CENTRIFUGAL COMPRESSOR APPLICATION SIZING, SELECTION, AND MODELLING

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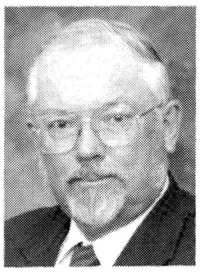

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ABSTRACT

The objective here is to build on the fan law paper presented at the Twentieth Turbomachinery Symposium [1].

The centrifugal sizing and modelling tutorial draws on the pseudo compressor stage curves presented in the fan law paper. The presenters will elaborate and demonstrate the effects of curve shape, gas properties, and other factors on stage matching. For comparison, fan law and actual section properties will be compared to help the ultimate practitioner with the decision of when various methods are reasonable and applicable.

Estimating centrifugal compressor fit for a given application using manufacturer's catalog data will be covered, along with aerodynamic modelling of actual compressor hardware after purchase for monitoring of off-design, fouling, or potential rerating.

SIZING AND SELECTION

Are You in the Range for a Centrifugal Compressor ?

A common question often asked on a new compression application, is whether or not the application will best be fit by a centrifugal *vs* another type of compressor. Many types of compressors other than the centrifugal type are available and can in selected cases better serve a particular application due to peculiarities in their design. A typical chart of various compressor types is presented in Figure 1, although types other than the centrifugal compressor are not discussed in detail. Note that each manufacturer of compressors publishes a similar chart which covers his own particular product line.

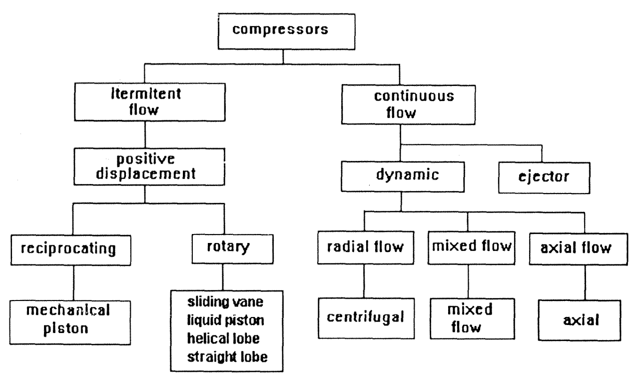

Figure 1. Chart of Compressor Types.

Some of the factors which need to be considered in choosing the proper compressor type are:

Volume Flow: Centrifugal compressors handle volume flows typically higher than reciprocating compressors or other positive displacement types, and lower than the axial flow types.

Pressure ratio and head: Centrifugal compressors are dynamic rather than positive displacement machines and produce head (a specific work term which is shown later), rather than pressure ratio as in a positive displacement compressor. Since head is a function of molecular weight (MW), compressing a light MW gas such as hydrogen to even modest pressure ratios can require significant head (and thus number of stages) in a centrifugal compressor. Very low MW gases with significant pressure boost required, will, therefore, be better handled in a positive displacement type of compressor such as a screw compressor or reciprocating compressor. The two examples given below illustrate the difference in Head required to dynamically compress gases with molecular weights of two and 30, respectively, to the same pressure ratio. (Note from Equation (1) that inlet temperature also affects the head required; however, since this term is in absolute temperature, the affect is relatively minor.)

$$\text{Head} = Z \times \frac{1545}{MW} \times T \times \frac{n}{n-1} \times \left(\frac{p_2}{p_1}^{\frac{n-1}{n}} - 1\right) \qquad (1)$$

where:
- Z = compressibility
- MW = Molecular weight
- T = inlet Temperature (°R)
- n = polytropic exponent
- p_2 = discharge pressure (psia)
- p_1 = inlet pressure (psia)
- Head = polytropic Head (ft-lb_f / lb_m)

$n/n-1$ may be determined from the 'k' value of the gas (ratio of c_p/c_v) and the polytropic efficiency as follows:

$$\frac{n}{n-1} = \frac{k}{k-1} \times \eta \qquad (2)$$

where:
- k = c_p/c_v
- η = polytropic efficiency

example 1:
- MW = 2
- Z = 1.0
- T = 560 R (100 F)
- p_2/p_1 = 1.25
- $n/n-1$ = 2.625 (1.4 k value at 75% efficiency - per equation 2)
- Head required = 100,750 ft.

example 2:
- MW = 30
- Z = 1.0
- T = 560 R (100 F)
- p_2/p_1 = 1.25
- $n/n-1$ = 2.625 (1.4 k value at 75% efficiency - per equation 2)
- Head required = 6715 ft.

