centrifugal compressors, EPC consultant rotating equipment engineers often face challenging tasks (after official process conditions are forwarded to compressor vendor to perform equipment sizing and selection):

a) To perform sensitivity checks against change of process conditions, typical scenarios are changes of suction temperature, suction pressure, change of intermediate pressure (if compressor duty gets split into two process stages due to temperature limitation caused by high pressure ratio), inlet volume flow rate, process gas composition, etc.

b) To establish appropriate alarm set-point(s) on compressor discharge header to pre-alert the operator from operating compressor into Choke region for prolonged duration, it will require a better understanding of compressor head and inlet volume flow rate with respect to compressor performance maps.

c) For a given worst scenario, i.e. highest head required (either low suction pressure or high discharge pressure, or even combination of both), typical query from client is how much gas flow can be handled by the selected centrifugal compressor within the capacity of selected driver (e.g. gas turbine, electric motor, steam turbine, etc).

THIS SECTIONPROVIDES GUIDANCE FOR PREDICTING CENTRIFUGAL COMPRESSOR AERODYNAMIC PERFORMANCES

1) To obtain centrifugal compressor performance curves from vendor, essential data set are:
   a. Polytropic head vs. inlet volume flow rate
   b. Polytropic efficiency vs. inlet volume flow rate
   c. Discharge pressure vs. inlet volume flow rate (this is used to check if calculated values are within ballpark figures).
2) Extraction of performance data from vendor's curves of item 1a and 1b above.
3) Converting polytropic head into work input factor (as detailed in the following section, refer to Eq. 9).
4) In parallel with item 3, convert inlet volume flow rate into dimensionless parameter, i.e. flow coefficient (refer to Eq. 10).
5) Pairing polytropic efficiency with flow coefficient as mentioned in item 4.
6) Item 3 and 4 will then form the compressor characteristic curve, namely work input factor vs. flow coefficient (refer to Fig. 8).
7) Reduce the following data set (7a, 7b and 7c) into flow coefficient and find the corresponding work input factor per item 6) above:
   a. inlet volume flow rate
   b. equivalent impeller diameter
   c. equivalent impeller tip speed.
8) Use the same flow coefficient of item 7 to obtain polytropic efficiency as depicted by the relation of item 5.
9) Finally, expand work input factor into polytropic head coefficient using Eq. 8 described in the following section. Transforming polytropic head coefficient into polytropic head per Eq. 8.
10) Compressor discharge pressure, discharge temperature and compression power can be estimated with Eq. 2, Eq. 5 and Eq. 12, respectively.

Polytropic head coefficient is defined as follows.

$$\psi_p = \frac{2H_p}{u^2} = 2\eta_p s \quad \text{Eq. 8 [1]}$$

Re-arranging Eq. 8 so that work input factoris related topolytropic head, polytropic efficiency and impeller tip speed.

$$s = \frac{H_p}{\eta_p u_{eq}^2} \quad \text{Eq. 9}$$

(Note: "equivalent impeller tip speed" is used instead of impeller tip speed)

To estimate compressor discharge pressure (P2), basic information needed are:

a) Aerodynamic component: Polytropic head
b) Thermodynamic components: Gas inlet compressibility factor, specific gas constant, suction temperature, polytropic volume exponent and suction pressure.

This section outlines the essence of aerodynamic part of centrifugal compressor, i.e. polytropic head. Gas thermodynamics (e.g. polytropic exponents, isentropic exponents and compressibility factor) are briefly explained in Section 1 of this article.

Flow coefficient (which is widely used in centrifugal compressor terminology) is defined as compressor inlet volume flow rate divided by the product of impeller area (from impeller tip-to-tip) and impeller tip speed:

$$\varphi = \frac{4\dot{Q}}{\pi D_{eq}^{\ 2} u_{eq}} \qquad \text{Eq. 10 [1]}$$

Eq. 10 is identical with PTC10's [4] definition for flow coefficient (refer to clause 2.5.2 of PTC10, 1997). PTC10 defines flow coefficientas follows:

"The flow coefficient is a dimensionless parameter defined as the compressed mass flow rate divided by the product of inlet density, rotational speed, and the cube of the blade tip diameter. Compressed mass flow rate is the net mass flow rate through the rotor"

To line-up Eq. 10 and PTC10's definition for flow coefficient, it can be seen that ratio of mass flow rate to inlet density of PTC10 is equivalent to compressor inlet volume flow rate of Eq. 10, and product of rotational speed with cube of blade tip diameter of PTC10 is equivalent to the product of impeller area with tip speed of Eq. 10.

Please note that $\frac{4}{\pi}$ in Equation 4 may be omitted since it is a mere combination of numerical constant, i.e. 1.27324.

Example of flow coefficient calculated with Equation 4 and simplified version of Equation 4 omitting $\frac{4}{\pi}$ (please refer to Table 1).

