THERMODYNAMIC OPTIMIZATION OF CENTRIFUGAL COMPRESSOR FOR SMALL REFRIGERATION CAPACITY

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Abstract. *Centrifugal compressors are usually applied in large capacity refrigeration systems because they offer several technical advantages such as relatively low cost, low weight, low starting power requirements, and operating efficiency over a wide range of rotational speeds. Their primary drawback is that they cannot achieve the high compression ratio of reciprocating compressors without multiple stages. More recently, the possibility of using high speed motors has allowed the design of very compact centrifugal compressors for very small refrigeration capacities. However, some important technical issues include the development of high-speed, oil-free bearings and fabrication technologies for cost savings. The present paper considers an optimization of centrifugal compressor for small refrigeration capacity based on an integral thermodynamic approach. Results for the coefficient of performance, COP, are shown to allow an analysis of the influence of design parameters (inlet and outlet geometric elements, rotational speed and refrigerant fluid) on the compressor performance. An assessment of the optimized configuration is carried out with reference to three types of positive displacement compressors.*

Keywords*: centrifugal compressor, turbo machinery, refrigeration.*

1. INTRODUCTION

Air conditioning and refrigeration units are among the main sources of electrical energy consumption. For instance, in commercial buildings and industrial processes, such units can represent the largest operating cost. Environmental emissions associated to electrical energy generation contribute to global warming and climate change. Recent energy market deregulation and proposal of new standards for energy efficiency are setting new challenges on how organizations should use and manage energy resources. The global warming scenario is a strong incentive for improving the efficiency of equipments and for developing new refrigeration technologies.

Centrifugal compressors have been traditionally employed for high refrigerating capacities, above 500 kW. In fact, it is well established that centrifugal compressors are the best choice for large refrigeration applications. However, recently a two-stage air conditioning centrifugal compressor in the range between 60 and 90 kW, operating with R134a, has been lunched in the market, becoming the smallest commercially available centrifugal compressor for refrigeration purpose. The availability of new technologies, such as gas or magnetic bearing and high-speed electric motor, is making it possible to manufacture centrifugal compressors even for smaller sizes. For instance, Shiffmann (2006) presented a 6 kW centrifugal compressor prototype, with an impeller diameter of 20 cm, designed to operate with R134a as the refrigerant fluid and without lubricating oil. Pandy (1998) had showed that a centrifugal compressor, with a permanent magnetic motor, could achieve a coefficient of performance, COP, nearly 25% higher than the efficiency of any other type of compressor, when applied for a refrigerating capacity of 90 kW.

The most challenging aspect in designing a small centrifugal compressor is the extreme high velocity required to guarantee a satisfactory efficiency. In this respect, magnetic bearings, which have been proven in several applications, can allow a dramatic improvement in the performance of small centrifugal compressors by reducing mechanical friction. Moreover, the possibility of motor speed variation in centrifugal compressor can return optimal energy efficiency in a wide range of thermal load conditions. Conry (2002) showed the benefits of magnetic bearings and speed control in a 30 kW centrifugal compressor. The aforementioned technical issues can help the development of centrifugal compressors to refrigeration capacities much lower than one would imagine some years ago.

The present study considers a thermodynamic optimization of centrifugal compressors for small refrigeration capacities, based on an analytical simulation methodology. Results for the coefficient of performance, COP, are obtained to allow an analysis of the influence of design parameters on the compressor performance. An assessment of the optimized compressor geometry is carried out by comparing its efficiency with three types of positive displacement compressors: a reciprocating compressor, a scroll compressor and a rolling piston compressor.

2. COMPRESSOR DESIGN

According to Japikse (1996), a preliminary design of centrifugal compressors can be carried out through well-accepted rules derived from empirical relationships and the assumption of a one-dimensional flow condition. Theoretical analysis plays an important role in the centrifugal compressor design, allowing a parametric study of the influence of several parameters on the compressor performance under both design and off-design conditions. A reliable simulation tool can, therefore, decrease the cost and time of experimental development. Additionally, such models can also be used to predict the overall performance of the whole system, in which the compressor is one of the main components. Any methodology conceived to predict the performance of centrifugal compressors must account for losses in the compression process. A number of studies have analyzed the origin and the effect of losses in centrifugal compressors, proposing methods for their predictions.

