2002

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ABSTRACT

This paper proposes a method of evaluating the performance of reciprocating compressors. An analytical formula of the volumetric efficiency is presented. This formula, based on both theoretical and empirical approach, takes account of the most important factors, which occur during the working process of the compressor and affect the refrigeration performance.
An accurate determination of the clearance volume of the compressor and a good knowledge of the temperature of the suction gas in the cylinder are required, to get a good agreement between calculation and experiment. The simulation of the capacity and the compression work, under various operating conditions, can give a good idea of the compressor performance. The method presented can be easily computed and should help the compressor designer at very early step of his engineering process.

NOMENCLATURE

Q	:	cooling capacity
q_m	:	mass flow rate
ΔH	:	enthalpy difference
ρ	:	gas density
q_v	:	volume flow rate
V	:	volume
f	:	frequency
η_V	:	volumetric efficiency
τ	:	compression ratio
γ	:	polytropic exponent of compression and expansion process
ε	:	clearance volume ratio
M	:	molar mass
P	:	absolute pressure
R	:	ideal gas constant
T	:	absolute temperature
α	:	warm up coefficient
P_W	:	power
C_P	:	specific heat for constant pressure

INTRODUCTION

Compressor manufacturers have to design, develop and produce compressors to meet the energy efficiency requirement of their customers. The knowledge of the compressor efficiency should be investigated at early step of the development. The compressor efficiency is evaluated through the calculation of the coefficient of performance, defined as the ratio of the cooling capacity to the power input. The compressor efficiency is comprised of volumetric efficiency, compression efficiency, mechanical efficiency and motor efficiency.

This paper mainly insists on the method of evaluating the cooling capacity of reciprocating compressor thanks to the calculation of the volumetric efficiency and the estimation of the suction gas temperature in the suction plenum of the cylinderhead of compressor. For the determination of the thermodynamic properties required in the calculation, such as the enthalpy difference between superheat and subcooled points and suction gas density, computer programs are now available and are commonly used to obtain accurate data on refrigerant at the working condition of the compressor.

Among the long list of authors who propose their own approach of the volumetric efficiency, the reader can pay attention to the references, which include some very interesting research works. [1] and [2] are recalled here as historical references useful at their time when no computer facilities were available. [3], [4], and [5], are more sophisticated including analytical formulas to calculate coefficients to obtain the volumetric efficiency. [6] also proposes a very comprehensive work on the temperature rise of the gas inside the compressor.

This paper presents new formulas for the estimation of the volumetric efficiency and the suction gas temperature, to get the cooling capacity of reciprocating hermetic compressor.

DESCRIPTION

For a given displacement and at a certain speed of compressor, the cooling capacity can be determined as the product of the mass flow rate into the enthalpy increase during evaporation:

$$Q = q_m \ \Delta H$$

The mass flow rate is the product of the gas density into the volume flow rate:

$$q_m = \rho \ q_v$$

The volume flow rate is related to the effective volume of the gas ingested during the suction process, by the formula:

$$q_v = V_{suction} \ \ f$$

The volumetric efficiency of a reciprocating compressor is defined as the ratio of the effective volume of gas ingested in the cylinder bore during the suction process to the swept volume during a revolution, for a given operating condition:

$$\eta_V = \frac{V_{suction}}{V_{swept}}$$

The theoretical value of the volumetric efficiency can be easily determined by the following formula:

$$\eta_V = 1 - \left(\tau^{\frac{1}{\gamma}} - 1 \right) \varepsilon$$

The clearance volume ratio is the ratio between the clearance volume to the compressor displacement. Each compressor has its own dead volume, which depends on its geometrical features.

The effective volumetric efficiency is always smaller than the theoretical value because of many factors depending of the design of compressors. Among those well-known factors, let’s recall some of them:

- delay in opening and closure of valves implying back flow through ports
- throttling and flutter effects of valves
- leakage of the valves
- blow by between piston and cylinder
- oil mixed with refrigerant

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The practical formula mentioned in [7] takes only account of the pressure ratio in the following linear equation:

$$\eta_V = 1 - 0.05\,\tau$$

The effective volumetric efficiency is always higher than the practical value. So, experience on refrigeration compressor leads to discard both of the previous formulas.