MW or volume flow variation: As discussed above, head required is a function of MW. It is not enough to evaluate a single point of operation, but rather all operating points must be evaluated. In some cases, a centrifugal compressor might not be the compressor of choice due to off-design MW variations that may occur as a result of process upset or other conditions, even though the design conditions indicate a good fit. As always, there are two sides to an issue. A multiple operating point scenario may be able to take advantage of the flatter shape of the centrifugal's characteristic curve. The more vertical shape of a positve displacement compressor cannot accomodate similier head situations where the flow may be the variable.

Cost, efficiency, and criticality of service: In certain instances there are "overlaps" between machine types such that either a positive displacement or centrifugal compressor will adequately handle the service. In areas such as this, the positive displacement compressor may have the advantage of lower first cost, with the centrifugal compressor having the advantage of lower maintenance cost. Reliability of the equipment is also a concern, and may result in a decision to purchase an installed spare, which will double the purchased cost. The actual breakpoint where this occurs has changed over the years, and it is left to the operating personnel to provide input based on past experience to determine how much of a premium can be justified in initial installed cost for a given application to result in the lowest long term cost of ownership.

Controls Consideration to fixed *vs* variable speed operation must also be included in the final design decision, since fixed speed will restrict the envelope of operation for a given centrifugal compressor *vs* the same compressor with a variable speed driver, and lead to control solutions to match the operating envelope to the required operating conditions.

A typical speed map is illustrated on Figure 2. Speed variation can be accomplished using a steam turbine or variable speed motor. This solution offers the widest operating range for a centrifugal compressor.

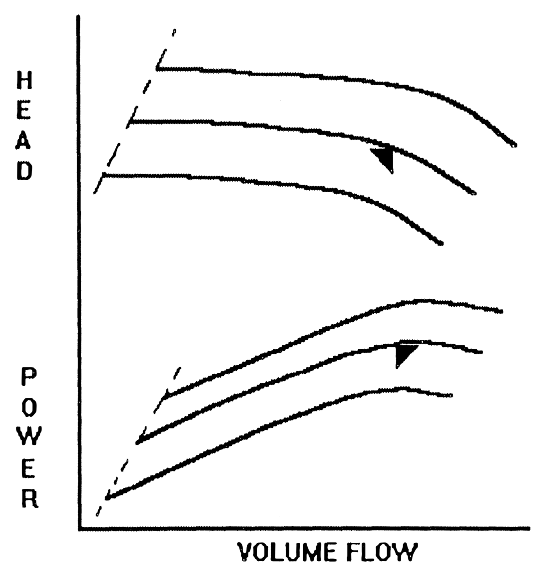

Figure 2. Variable Speed Centrifugal Compressor Operating Envelope.

Another form of control, which is limited to clean gas and normally to single stage centrifugal compressors, is the use of adjustable inlet guide vanes (AIGVs). While used most commonly in single stage machines, they are sometimes employed in multistage configurations. In multistage compressors, variable vanes are normally applied only to the first stage rather than all stages due to design and cost limitations. This limits their effectiveness. For toxic and flammable gases, sealing of the guide vane linkages into the compressor to prevent leakage of process gas is also a consideration.

The use of throttle valves can be a very effective compressor control device. A suction throttle valve is used quite often as an unloading device to aid in the starting of a compressor with a motor driver. Aside from being used to aid in starting, the suction throttle valve can be used to extend the effective operating range of a centrifugal compressor. It is not as effective as guide vanes from an efficiency standpoint, but is very cost effective.

Discharge throttling may stir negative connotations due to wasted energy; however, there are instances where it can be useful. While not too desirable for long term operation, it can be used to trim operation for short term off-design considerations.

Another control method that is commonly used in conjunction with the above methods, is the use of a cooled bypass around the compressor. While this method comes with a power penalty for long term operation, it is quite effective for certain short term off-design operating needs.

Typical ranges of various types of compressors are shown in Figure 3 [2], as a function of pressure ratio and volume flow.

