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ABSTRACT

There are currently no test codes and performance standards available for rating of centrifugal refrigeration compressors, like the ANSI/ARI standard 540 for positive displacement refrigerant compressors. The reason for this lack of a standard is the fact that until recently centrifugal compressors were not sold separately but always as part of a centrifugal chiller package. The need of the end customer could be met with ARI standard 550-590 that specifies how to measure chiller performance and ASHRA standard 90.1 that dictates minimum chiller efficiency requirements. With centrifugal refrigeration compressor manufacturers offering stand-alone compressors to OEM's for new chiller development duty as well as retrofit applications, a need for standards on how to test and rate centrifugal refrigeration compressors as individual components has surfaced. This paper discusses the issues involved in representing the performance of centrifugal refrigeration compressors for water- and air-cooled chiller duty. The polynomial performance representation of positive displacement compressors as prescribed by the positive displacement refrigeration compressor standard ANSI/ARI 540 fails to accurately describe centrifugal compressor behavior. The ASME-PTC10 power test code for centrifugal and axial compressor, used extensively by manufacturers serving the air compression and the petrochemical industry, is a more appropriate testing code for centrifugal compresors but is limited in scope to the aerodynamic performance of the compressor, neglecting mechanical, electrical and motor/inverter cooling losses important for the overall efficiency of a refrigeration compressor. An extension of the ASME PTC-10 code to include these effect is required for rating centrifugal refrigeration compressors installed on closed loop chillers.

1. INTRODUCTION

Until recently centrifugal refrigeration compressors were not sold individually. They were part of a centrifugal chiller package and designed in-house by the original equipment manufacturer (OEM) of the complete chiller package. Rating methods and test codes exist for the complete chiller unit. ARI standard 550-590 [1] specifies how to measure chiller performance and ASHRA standard 90.1 [2] prescribes minimum chiller efficiencies. These standards satisfy the needs of the end customer.

Internally, OEM's have developed a centrifugal compressor testing and performance rating procedures to enable the overall performance prediction of the complete chiller system. The internal compressor rating method used by the centrifugal chiller manufacturers is quite different from the rating methodology used for positive displacement compressors.

With centrifugal refrigeration compressor manufacturers now offering stand-alone compressors to OEM's for new chiller development duty as well as retrofit applications [3], a need for standards on how to rate and test centrifugal refrigeration compressors as individual components has surfaced. This paper discusses the issues involved in testing and rating centrifugal refrigeration compressors for water- and air-cooled chiller duty. One distinction between centrifugal compressor rating and positive displacement compressor rating is that the centrifugal compressor was always developed in-house by the chiller manufacturer as part of the complete chiller system. As a result its performance was normally expressed in terms of tank-to-tank or vessel-to-vessel (evaporator-to-condenser) performance instead of the flange-to-flange performance given by positive displacement compressor manufacturers.

Flange-to-flange performance does not account for the pressure drops in the suction and discharge piping, the loss of isolation and check valves or the loss of the dynamic head of the flow entering the condenser. As a result, compressor performance obtained according to the positive displacement refrigeration compressor standard ANSI/ARI 540 [4] has to be de-rated to account for the difference between flange-to-flange and tank-to-tank performance.

2. RATING METHODOLOGY FOR POSITIVE DISPLACEMENT COMPRESSORS

For positive displacement compressors it is common to show compressor performance of four quantities, viz. power $\widehat{P}$, mass flow rate $\dot{m}$, electrical current I and compressor overall efficiency η as a function of suction and discharge saturation temperatures S and D, respectively. According to ANSI/ARI 540 published rating of these quantities need to be described by a third order polynomial:

$$X = c_1 + c_2 S + c_3 D + c_4 S^2 + c_5 SD + c_6 D^2 + c_7 S^3 + c_8 S^2 D + c_9 SD^2 + c_{10} D^3 \qquad (1)$$

Where X could be $\widehat{P}$, $\dot{m}$, I or η. A least squares method is required to establish the coefficients in Equation 1 from compressor performance test data.