The author proposes a new empirical formula to approach effective volumetric efficiency. This new formula takes mainly account of the compression ratio, depending of refrigerant and operating condition, and includes the clearance volume ratio, depending of the compressor features. The formula has also coefficients of correction, and takes account of the polytropic exponent and the evaporating temperature in degree Celsius:

$$\eta_V = 1 + \frac{a + b\,t_{evap}}{\gamma} - \left(\frac{c + \varepsilon}{1 + d\,\tau} \right) \tau$$

The proposed formula has some common features with the formula found by Claude Marioton [9], mentioned in the references [7] and [8]. Compared with this formula in reference, the proposed one has different values of c and d, uses the clearance ratio instead of the clearance volume, includes a coefficient of correction with two coefficients a and b, the evaporating temperature and the polytropic exponent. The formula assumes ideal gas behavior with a constant polytropic exponent.

The coefficient has to be adjusted to well match experimental results. The suitable set of coefficients proposed below, was used by the author in his paper:

$$a = 0.04 \qquad b = 0.0005 \qquad c = 0.01 \qquad d = 0.004$$

The knowledge of the effective volumetric efficiency is a first step in the calculation of the cooling capacity, but it is also necessary to estimate the temperature of the gas in the suction plenum of the compressor, to get the suitable value of the density of the suction gas:

$$\rho = \frac{M\ P}{R\ T}$$

The superheat of the suction gas is caused by overheat due to the losses coming mainly from the compression, the mechanic, and the motor. It is, of course, very complicated to get a global view of all the phenomena, and methods proposed by [5] and [6] are very difficult to use because requiring a lot of data to get an idea of the temperature of the suction gas. The author proposes a simple approach, which gives a very good agreement between experimental results obtained from thermal audit on compressor versus analytical calculation.

The method is based on two simplified equations giving the suction temperature according to the ambient temperature around the compressor, and the heat increase related to the compression process with its associated power:

$$T_{suction} = \alpha\ T_{ambient} + \Delta T$$

$$P_W = C_P\ q_m\ \Delta T$$

From the two equations, it is possible to get the heat increase during suction process thanks to the calculation of the power during compression process :

$$\Delta T = \frac{P_W}{C_P\ q_m} = \frac{T_{suction}\ \ln \tau}{C_P\ \dfrac{M}{R}}$$

Then, the suction temperature can be estimated from the previous equations:

$$T_{suction} = \frac{\alpha \, T_{ambient}}{1 - \dfrac{\ln \tau}{C_P \, \dfrac{M}{R}}}$$

Experience shows that this formula needs to be modified to the following one in order to better take account of the evaporating and return gas temperatures:

$$T_{suction} = \frac{\alpha \left(273 + \dfrac{t_{ambient} + \dfrac{T_{return\ gas}}{T_{evaporation}} t_{return\ gas}}{2} \right)}{1 - \dfrac{R \ln \tau}{C_P \; M}}$$

The warm up coefficient α depends on the way the compressor is cooled. As starting proposal, the coefficient can be put at 1 if the compressor is static and 0.95 if the compressor is fan.

The coefficient of performance can be estimated as the ratio of the cooling capacity to the power input. The power input to a compressor can be expressed as:

$$\frac{\dfrac{\gamma}{\gamma - 1} P_{suction} \, V_{swept} \left(\tau^{\frac{\gamma - 1}{\gamma}} - 1 \right) \eta_V \;\, f}{\eta_C \; \eta_{Mech} \; \eta_{Motor}}$$