2.1. Refrigerant fluids

The optimum rotational speed, N, of a radial compressor stage can be evaluating using the specific speed:

$$N_{ss} = \frac{N\,Q^{1/2}}{H^{3/4}} \tag{1}$$

where N_{ss} is the specific speed, H is the head and Q is the flow rate.

The desired pressure ratio and flow rate, together with the properties of the fluid (density, heat capacity, etc.), determine the rotational speed, N, of the compressor. Following Eq. (1), it can be shown that for small size compressors the rotational speeds becomes extremely high due to the small flow rate. Therefore, a suitable refrigerant fluid for small radial compressors should have low density and low enthalpy rise in order to keep the compressor speed at moderate values. These requirements for the refrigerant fluid are in most cases opposite to those that make a refrigerant optimal for a positive displacement compressor.

The adoption of conventional fluids developed for positive displacement compressors lead to high speeds and very small impellers. It is expected that low density refrigerants, such as R601a (iso-pentane) and R600a (iso-butane), would result relatively higher performance and moderate speeds, when compared with heavier refrigerants commonly adopted in positive displacement compressors, such as R290, R410a and R134a. As already mentioned, the compressor can operate totally oil-free if gas or magnetic bearings are used. This is very convenient design option not only due to the reduction of mechanical losses, but also because it allows a wider range of options for the refrigerant fluid, since no issue would arise about the chemical compatibility between the refrigerant and the lubricating oil.

2.1. Geometric Parameters

The impeller is a key component of centrifugal compressors, being responsible for transferring energy to the fluid. The velocity triangles is a convenient way of showing the relationship between all relevant velocity components, including absolute and relative velocities, and blade speed, as illustrated in Fig. 1. The relatives velocities, measured with respect to the rotating system, are detonated by W. The absolute velocities, which are taken with respect to a fixed coordinate system, are designated C. Relative flow angles are denoted by β and absolute flow angles by α. Naturally, a centrifugal compressor can be designed to have several impellers, according the required pressure ratio, forming different compression stages. Additional information on vector diagrams on parameters involved in their construction can be obtained in Japikse (1996).

A number of parameters must be carefully considered in the impeller design, including the approach relative flow velocity, W_1, the incidence angle, β_1, and a flow prewhirl, $C_{\theta 1}$. After deciding the system operating condition, fluid refrigerant and capacity requirement, the first step is to design the impeller eye through an optimization study based on the inlet velocity triangle. The inlet tangential flow velocity, $C_{\theta 1}$, can be expressed as:

$$C_{\theta 1} = C_{m1} \tan \alpha_1 \tag{2}$$

where C_{m1} is the inlet radial velocity, for which a guess value must be prescribed. Then, the absolute inlet velocity, C_1, can be obtained as follows:

$$C_1 = (C_{m1}^2 + C_{\theta 1}^2)^{1/2} \tag{3}$$

By knowing the required mass flow rate, $\dot{m}$ and the inlet radial velocity, C_{m1}, the flow area necessary for the impeller eye can be evaluated from a simple mass flow rate relationship:

$$A_f = \dot{m} / [\rho_1 C_{m1} (1 - B_1)] \tag{4}$$

where B_1 is an area blockage coefficient of inlet configuration, which can be obtained from empirical data according to the inlet geometry.