The following observations can be made about this rating procedure:

1. Neither suction nor discharge actual temperatures (just the saturation temperatures corresponding to the measured flange pressures) are used for rating the positive displacement compressor performance according to Equation 1. In section 3 of this paper it will be shown that these two temperatures are included for measuring and rating centrifugal compressors.
2. The effect of superheat on compressor performance is neglected in the ANSI/ARI 540 standard. Higher inlet superheat reduces inlet density and therefore reduces mass flow rate. It also increases the isentropic enthalpy rise for a given pressure ratio since the lines of constant entropy become more horizontal with superheat.
3. The discharge temperature at the exit flange of the positive displacement compressor does not contain useful performance rating information since the oil injection during the compression process involves heat transfer between compressed vapor and injected oil.
4. The suction and discharge saturation temperatures S and D are derived from static pressure probes at the suction and discharge flange of the compressor. Compressor work is controlled by inlet and exit total pressures as measured by Pitot tubes and not by static pressures. See Figure 1 for the distinction between the static and total pressure measurements. If the gas velocity is equal on suction and discharge side the static-to-static pressure ratio approaches the total-to-total pressure ratio.
5. The notion of saturation temperature instead of inlet pressure tries to equate these temperatures to the evaporation and condensation temperatures in the evaporator and the condenser. However, pressure drops in suction and discharge piping cause differences between the suction and discharge saturation temperatures at the inlet and exit flanges of the compressor versus those in the heat exchanger vessels. It becomes the responsibility of the OEM to account for piping and vessel pressure drops in order to obtain overall chiller efficiency in terms of COP or kW/tons
6. The ten coefficients of the third order polynomial function for each of the four quantities require a large amount for test data with the possibility of unrealistic wiggles in the polynomials.
7. Applying this same methodology to the rating process of centrifugal compressors will result in errors between the test data supplied to obtain the polynomials and the prediction of these polynomial functions due to the fact that surge and choke phenomena introduce functional relationships that can not be described accurately by a third order polynomial function.

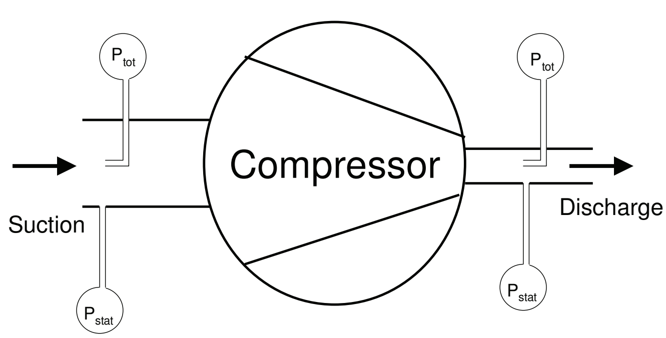

Figure 1. Static and total pressure measurements at compressor suction and discharge flange

3. FLUIDDYNAMIC EFFICIENCY OF CENTRIFUGAL COMPRESSORS

The essentially oil-free vapor being compressed adiabatically in a centrifugal compressor allows the calculation of fluid dynamic compression efficiency, defined as isentropic enthalpy rise divided by actual enthalpy rise, from inlet and exit total temperatures and pressures:

$$\eta = \frac{\Delta h_s}{\Delta h_{act}} = \frac{h(P_2, s_1) - h(P_1, T_1)}{h(P_2, T_2) - h(P_1, T_1)} \tag{2}$$

Equation (2) can used to determine compressor fluid dynamic efficiency as a function of inlet and exit temperatures and pressures:

$$\eta = \eta(P_1, T_1.P_2, T_2) \tag{3}$$

Note that no mass flow rate or power measurement is required to calculate the fluid dynamic compressor efficiency. The four measurements shown in Figure 2 determine the aerodynamic compressor efficiency. This methodology can not be used for positive displacement compressors since the oil injection during the compression process needed for lubrication reduces the compressor discharge temperature T_2, thus invalidating the use of Equation (2).