Impeller diameter = 0.7000metre
Impeller tip speed = 220 m/s

Table 1: Example of flow coefficient calculations

Q_dot (m^3/s)	4*Q_dot/(pi*D^2*U)	Q_dot/(D^2*U)
4	0.04724	0.03711
4.2	0.04961	0.03896
4.4	0.05197	0.04082
4.6	0.05433	0.04267
4.8	0.05669	0.04453
5	0.05906	0.04638

Since there are many variances of flow coefficient (some may define volume flow rate in term of US customary unit, e.g. gallons per minute, cubic feet per minute, etc) being used in the oil and gas industry, it would be wise to havea definition (including unit of measurements) of flow coefficient clearly defined upfront to avoid potential misunderstanding or misinterpretation between the end user and the compressor manufacturer.

With the knowledge of work input factor and flow coefficient, oneshould be able to convert the following vendor data, namely:

a) Polytropic head vs. inlet volume flow rate into work input factor vs. flow coefficient

b) Polytropic efficiency vs. inlet volume flow rate into polytropic efficiency vs. flow coefficient.

EQUIVALENT IMPELLER DIAMETER

Note that equivalent impeller diameter (i.e. pseudo impeller diameter that represents overall impeller diameter of multi-stage compressor line-up) and associated equivalent impeller tip speed shall be used to expand Eq. 9 and Eq. 10 into polytropic head and inlet volume flow rate, respectively.

This is due to the fundamental fact that cumulative (or total) polytropic heads are typically reported in vendor's performance maps (i.e. not polytropic head developed by one impeller); refer to Fig. 6 for illustrative purpose.

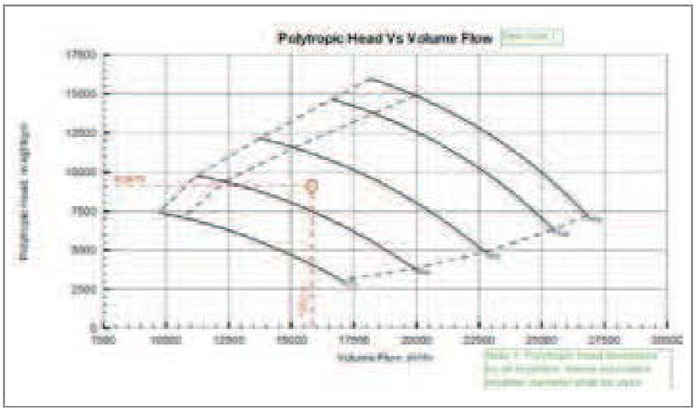

Fig. 6: Typical centrifugal compressor performance maps; polytropic head vs. inlet volume flow rate

Equivalent impeller diameter is defined as follows:

$$D_{eq} = \sqrt{d_1^2 + d_2^2 + d_3^2 + d_4^2 + \ldots} \qquad \text{Eq. 11}$$

(Note: subscript denotes impeller number)

WORKED EXAMPLE DEMONSTRATING DERIVATION OF CENTRIFUGAL COMPRESSOR CHARACTERISTIC CURVES BASED ON COMPRESSOR PERFORMANCE MAPS

Centrifugal compressor performance data extraction (both polytropic head vs. inlet volume flow rate and polytropic efficiency vs. inlet volume flow rate) from vendor's compressor performance maps (arbitrary selected near 100% speed, as it would yield accurate results if new compression duties are near to the selected speed region).

With the following impeller information:

a) Equivalent impeller diameter = 0.3810 meters

b) Equivalent impeller tip speed = 279.29 m/s

Work input factor, flow coefficient and associated polytropic efficiency are derived as per Table 3.

Table 2: Tabulated compressor performance data at 14000 rpm speed region with 5 points cover from surge to Stonewall limit

Inlet Volume Flow Rate (m^3/hr)	Polytropic Head (meter)	Polytropic Head (J/kg)	Polytropic Efficiency
412.76	4043.76	39655.78	0.67436
559.92	3875.01	38000.84	0.72617
680.00	3472.69	34055.45	0.72261
784.57	2825.34	27707.08	0.65987
867.40	2023.40	19842.79	0.52796

Table 3: Calculated work input factor and flow coefficient based on data reported in Table 2 with Equation 9 and 10

Flow Coefficient	Work Input Factor	Polytropic Efficiency
0.00360	0.75389	0.67436
0.00489	0.67089	0.72617
0.00593	0.60420	0.72261
0.00684	0.53830	0.65988
0.00757	0.48184	0.52796