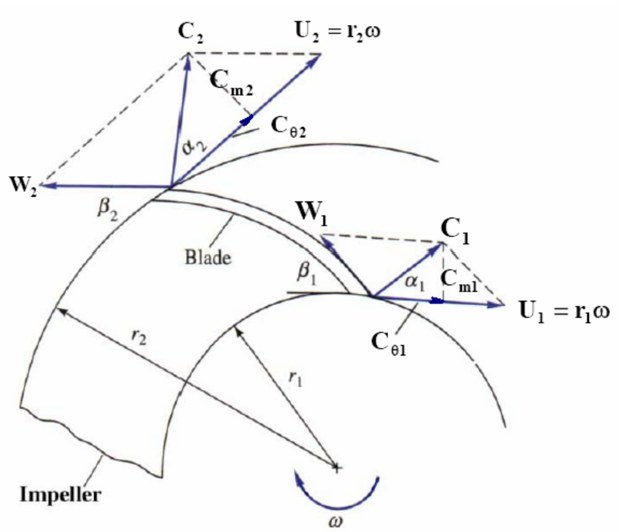


Figure 1. Velocity vector diagrams of a centrifugal impeller.

Based on the flow area, A_f, and the hub radius of the impeller eye, r_{1h}, a first estimate for the tip radius of the impeller eye, r_{1t}, can be obtained from:

$$r_{1t} = \left\{ \frac{A_f}{\pi[1-(r_{1h}/r_{1t})^2]} \right\}^{1/2} \tag{5}$$

As can be seen in Eq. (5), an initial guess for r_{1t} is required. The blade tangential velocity at the tip edge can be obtained from the angular velocity N [rpm], and than the inlet relative flow velocity, W_{1t}, can too be expressed:

$$U_{1t} = 2\pi r_{1t} N \tag{6}$$

$$W_{1t} = \sqrt{(U_{1t} - C_{\theta 1})^2 + C_{m1}^2} \tag{7}$$

A good design of the centrifugal compressor should return a small value for W_{1t}. Because the inlet radial velocity, C_{m1}, is just an initial guess, a geometric optimization is required for a given mass flow rate, so as to find the most efficient stage design. In summary, an iterative procedure is adopted to seek values of C_{m1} that minimizes W_{1t}.

The inlet angle of the blade is evaluated by:

$$\beta_1 = \tan^{-1}[(U_{1t} - C_{\theta 1})/C_{m1}] \tag{8}$$

Thus, using the pressure ratio requirement, pr_{st}, the diffuser pressure recovery coefficient, C_{PD}, the impeller mechanical efficiency, η_{mec}, and the swirl parameter, λ_2 (tan α_2), the impeller outlet conditions can be evaluated.

Considering Newton Second Law of Motion applied to a rotating system, the torque developed for a constant mass flow rate is equal to the change of angular momentum:

$$\tau = \dot{m}\,\Delta(rC_\theta) = \dot{m}\,(r_1 C_{\theta 1} - r_2 C_{\theta 2}) \tag{9}$$

The rate of energy transfer per unit of mass flow rate, W, is the product of the torque and angular velocity, $\dot{\omega}$:

$$W = \tau\,\dot{\omega}/\dot{m} = \dot{\omega}\,(r_1 C_{\theta 1} - r_2 C_{\theta 2}) \tag{10}$$

It should be noticed that $r\,\dot{\omega}$ corresponds to the blade speed U at a certain radius position. Also, the energy equation states that the work done per unit of mass flow is equal to the change in total enthalpy of an adiabatic process. Introducing the stage efficiency, η_{st}, Eq. (10) can be rewritten to evaluate the effective work input, W_{ef}, necessary for the corresponding stage:

$$W_{ef} = (U_1 C_{\theta 1} - U_2 C_{\theta 2})/\eta_{st} = \Delta h_{0s}/\eta_{st} \tag{11}$$

In the previous equation, the stage efficiency, η_{st}, is a variable whose value must be found through an iterative procedure, as will be explained. The stage efficiency includes the effect of several phenomena that reduce the impeller efficiency, such as flow slip, recovery pressure in the diffuser and mechanical losses.