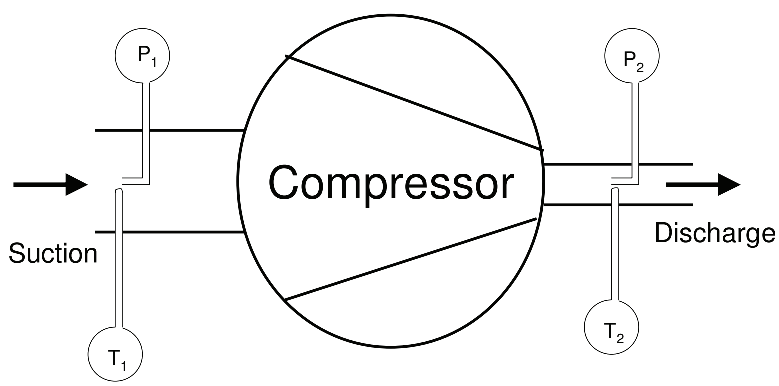

Figure 2. Total pressure and temperature measurements required to determine aerodynamic compressor efficiency

It should however be realized that the aerodynamic efficiency is not equal to the overall compressor efficiency, since motor inefficiency, mechanical losses and internal cooling losses are not accounted for in the fluid dynamic efficiency. Being able to separately determine the fluid dynamic efficiency has been of great help in the development of turbo machinery in general and centrifugal compressors in particular. Turbo machinery phenomena such as surge and choke are pure fluid dynamic effects and can be clearer described independent from other compressor loss mechanisms.

It is important to realize the sensitivity of small changes in discharge temperature on compressor efficiency. For example, an R134a refrigeration compressor with an 80% fluid dynamic efficiency starting from an inlet condition of saturated vapor (no superheat) at 5 ^{0}C and a corresponding inlet pressure of 349.7 kPa, and compressing to a exit saturated vapor temperature of 35 ^{0}C (corresponding to a discharge pressure of 887.0 kPa) will have a discharge temperature of 40.89 ^{0}C.

High accuracy pressure and temperature measurements are required given the sensitivity of aerodynamic efficiency on variations in temperature and pressure. Table 1 shows the variation in efficiency, isentropic enthalpy rise and actual enthalpy rise with small changes in pressure and temperature for this case of the 80% efficient R134a compressor. For example, an increase of 1 ^{0}C in discharge temperature due to a measurements error will reduce the calculated compressor efficiency from 80.00 % to 76.6 %: a 4.2% reduction in efficiency. Other compressor performance parameters are similarly affected by errors in inlet pressure and temperature as well as exit pressure and temperature measurements. Table 2 shows the sensitivity of compressor performance in relative terms (as percentages) on measurement error. The isentropic enthalpy rise not being a function of the discharge temperature T_2 is obviously not affected by discharge temperature measurement errors.

Table 1. Changes in measured isentropic efficiency, isentropic enthalpy rise and actual enthalpy rise of an 80% efficient compressor due to a 1 ^{0}C error in temperature measurement or a 1% error in pressure measurement.

T_1 [^{0}C]	**5**	**6**	**5**	**5**	**5**
P_1 [kPa]	**349.7**	**349.7**	**346.2**	**349.7**	**349.7**
T_2 [^{0}C]	**42.83**	**42.83**	**42.83**	**43.83**	**42.83**
P_2 [kPa]	**887.0**	**887.0**	**887.0**	**887.0**	**878.1**
η [-]	**0.800**	**0.836**	**0.813**	**0.766**	**0.785**
Δh_s [kJ/kg]	**19.31**	**19.42**	**19.56**	**19.31**	**19.10**
Δh_{act} [kJ/kg]	**24.15**	**23.23**	**24.05**	**25.21**	**24.35**
h_1 [kJ/kg]	401.50	402.42	401.60	401.50	401.50
s_1 [kJ/kg-K]	1.7245	1.7278	1.7256	1.7245	1.7245
h_{2s} [kJ/kg]	420.82	421.84	421.16	420.82	420.61
h_2 [kJ/kg]	425.65	425.65	425.65	426.71	425.85

Table 2. The sensitivity of isentropic efficiency, isentropic enthalpy rise and actual enthalpy rise of an 80% efficient R134a compressor due to a 1 ^{0}C error in temperature measurement or a 1% error in pressure measurement.