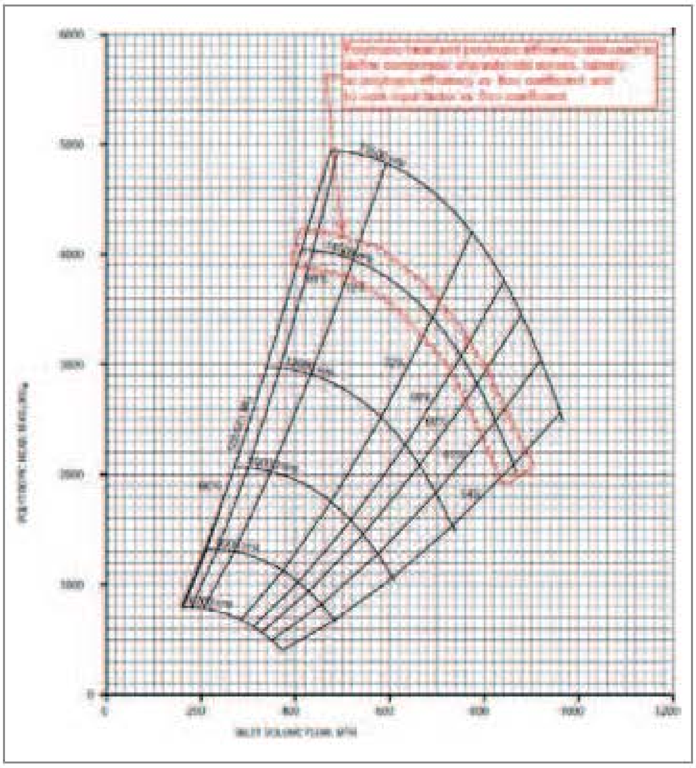

Fig. 7: Data extraction from vendor's performance maps

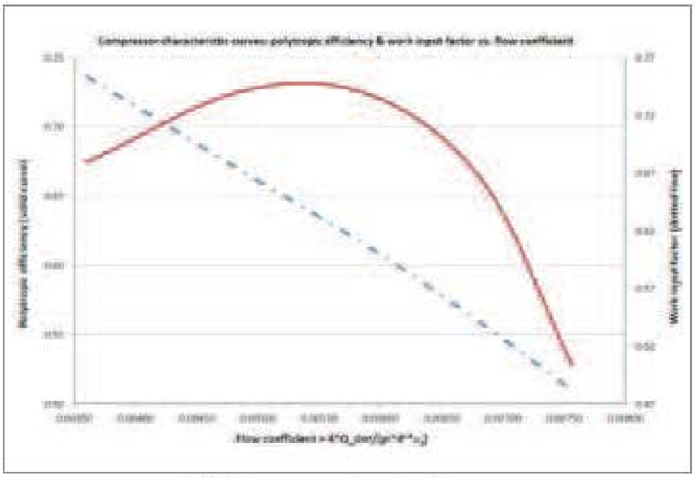

Fig. 8: Compressor characteristic curves (work input factor and polytropic efficiency vs. flow coefficient)

After compressor characteristic curves are obtained, proceed to the following steps utilising input data as listed below:

a) Impeller diameter (remember to convert into equivalent diameter)

b) Rotational speed (which in turn will be transformed into impeller tip speed)

c) Process data (i.e. suction pressure, suction temperature, inlet volume flow rate and gas composition).

Evaluate flow coefficient with input (a) and (b) above, it then gets mapped with work input factor and polytropic efficiency per Fig. 8, and finally expanded to polytropic head coefficient and polytropic head according to Eq. 8.

Compressor discharge pressure and discharge temperature are calculated with Eq. 2 and Eq. 5, respectively.

Lastly, compression power can be estimated using the following formula.

$$power = \frac{\dot{m} H_p}{\eta_p} \qquad \text{Eq. 12}$$

Predicted centrifugal compressor performance data per compressor characteristic curves of Fig. 8:

Compressor Parameter	Data	Unit of Measurement
Suction pressure	91.24	barA
Discharge pressure	147.88	barA
Suction temperature	37	deg. C
Discharge temperature	67.11	deg. C
Gas power required	988.2	kW
Rotational speed	15140	rpm
Polytropic head	46636.53	J/kg
Inlet volume flow rate	500	m^3/hr
Mass flow rate	53581.91	kg/hr
Polytropic efficiency	70.24	%

CONCLUSION

With the methodology described above, it is possible for Rotating Equipment consultants to perform preliminary checks of new compression duty under the performance maps furnished by compressor vendor, which may be beneficial for fast track conceptual study or even Front End Engineering Design projects on extremely tight project schedules.

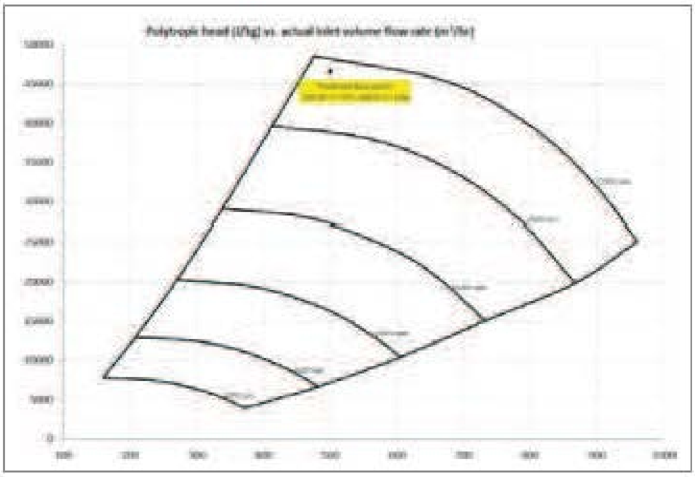

Fig. 9: Predicted centrifugal compressor performance curves

Furthermore, as more and more actual curves/data become available, the in-house developed selection tools will become more accurate, in turn providing even more accurate predictions for future projects to come.

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