For an isentropic process, the enthalpy increase in a certain stage is obtained from:

$$\Delta h_{0s} = \frac{kRT_{\infty}}{k-1}(pr_{st}^{(k-1)/k} - 1) \tag{12}$$

which can be used with the Eq. (11) to evaluate the impeller outlet velocity of a corresponding stage as:

$$U_2 = \left[(U_1 C_{\theta 1} + W_{ef})/\mu\right]^{0,5} \tag{13}$$

where μ is the work input coefficient, which can readily be related to the slip factor coefficient, σ, to be defined in Section 3. Through the velocity diagram at the impeller outlet, the following equation is obtained:

$$\mu = \sigma\,\lambda_2 /(\lambda_2 - \tan\beta_2) \tag{14}$$

According with Japikse (1996), the work input coefficient μ is related to U_2 by:

$$C_{\theta 2} = U_2 \mu \tag{15}$$

The impeller diameter at the outlet is determined by:

$$D_2 = 60\,U_2 / \pi N \tag{16}$$

and, hence, the radial velocity components can be obtained as follows:

$$C_{m2} = C_{\theta 2} / \lambda_2 \tag{17}$$

Consequently, it is possible to find the outlet flow area, A_2, and the blade height, b_2, as:

$$A_2 = \dot{m} / \rho_2 C_{m2}\,; \qquad b_2 = A_2 / \pi D_2 \tag{18}$$

In order to evaluate the pressure at diffuser outlet, it is introduced the impeller mechanical efficiency, η_{mec}, and the diffuser pressure recovery coefficient, C_{PD}:

$$p_{02} / p_{\infty} = \{[(k-1)(W_{ef}\eta_{mec} / kRT_{\infty})] + 1\}^{k/(k-1)} \tag{19}$$

$$p_5 = p_2 + C_{PD}(p_{02} - p_2) \tag{20}$$

Finally, the stage efficiency is evaluated according to:

$$\eta_{st} = \frac{(p_5 / p_{\infty})^{k/(k-1)} - 1}{(T_{02} / T_{\infty}) - 1} \tag{21}$$

Because the impeller efficiency η_{st} was initially prescribed, it is necessary to return to Eq. (11), replace the value for η_{st} and repeat again all the calculations in an iterative manner, until a converged value for η_{st} is found. After convergence is reached, new values for the hub radius and the tip radius of the impeller eye can be tested, in order to find the best design for the centrifugal compressor.

Finally, other parameters that affect the stage efficiency (electrical motor losses, flow regime in the impeller, leakages through clearances, rear disk friction) are included to evaluate the overall stage efficiency:

$$\eta = \eta_{st}\,\eta_{el}\,\eta_{Re}\,\eta_{cl}\,\eta_{rf} \tag{22}$$

3. THERMODYNAMIC AND MECHANICAL LOSSES

3.1. The slip factor

Even under ideal conditions, the velocity at the exit of the impeller would not be perfectly aligned with the vanes. Such a flow misalignment is commonly referred as slip velocity or just slip. If the impeller could be imagined as being made with an infinite number of infinitesimally thin vanes, then an ideal flow would be perfectly guided by the vanes and would leave the impeller at the vane angle.

The slip factor is a vital piece of information needed in the compressor design, since it allows an estimate of the energy transfer between impeller and fluid. There are a variety of slip factor correlations available and details can be obtained in many references. The slip factor correlation proposed by Wiesner (1967) is given by:

$$\sigma = 1 - \sqrt{\cos\beta_2} / Z_R^{0.7} \tag{23}$$

However, this correlation applies only to impellers where the following geometric relationship is satisfied:

$$\bar{r}_1 / r_2 < \exp(-8.16\cos\beta_2 / Z_R) \tag{24}$$

where Z_R is the number of blades.

3.2. Reynolds number

The effect of the Reynolds number on the compressor performance has received considerable attention in the literature. Compressors are subjected to losses due to shear stresses and the level of such losses is strongly affected by the extent to which the flow is laminar or turbulent.

If the flow remains laminar, the shear stresses are low and the associated loss by viscous friction will be corresponding small. However, in such situations there will probably be extensive regions of separate flow, leading to lower pressure rise and lower efficiency than otherwise would be achieved with a turbulent flow.