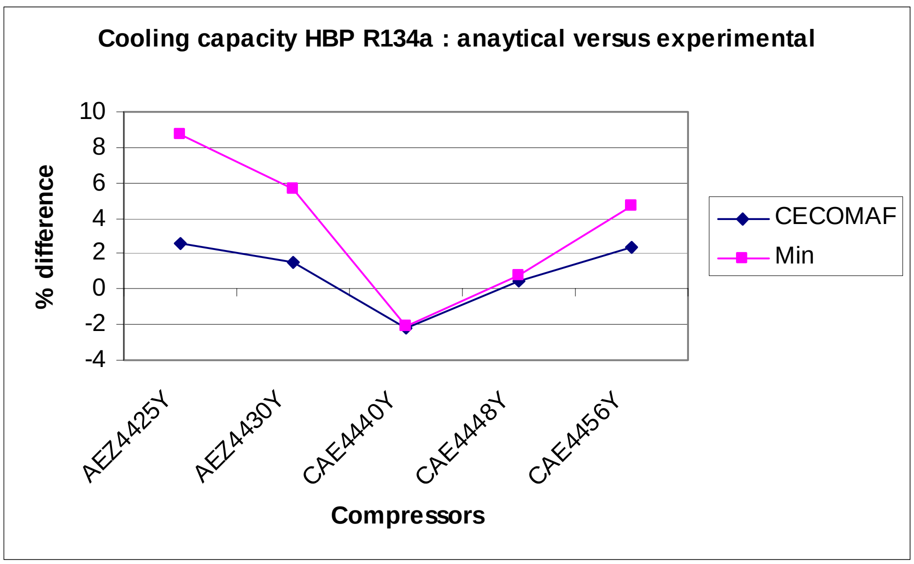
Where η_C is the compression efficiency, ratio between the theoretical power to the indicated power, η_{Mech} the mechanical efficiency, and η_{Motor} the motor efficiency.

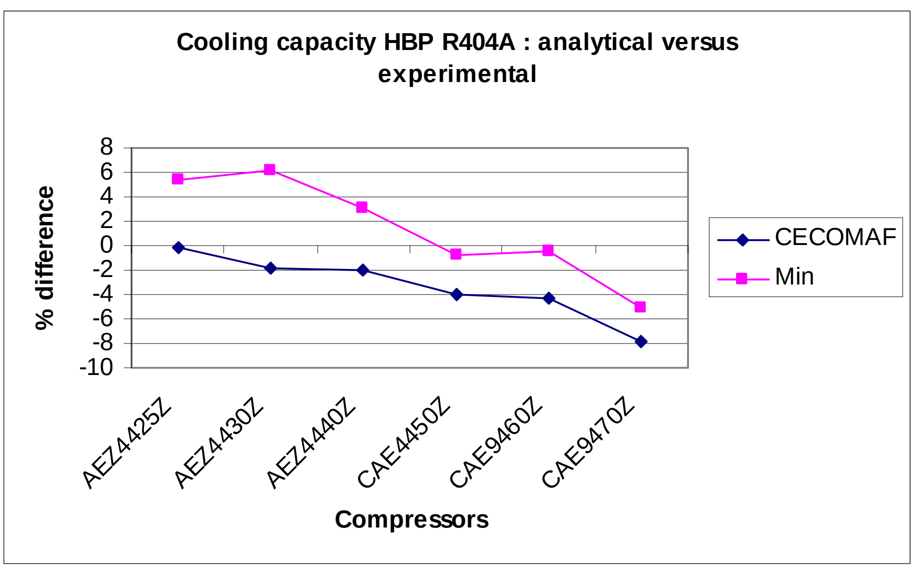
The calculation of the power input, and the coefficient of performance, require the knowledge of three new coefficients. Those values can be assumed or estimated thanks to software related to valves, bearing and motor. It is not the topic of this article to detail those coefficients. The formula is just recall here for memory.