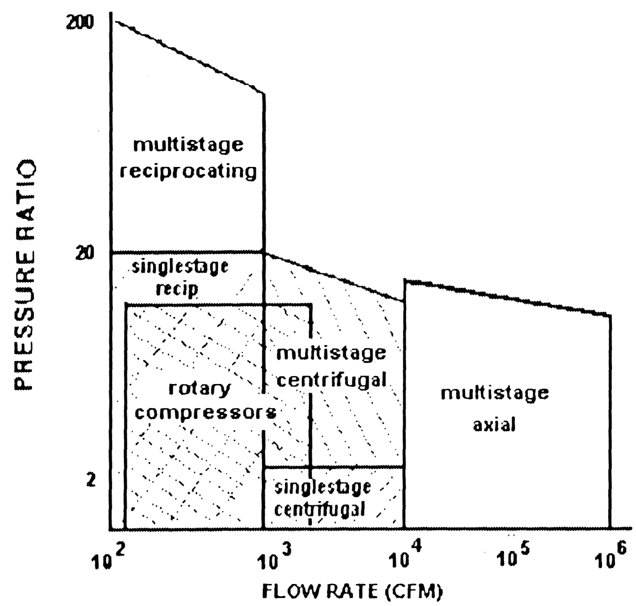

Figure 3. Dynamic and Positive Displacement Compressor Operation Ranges.

CENTRIFUGAL COMPRESSOR CONFIGURATIONS

One advantage of the centrifugal compressor is the wide variety of configurations to which the compressor may be constructed. Sidestream pressures (both extraction and induction), intermediate cooling points, and other custom configurations can all be readily accommodated. This flexibility is not as readily available to a positive displacement type of compressor. The following figures show some of the more common arrangements which have been used for centrifugal compressor [2]. Other, custom arrangements consisting of combinations of these cases also may be available, especially in a fabricated casing. Any estimating program must retain flexibility to handle a variety of configurations.

APPLICATION FACTORS

Gas Properties

A wide variety of gases may be compressed in a centrifugal compressor. The thermodynamic properties of the gas or gas mixture compressed must be established in order for the supplier to properly design the unit.

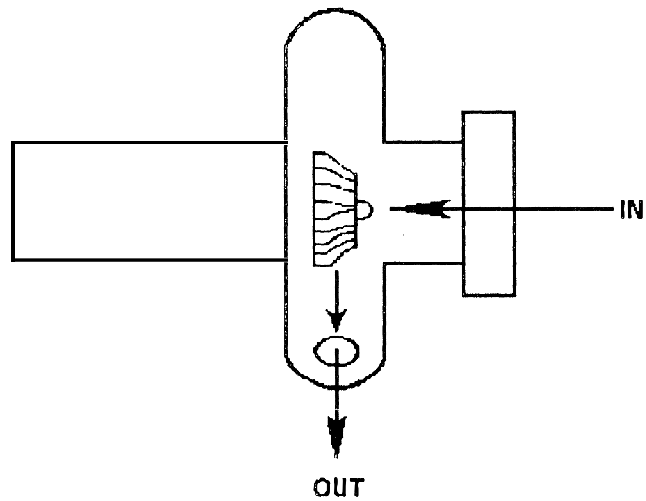

Figure 4. Single Stage Overhung Machine.

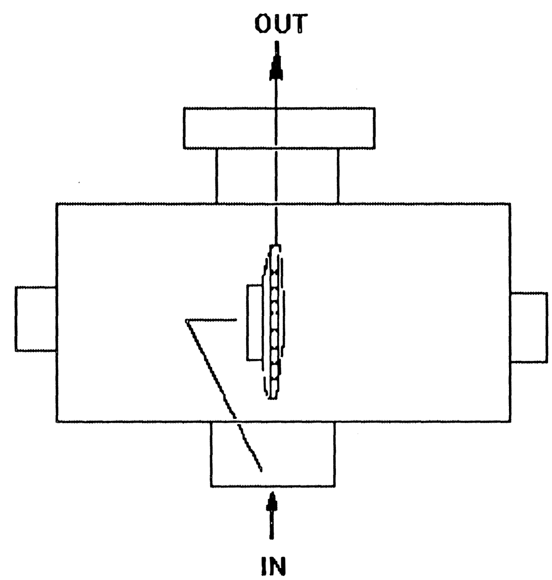

Figure 5. Beam Type Single Stage.