Deviation in temperature or pressure	Sensitivity of performance parameters $\Delta\eta$	Δh_s	Δh_{act}
+1% ΔP_1	-1.67%	-1.25%	0.41%
+1% ΔP_2	1.88%	1.09%	-0.81%
+1^{0}C ΔT_1	4.55%	0.57%	-3.81%
+1^{0}C ΔT_2	-4.20%	0.00%	4.39%

4. NON-DIMENSIONAL PERFORMANCE PARAMETERS

Using dimensional analysis and similitude, the number of variables that affect the thermodynamic performance of a turbo compressor can be reduced, as described in many textbooks on turbo machinery, e.g. [5-7]. For incompressible flows a non-dimensional head coefficient

$$\frac{gH}{N^2D^2}$$

and a non-dimensional flow coefficient

$$\frac{\dot{Q}}{ND^3}$$

can be introduced. Using these non-dimensional variables, the various speed lines that show up when dimensional head and flow are used as independent variables, collapse to a single performance curve.

The application of dimensional analysis to compressible fluids increases the complexity of the functional relationships. The various speed lines do not collapse to a single curve anymore with head and flow coefficient as independent non-dimensional parameters. Another choice of variables is preferred when appreciable density changes occur across the machine. Isentropic total enthalpy change Δh_s should be used instead of head and mass flow rate $\dot{m}$ should replace the volume flow rate $\dot{Q}$. The performance parameters Δh_s, η for a turbo machine handling a compressible flow are expressed functionally as:

$$\Delta h_s, \eta = f(\mu, N, D, \dot{m}, \rho, a, \gamma) \tag{4}$$

Because ρ and a change through a turbo machine, values for these fluid variables are selected at compressor inlet. Selecting ρ, N, D as common factors, Equation (4) can be re-written in dimensionless form with 4 instead of 7 independent variables as:

$$\frac{\Delta h_s}{a_0^2}, \eta = f\left\{\frac{\dot{m}}{\rho_0 a_0 D^2}, \frac{\rho_0 N D^2}{\mu}, \frac{ND}{a_0}, \gamma\right\} \tag{5}$$

where $\frac{\Delta h_s}{a_0^2}$ = the Mach number corrected isentropic enthalpy rise also known as the Mach number corrected head factor *HF*

η = the compressor efficiency

$\frac{\dot{m}}{\rho_0 a_0 D^2}$ = the Mach number corrected flow factor *FF*

$\frac{\rho_0 N D^2}{\mu}$ = the machine Reynolds number

$\frac{ND}{a_0}$ = blade tip Mach number M_{tip}

γ = isentropic exponent

Neglecting the effects of Reynolds number and changes in isentropic exponent, Equation (9) reduces to

$$HF, \eta = f(FF, M_{tip}) \tag{6}$$

The advantage of the use of Mach number corrected values for head and flow factor is that aerodynamic compressor performance maps obtained for one refrigerant vapor having a certain speed of sound can be applied to other refrigerants with a different speed of sound. Calculating the required head factor (=isentropic enthalpy rise over the compressor divided by the square of the speed of sound) and the required flow factor (=volumetric flow rate divided by speed of sound and the square of the impeller diameter), the required impeller tip Mach number and therefore the required turbo machinery speed and the resulting efficiency can be determine from the performance map generated with another fluid. This procedure has successfully been applied to large industrial refrigeration compressors. For expensive low volume, large capacity centrifugal compressors capacity changes by changing the working fluid is sometimes more cost effective than building new hardware. The same approach has also been used during the transition from CFC- to HCFC- and HFC-refrigerants. It is expected to be useful again during the current transition to low GWP refrigerants.

Thermodynamic compressor performance can be shown graphically on a two-dimensional map with head factor as function of flow factor for different blade tip Mach numbers and inlet guide vane setting angles. If points of equal efficiency are connected, one obtains the compressor efficiency islands, as shown in Figure 3.

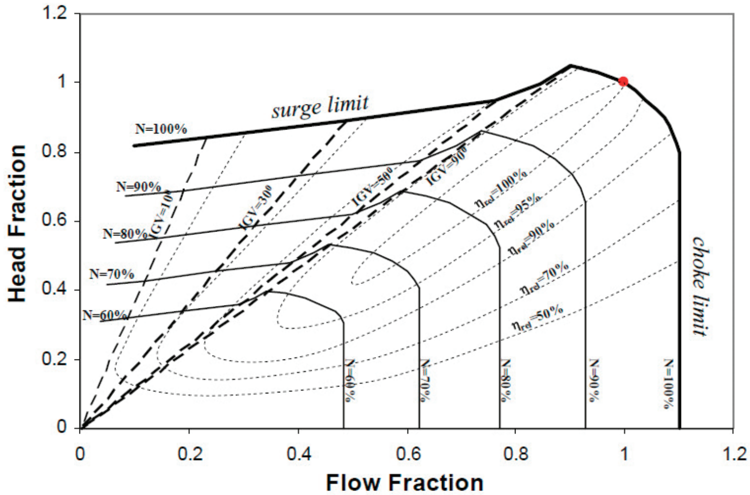

Figure 3. Compressor map of a variable-speed, variable-IGV centrifugal compressor