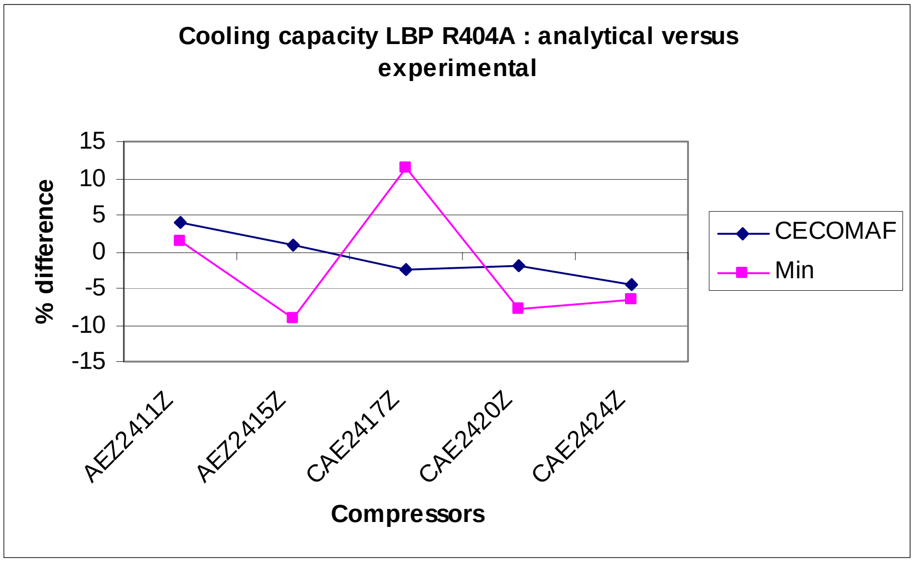
RESULTS AND DISCUSSION

The cooling capacity has been calculated thanks to the previous formulas described before. A comparison with experimental data has been led. Experimental data measured on calorimeter and analytical values obtained from calculation have been evaluated, and the percentage difference has been plotted.

Two operating conditions were chosen. The CECOMAF and a minimum point on the evaporating range both depending of the compressor applications studied, high back and low back pressure, for commercial application, for R134a and R404A. The agreement between experimental results and analytical calculation is shown below.







The percentage difference between analytical and experimental results stays within a range of around +/- 10%. This accuracy is a good one keeping in mind that cooling capacity of compressor on calorimeter is generally given at +/-5%.

There is no typical trend we can underline between analytical and experimental result. Cooling capacity gives a global view of results; more detail analysis should be done to get the individual effect of each formula used for the determination of the volumetric efficiency and the suction temperature, even if calorimeter just provides the mass flow as an output.

All analytical model need to make some assumptions to obtain a practical view of phenomena. The two formulas presented are certainly unperfected, being far from taking account of all the factors acting during the suction process, but they have the advantage to be simple to use, and allows to provide quick answer to designer.

Among the critics, which can be made to the formulas presented, attentive reader has probably noticed that the volumetric efficiency doesn't include the effect of the speed of the compressor. Since the very early approach of the volumetric efficiency [1], it is well known that the speed has influence: at low speed, the blow by due to cylinder-piston gap, and at high speed the valves back flow affects the volumetric value. Further investigation is required to include the effect of the speed in the volumetric efficiency formula, especially in case of variable speed compressor.

An other aspect which can be discussed in the formula of the suction gas temperature, is the use of an isothermal compression process instead of the usual polytropic one. This assumption leads to estimate the heat increase in the cylinder thanks to the useful power based on an isothermal compression, but allows to get very good result when comparing the gas temperature in the suction plenum obtained from calculation to values measured with thermo couple in the cylinderhead on calorimeter.

CONCLUSIONS

A method has been described to predict the performance of reciprocating hermetic compressors for refrigeration. Analytical formulas of the volumetric efficiency and suction gas temperature have been proposed. A comparison between experimental and analytical values of the cooling capacity on some compressors of the Company's author gives results within an accuracy, on calculation versus measurement, of around +/-10%.

The method presented can help the designer to estimate the cooling capacity at early step of his design process. Further developments need to be done to better take account of the influence of the speed of the compressor and improve the assumptions made on the gas behavior.

ACKNOWLEDGMENTS

The author acknowledges the helpful information on the volumetric efficiency given by Professor Maxime Duminil, Vice President of the French Refrigeration Association.

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