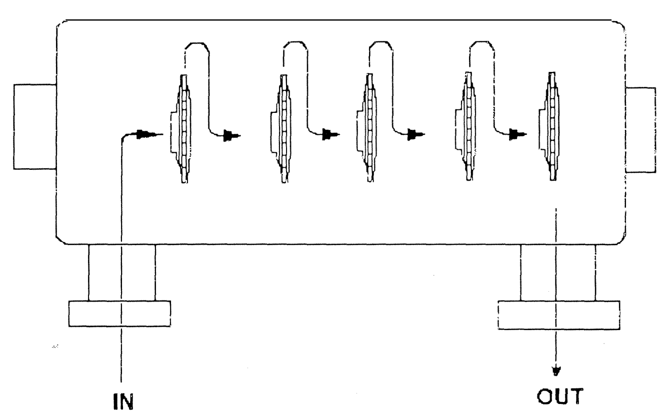

Figure 6. Straight-Through Multistage.

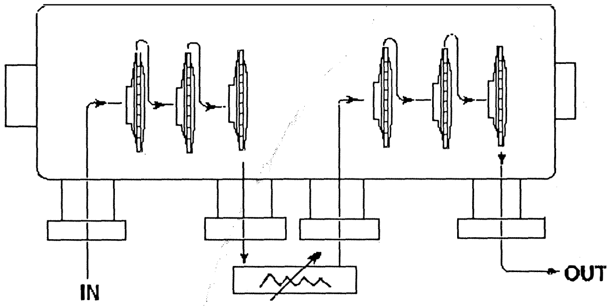

Figure 7. In-out with Single Intercooling.

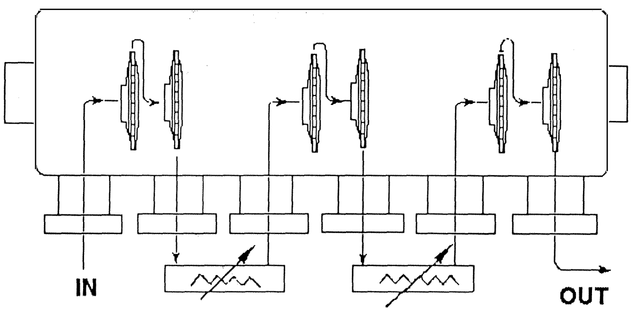

Figure 8. Double-Cooled.

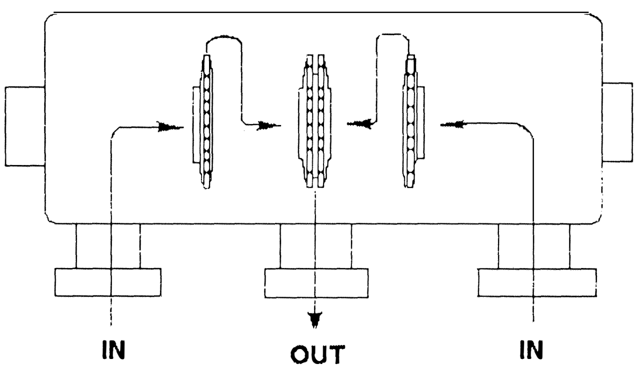

Figure 9. Double-Flow.

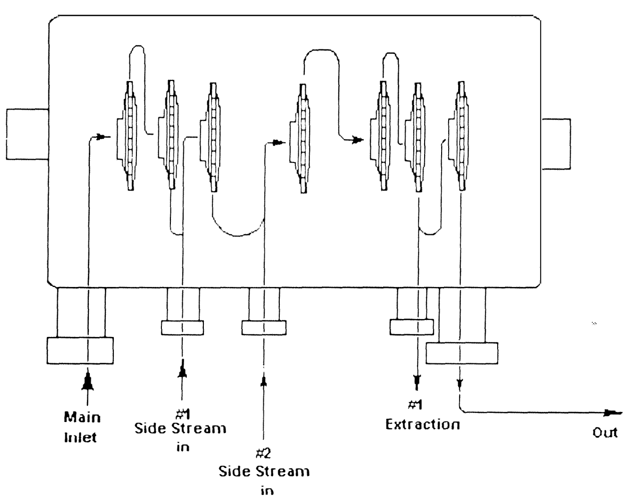

Figure 10. Sidestream Compressor.