To describe the influence of the flow regime on the compressor efficiency, Wiesner (1979) and Casey (1985) have proposed the following empirical correlation:

$$\frac{1-\eta_{Re}}{1-\eta_{ref}} = a + (1-a)(Re_{ref} / Re)^n \tag{25}$$

where a and n are derived from experimental data according to the Reynolds number at the trailing edge:

$$Re = C_2 b_2 \rho_2 / \mu_2 \tag{26}$$

3.3. Blades

The specification of the number of blades is an important step in the compressor design. Several empirical correlations, based on geometric parameters, are available for this purpose but were prepared with reference to experimental data of large compressors. In spite of that, the correlation proposed by Pfleiderer (1960) has been seen to be satisfactory also for low capacity compressors and is given by:

$$Z_R = 6.5\left(\frac{D_2 + D_1}{D_2 - D_1}\right)\operatorname{sen}\beta_m \tag{27}$$

Use of partial blades or splitters is a very common design practice but without a solid technical criterion. It is generally recognized, and confirmed in numerous investigations, that higher mass flow rate can be obtained by reducing flow blockage in the impeller with the use of splitters. In this paper, splitters have been considered in the design procedure. In addition to that, the blade thickness was also a parameter included in the analysis.

3.4. Clearance effect

For the unshrouded impeller, it is important to recognize the influence of flow leakage. Senoo and Ishida (1987) developed a simple model for the losses caused by leakage:

$$\Delta\eta_{cl} / \eta = (t / b_2) / 4 \tag{28}$$

According to Cumpsty (2004), remarkably good predictions of the loss have been verified for a number of compressors.

3.5. Rear disk friction

The viscous friction caused by the presence of the rear disk is also information required to evaluate the compressor efficiency. Japikse (1996) points out that the most common approach to estimate the rear disk friction loss, W_f, is the proposal of Daily and Nece (1960):

$$W_f = C_m \rho_2 (U_2 / r_2)^3 r_2^5 / 4\dot{m} \tag{29}$$

where the radial velocity, C_m, and the Reynolds number, Re, are defined by:

$$C_m = 0.0402 / Re_D^{1/5} \quad ; \qquad Re_D = U_2 r_2 \rho_2 / \mu_2 \tag{30}$$

It must be pointed out that no backflow was assumed at the impeller outlet.

3.6. Diffuser

This is a component designed to increase the fluid pressure by reducing the flow velocity. The basic diffuser is a geometrically simple device with a rather long history of investigation by many researchers. According with Cumpsty (2004) a well-design diffuser has a typical value of pressure recovery, C_{PD}, of 70%. In this study, a conservative value of 65% has been adopted for C_{PD}.

4. RESULTS

The assessment of centrifugal compressor for small refrigerating capacities is only possible after an adequate design optimization, including a number of parameters: i) refrigerating system specification (fluid refrigerant, system operating condition, refrigerating capacity); ii) geometric parameters (inlet radius, clearance gap, etc); iii) Target function (in the present study represented by the coefficient of performance COP) and iv) constraints (associated to the geometry and fluid flow). In this investigation, a simulation methodology for the compressor was developed in which the commercial code modeFRONTIER (2005) was adopted for the optimization process. Several optimization algorithms are available, but in this study the genetic algorithm was chosen.

The choice of a fluid refrigerant is dictated by environmental aspects (greenhouse effect, ozone layer depletion) and thermodynamic properties. The latter define the pressure ratio and the mass flow rate required in the system, and have a considerable influence on the COP. In the case of the centrifugal compressor, the refrigerant also affects the compressor size and rotational speed. The refrigerants included in the present analysis were R134a, R410a, R290 (propane), R600a (iso-butane) and R601a (iso-pentane). Synthetic refrigerants such as the R134a do not have any ozone depletion potential but, on the other hand, have a large greenhouse effect when liberated into the atmosphere. Natural refrigerants, such as R290, R600a and R601a are environmentally friendly, but are flammable. An interesting alternative could be CO_2, but its use implies in high pressure levels, resulting in large forces on the bearings and very small compressor impellers (Schiffmann, 2006).