4. The transition from aerodynamic efficiency to overall efficiency

The American Society of Mechanical Engineers (ASME) has developed a power test code for centrifugal and axial compressors and blowers/exhausters ASME-PTC10 [8]. The purpose of this code is to establish rules for conducting and reporting tests on a turbo compressor under such conditions and in such a way that its thermodynamic performance on a specified gas of known properties under specified conditions can be predicted. The code is very specific in its recommendation for instrumentation set-up and required accuracy. It is suggested that many of the ASME PTC procedures are to be included in a future code for centrifugal refrigeration compressor testing and rating.

However, PTC-10 limits its scope to the thermodynamic performance of the compressor, neglecting mechanical, motor and drive losses that have to be included for rating centrifugal refrigeration compressors. To determine the overall efficiency of a centrifugal refrigeration compressor we have to account for two additional efficiencies / loss mechanisms:

1. the mechanical efficiency η_{mech}. This efficiency accounts for the power loss between drive shaft power and impeller shaft power as a result of the frictional forces experienced by the bearings and the transmission. It is proposed to define the mechanical efficiencies as a polynomial function of power, suction density and shaft speed:

$$\eta_{mech} = f(\hat{P}, \rho_0, N) \tag{7}$$

2. the electrical efficiency (motor + inverter (if present)) η_{elec}. This efficiency accounts for the electrical losses in the motor and the inverter. It is proposed to define the electrical efficiencies as a polynomial function of power, suction density and shaft speed:

$$\eta_{elec} = f(\hat{P}, N) \tag{8}$$

The overall compressor efficiency is the product of the aero, mechanical and electrical efficiencies:

$$\eta_{overall} = \eta_{aero}\eta_{mech}\eta_{elec} \tag{9}$$

5. The effect of internal cooling on compressor efficiency

For centrifugal compressors the motor is normally cooled with high-pressure liquid refrigerant obtained from the condenser. A similar arrangement exits for liquid-refrigerant cooled inverters. Removing liquid from the condenser for motor and inverter cooling causes a difference between evaporator and compressor/condenser mass flow rates. As a result the apparent overall efficiency of the refrigeration compressor further reduced.

With the definition of the concept of the ideal cycle coefficient of performance:

$$COP_{cycle,ideal} = \frac{\Delta h_{evaporator}}{\Delta h_{s,compressor}} \tag{10}$$

we can define the overall compressor efficiency including the effect of internal cooling as follows:

$$\eta_{overall} = \eta_{aero}\eta_{mech}\eta_{elc}\left(1 - \frac{1-\eta_{mech}\eta_{elec}}{\eta_{aero}\eta_{mech}\eta_{elec}COP_{cycle,ideal}}\right) \tag{11}$$

It is this overall efficiency that should be used to evaluate the centrifugal compressor for chiller duty.

6. Conclusions

Centrifugal compressor ratings should be based on aerodynamic performance maps relating compressor head and flow to efficiency and blade tip Mach number in combination with mechanical and electrical efficiencies and the performance reduction resulting from the internal refrigerant cooling of the compressor.

The aerodynamic efficiency of a centrifugal compressor as derived from test data should be documented in the form of a compressor map as a function of head and flow factor. The use of polynomial functions is not recommended for the determination of the compressor aero efficiency. .

The mechanical efficiency as derived from test data might be described as a polynomial function of power, speed and gas density inside the motor housing

The electrical efficiency as derived from test data might be described by a polynomial function of power and speed

The inverter efficiency as derived from test data might be described by a polynomial function of power and speed

Mechanical cooling (oil cooler), motor cooling and inverter cooling should be accounted for in the overall compressor performance rating

7. References

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