Real vs *Ideal gases*

Ideal gases conform exactly to Boyle's Law and Dalton's Law, and are represented by the equation:

$$PV = RT \tag{3}$$

where: P = pressure
V = specific volume
R = Universal Gas Constant
T = Temperature

Real gases deviate from this relationship by a factor known as compressibility (Z). Thus:

$$PV = ZRT \tag{4}$$

Z may be computed for a pure gas or a mixture of gases using generalized compressibility charts based on reduced temperature and pressure. (Ratios of actual to critical values of temperature and pressure.) These charts are widely available from multiple sources [3, 4, 5].

Mollier Methods

A Mollier chart is a diagram of pressure *vs* enthalpy designed to present state properties of a pure gas. Mollier charts have been constructed to very accurately portray a number of pure gases, and were commonly used for such pure gas applications such as refrigeration and compressor performance testing using hand calculations prior to the advent of computers. Later, regions of these charts were curve-fit into the compressor manufacturer's computer selection programs for a number of pure gases. Unfortunately, most compressors compress mixtures of gases, rather than a pure gas. Equations of state, therefore, needed to be run for the actual mixture being compressed to determine the combined properties of the gas mixture. What was needed therefore, was a state point calculation method for gas mixtures that could be incorporated into the manufacturer's actual impeller calculation procedures.

Gas Equations of State

With the advent and availability of computers, it became possible to routinely solve complex equations of state as part of the compressor selection process. Equations of state are gas equations which had been developed to predict state point properties of gases and gas mixtures. Equations of state allow a "Mollier" type calculation for mixtures of gases as well as pure gas, and have been one of the technological innovations in compressor selections over the past 20 years. Some of the various equations of state used include Benedict-Webb-Rubin (including the modifications made by Starling), Peng-Robinson, Redlich-Kwong, and others. For gas mixtures, mixing rules such as Kay's rules or Leland-Muellar are used to established the "mixed" coefficients based on the gas analysis given for the particular equation of state. Specialists in gas properties and equations of state are often required to advise which equation of state is best suited for a given gas or gas mixture and region of interest. Reid, et al. gives additional detail on this topic [4].

Manufacturer's Stage Data

Manufacturer's Catalog Data

Some manufacturers provide catalog data that allows the user to estimate compressor performance for a given service. Typically, these data are a 'locus' of design points for a family of impellers or an average value typifying the impeller design characteristics. Where manufacturers have not provided this type of catalog data, performance can be estimated based on performance of similar design impellers from other manufactures corrected for the actual impeller diameters available, or past units actually quoted or built by the manufacture of question. These data, although adequate for an initial estimate, are inadequate for process refinement or online monitoring against a computer model. More specific data are required for these purposes.

Manufacturer Provided Sectional Curves

In the proposal, the manufacturer will present curves that represent sectional (between nozzle) performance. (Note that process people typically refer to this as a stage, whereas to the manufacturer, a stage refers to an individual impeller.) Overall curves may also be presented. These curves are calculated from the manufacturer's computer prediction programs based on gas properties, impeller empirical design data (discussed next), and various pressure drops, internal leakages, and heat transfers within the compressor for the actual configuration chosen.

Manufacturer's Proprietary Individual Stage Data

A typical representation of what one manufacturer's internal stage curve might look like is shown in Figure 11. Another similar representation for another manufacturer is shown in Figure 12. (These are typical representations rather than actual curves.) Manufacturers have historically closely guarded these stage data, since they represent considerable design effort, and could afford a competitive edge to their competitors if made readily available.

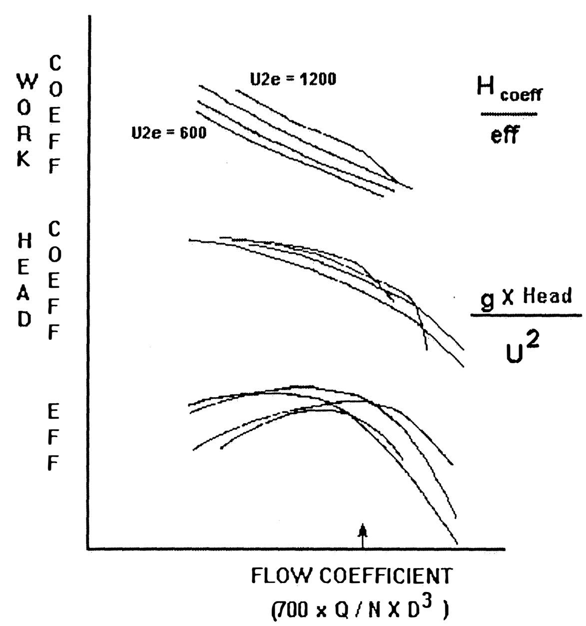

Figure 11. Typical Manufacturer Number 1 Impeller Data.