The overall COP is affected by losses in bearings and in the electrical motor. Currently, it is possible to design have bearings (gas or magnetic) with efficiency of up to 90%. High efficient electrical motors are also available and in this work a typical value of 93% was considered. The tip clearance between impeller and shroud of 15 μm is consistent with the current machining techniques (Schiffmann, 2006).

The system operating condition chosen for the analysis corresponds to T_{evap} = 7.2 ^oC and T_{cond} = 54.4 ^oC, associated to a HBP application under ASHRAE condition. In order to keep the Mach number, pressure ratio and, in some cases, impeller stress bellow critical values, a two stage compressor was selected.

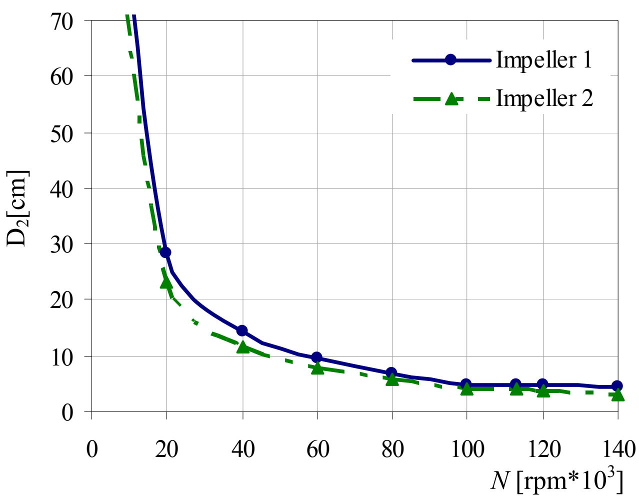
As already mentioned, a suitable refrigerant for centrifugal compressors should have low density and low specific heat capacity, which reflects on the enthalpy increase, so that the compressor can operate at a moderate speed. Table 1 presents a summary of operating conditions associated with different refrigerants, considering a refrigerating capacity of 8.6 kW (HBP condition).

It is very clear that the less dense fluid (R601a) is the best choice, because it allows the best efficiency (COP) and the lowest rotational speed combined with the greatest diameter. A larger diameter is convenient for the impeller manufacturing. As can be seen, higher density fluids require a higher tip speed at the impeller outlet (i.e., higher rotational speed for the compressor) or the adoption of larger diameter impellers.

Table 1. Compressor operating conditions according with the fluid refrigerant, considering a refrigerating capacity of 8.6 kW (HBP condition).

Fluid	Δh [kJ/kg]	ρ_1[kg/m^3]	N [rpm]	Q [m^3/s]	D_2[cm]	COP
R601a	315.38	1.34	131,000	0.02038	4.18	3.77
R600a	299.75	4.78	230,500	0.00600	2.42	3.50
R134a	162.80	16.2	195,000	0.00327	2.17	3.20
R290	308.28	11.1	318,500	0.00252	1.89	3.06
R410a	176.24	32.4	271,500	0.00151	1.64	2.61

Figure 2 shows results for the impeller diameter, COPpv (thermodynamic coefficient of performance) and COP according to the compressor rotational speed, considering an 11.6 kW HBP compressor operating with R601a. The results show that a high speed impeller condition is required for a combination of adequate efficiency and small impeller dimensions. Such rotational speeds can be achieved by using high-speed electric motors, and gas or magnetic bearings, as discussed by Schiffmann (2006).



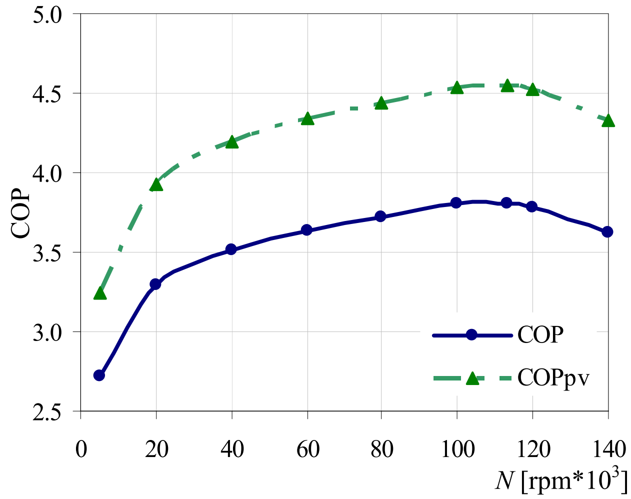


Figure 2. COP and diameter requirements versus rotational speed for an 11.6 kW R601a centrifugal compressor