Since each manufacturer may have a slightly different representation of individual impeller performance, it is necessary to convert stage data if available to a consistent form of representation for aerodynamic modelling in a single program. For the purposes of the author's program, stage data are converted to flow coefficient *vs* Head coefficient and Work input coefficient (These terms are defined below). By using Work input coefficient instead of efficiency, data may be more accurately curve fit since head and work are more nearly straight lines, whereas efficiency is more difficult to curve fit. Efficiency is then

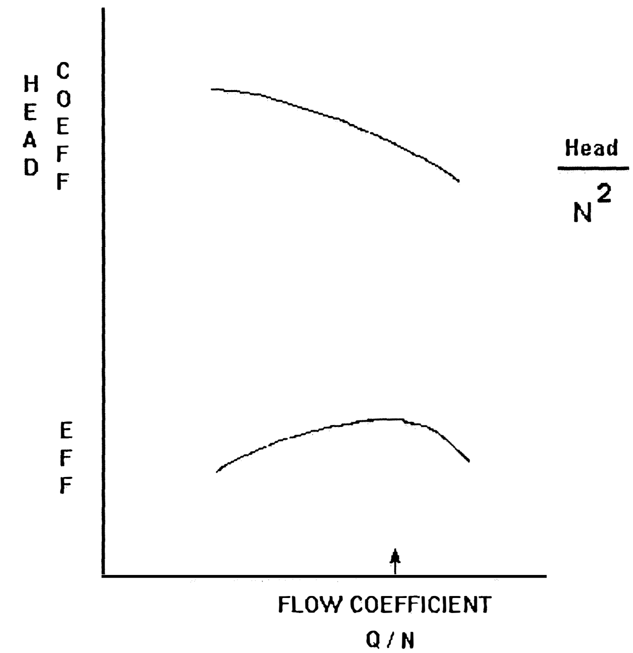

Figure 12. Typical Manufacturer Number 2 Impeller Data.

calculated by dividing the head coefficient by the work coefficient [5, 6].

$$\delta = \frac{700 \times Q}{N \times D^3} \quad \text{(Flow Coefficient)} \qquad (5)$$

$$\mu = \frac{H \times 32.2}{U^2} \quad \text{(Head Coefficient)} \qquad (6)$$

$$\phi = \frac{\mu}{\eta} \quad \text{(Work Coefficient)} \qquad (7)$$

where:

- Q = Volume flow (CFM)
- N = Speed (Rev/min)
- D = Impeller Diameter (inch)
- U = Impeller tip speed (ft/sec)
- H = Polytropic Head
- η = Polytropic efficiency

Due to the cascading or 'snowball' effect on performance, sectional curves discussed above are accurate only for the calculations assumed, and decrease in accuracy as design parameters such as MW, temperature, etc., vary. This topic was discussed in some detail by Brown [1], when he quantified the accuracy expected by 'fan laws.' Curves showing expected deviations in accuracy due to fan law assumptions for various cases are included in that paper. (The data for that paper were generated using the aerodynamic modelling programs discussed here.)

Aerodynamic Modelling

Stage by Stage Methods

The actual calculation of a given impeller's contribution as calculated by the 'n'-method is illustrated below:

This calculation is actually called iteratively for each impeller in each compression section.

example:

MW = 20
k = 1.3
Z = 1.0

p1 = 20 psia
t1 = 100 °F (560 °R)

N = 10000 rpm

1.	mass flow	w (lb/min)	800 lb/min	(given)
2.	specific volume	v1 (ft^3/lb)	15.02 = Z × R × T / 144 × p_1	
3.	volume flow	Q (CFM)	12016 = step 2 × step 3	
4.	impeller diameter	D (inch)	20	(given)
5.	tip speed	(U2 (ft/sec)	872.6 = π × D / 720	
6.	flow coefficient	δ	0.1051	(equation 5)
7.	equivalent tip speed	U_{2e} (ft/sec)	740.3	(equation 3)
8.	head coefficient	μ	0.51	(from manufacturer)
9.	work coefficient	φ	0.68	(from manufacturer)
10.	efficiency	η	0.75 = step 8 / step 9	
11.	$\frac{n}{n-1}$		3.25	(equation 2)
12.	polytropic head	H (ft-lb_m/lb_f)	12060 = step 8 × (step 5)^2 / 32.2	
13.	H / (Z RT)		0.27878 = step 12 / (Z R T)	
14.	temperature ratio	r_t	1.0858 = 1 + (step 13 / step 11)	
15.	pressure ratio	r_p	1.3067 = step 14 ^ step 11	
16.	discharge pressure	p_2	26.134 = step 13× p_1	
17.	discharge temperature	T2	608 °R = step 14 × (t1 °R)	

Both the 'n'-method stage by stage and 'Mollier' stage by stage methods and examples are covered in the Elliott Publication [5] and from other sources.

Other routines comprising balance piston leakage, nozzle pressure drops, machine configuration, etc., must of course by 'built around' these stage calculations to model an entire compressor. Iterations must also be performed for calculations such as balance leakage mixing that uses a discharge temperature that can only be initially estimated until after the final compressor stage is calculated.

Impeller Tip Speed Limitations

Impellers have design limits imposed both from an aerodynamic and a mechanical standpoint. From an existing machine standpoint, the mechanical limit of an impeller is best estimated from the overspeed test that was originally imposed on the impeller. API Standard 617 [7] requires that an impeller be overspeed tested at least 15 percent above the maximum continuous speed. (Note that the manufacturer may have oversped the impeller only to the required API limit rather than to his predicted design limit, in which case additional margin might be available if the impeller were requalified with a new overspeed test.) For the purposes of estimation using catalog data, a value of approximately 12,000 ft of head per stage may be used as a typical mechanical limitation in the absence of better data from the manufacturer's catalog. This value can vary widely, based on material used, impeller design flow coefficient, and of course, impeller design itself.

Mach Number Limitations

Mach number is a ratio of the actual velocity of a gas to the acoustic velocity of the gas at the particular temperature and pressure conditions under consideration. From an aerodynamic standpoint, an "equivalent tip speed" parameter has been developed based on methods derived from one of the manufacturers [5] to describe aerodynamically what the performance of an impeller would be if operating on air rather than the actual gas being compressed. This value is computed by multiplying the actual tip speed (U_2) by the ratio of Mach numbers for the gas being compressed and a base value based on air as follows:

$$U_{2e} = U_2 \times \sqrt{\frac{26.2 \times MW}{k \times Z \times T}} \tag{8}$$

This method partially corrects the impeller design for volume ratio variation. In the absence of more specific information, it is recommended as a typical value that this parameter be kept below 1200 ft/sec, although impeller designs are available that adequately handle higher values.

Mechanical Strength Limitations

Impellers each have their own mechanical stress limitations, as previously discussed, which will lead to yielding of the impeller due to centrifugal forces if exceeded. Each impeller design will have a unique limit due to such factors as the design of the impeller, material of the impeller, fabrication methods, and flow coefficient. (Lower flow coefficient impellers of the same design family can generally operate at higher tip speed, due to the change in geometry as tip width is reduced.) In the absence of more specific data from the actual manufacturer, a tip speed limit of approximately 1000 ft/sec based on normal materials (100,000 yield strength steel) might be considered as typical.

Material Considerations

Specific gases compressed will impose material limitations on the compressor, especially to impellers and other components in the gas stream. (See also Appendix B of API Standard 617 [7].) The list of available materials can be exhaustive, however more common cases and their typical material solutions are listed below:

H_2S—Concentrations of H_2S can lead to sulfide cracking in high strength areas. NACE [8] gives guidelines for concentrations of H_2S which would potentially result in this condition and is referenced by API Standard 617 [7]. Suitable materials for these services are heat treated after fabrication to lower the finished yield strength below 90,000 psi. (A typical tolerance range might be 80,000 to 90,000 psi.) For estimating compressors in such services, impeller tip speed limits must be derated due to having lower yield strength than 'standard' material. This can be estimated from the relationship that stress is approximately proportional to speed squared.

H_2—API Standard 617 [7] requires a vertically split compressor for this service when the partial pressure of hydrogen exceeds 200 psig. Yield strength of impellers should be kept below 120,000 psi to prevent hydrogen embrittlement. These limitations may cause a compromise in allowable tip speed.

CO_2 + H_2O—Carbolic acid may form which would require corrosion-resistant impellers and sleeves of a austinetic stainless steel rather than a carbon steel material. Due to the lower strength of austinetic stainless steel, a penalty in allowable tip speed must be considered.

Lateral Critical Speed Guidelines

In cases where the compressor estimated is a long unit running relatively fast, it is recommended that the manufacturer be requested to furnish a preliminary analysis as early as possible in the quotation stage, or supply installation lists of other units with similar bearing spans (distance between bearings) or number of impellers, operating at the same or higher speeds with similar or heavier couplings.

'Rule of thumb' guidelines are difficult to establish here, but in general realize that a higher MW application needs to be slowed down from an aerodynamic standpoint, and thus may be acceptable mechanically even though the rotor is long. A 'normal' MW (20-30) with a length of eight to ten stages is probably adequate, but should be flagged. Low MW applications require lots of head and may grow to 10 to 12 stages at high speeds. These should be reviewed carefully. Much of course depends on the manufacturer's bearing designs, and stage spacing for impeller design. There is therefore no substitute here for an actual analysis conducted by the manufacturer.

For estimation purposes, prior to contacting manufacturers for more specific data, it is recommended that centrifugal compressors be limited to 10 stages or less in a single casing.

CONCLUSION

Many factors influence the selection of a centrifugal compressor. It is, therefore, out of scope herein to provide additional guidance beyond the rough estimation stage, which will help to determine how well a centrifugal compressor may fit a process application. It must be left up to the equipment manufacturers to provide additional estimates along with final equipment proposals to better define the applicability of their particular equipment for a given case.

There is a need to provide detailed aerodynamic models of compressors within the industry to allow detailed evaluation of off-design and to allow monitoring of machine condition *vs* time due to operational problems such as fouling. This need can be accomplished with manufacturer's stage data and aerodynamic computer models of specific equipment.

Fan law inaccuracies can be significant enough so that for longer units, variations introduced may require that modelling incorporate individual impeller performance curves.

Thermodynamic properties of the gas or gas mixture being compressed, stage mismatching, nozzle pressure drops, Mach number effects, internal heat transfers, and balance piston leakages are some of the other parameters besides the actual impeller data which can effect the accuracy of the aerodynamic model. Only through a rigorous computer solution can all of these factors be adequately considered.

REFERENCES

1. Brown, R. N., "Fan Laws, The Use and Limits in Predicting Centrifugal Compressor Off-design Performance," *Proceedings of the Twentieth Turbomachinery Symposium*, The Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1991).
2. Brown, R. N., *Compressors—Selection and Sizing*, Houston, Texas: Gulf Publishing Company (1986).
3. Nelson, L. C. and Obert, E. F.,"How to Use the New Generalized Compressibility Charts," Chemical Engineering (July 1954).
4. Reid, Prausnitz, and Poling, *The Properties of Gases and Liquids*, Fourth Edition, New York, New York: McGraw-Hill (1987).
5. Elliott Company, "Compressor Refresher," Elliott Company, Jeanette, Pennsylvania.
6. Fullemann, John, "Centrifugal Compressors," reprinted from *Advances in Petroleum Chemistry and Refining*, Department of Chemical Engineering, The University of Texas, New York, New York: John Wiley & Sons (1962).
7. "Centrifugal Compressors for General Refinery Services," API Standard 617, Fifth Edition, American Petroleum Institute, Washington D.C. (1988).
8. "Sulfide Stress Cracking Resistant Metallic Material for Oil Field Equipment," NACE MR-01-90, National Association of Corrosion Engineers.

BIBLIOGRAPHY

Shepherd, D. G., *Principles of Turbomachinery*, New York, New York: The Macmillan Company (1969).

Hallock, D. C., "Centrifugal Compressors, the Cause of the Curve," Air & Gas Engineering (January 1968).