According to turbomachinery design theory, a high level of efficiency can be obtained for centrifugal compressors at high specific speed, N_{ss}, as defined in Eq. (1). For a certain optimum value of specific speed, Eq. (1) shows that the compressor rotational speed must increase if the volume flow rate is decreased, so as to maintain the same value for N_{ss}. This explains why the efficiency (COP or COPpv) for a determined capacity decreases with a reduction in the compressor speed (Fig. 2.b).

An interesting fact, also associated with the specific speed, N_{ss}, is that low density refrigerants, such as the R601a, are more suitable for centrifugal compressors, allowing a higher efficiency level for the compressor, as illustrated in Fig. 3. In other words, for a fixed refrigerating capacity a low density fluid will be associated with a higher volume flow rate. For instance, a 17.6 kW capacity compressor operating with R601a has COP about 8% greater than one using R600a, and 44% more than other employing R410a. It becomes clear in Fig. 3 the increase in compressor efficiency with the compressor cooling capacity, mainly because the clearance losses are proportionally smaller for large impellers.

In terms of compressor speed, Fig. 4 shows that R601a and R134a are the best options, because they can maintain the compressor speed at a moderate level. As can be observed in Fig. 2, it is possible to operate the compressor at lower speeds by allowing a small decrease in the compressor efficiency. According to Fig. 3, the refrigerant R600a performs better than R134a but implies a higher compressor speed (Fig. 4).

Usually, low density refrigerants also imply in larger diameter impellers, which are less affected by leakages. However, this is not always the case as can be seen in Fig. 5, where the impeller diameter required by the R290 is smaller than that necessary for the higher density R134a. The aerodynamic demand for small clearance becomes difficult to concede for small compressor sizes, requiring a very fine machining or precision casting process.

When the centrifugal compressor is compared with positive displacement compressors (Tab. 2), it is seen that a suitable refrigerant fluid is a key aspect necessary to make the centrifugal compressor more efficient. For instance, the refrigerants R410a and R290 are not recommendable for centrifugal compressors. However, whereas the R601a is not suitable choice for positive displacement compressors, since its low density brings about a poor volumetric efficiency, it allows the design of a high efficient centrifugal compressor.

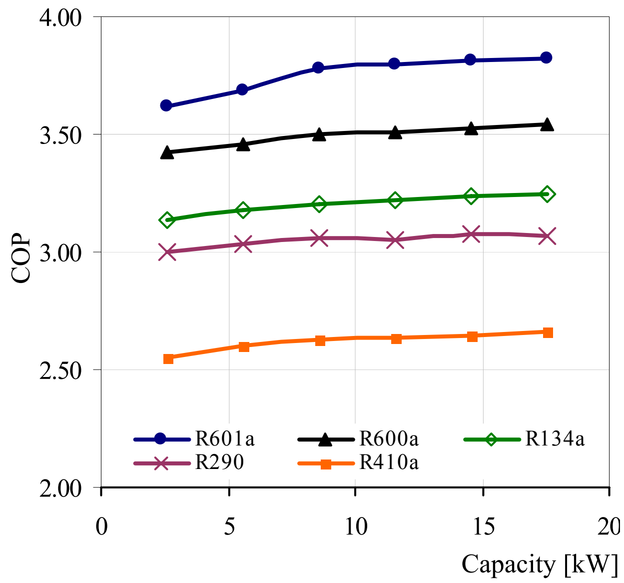


Figure 3. Coefficient of performance versus capacity for different refrigerants fluids

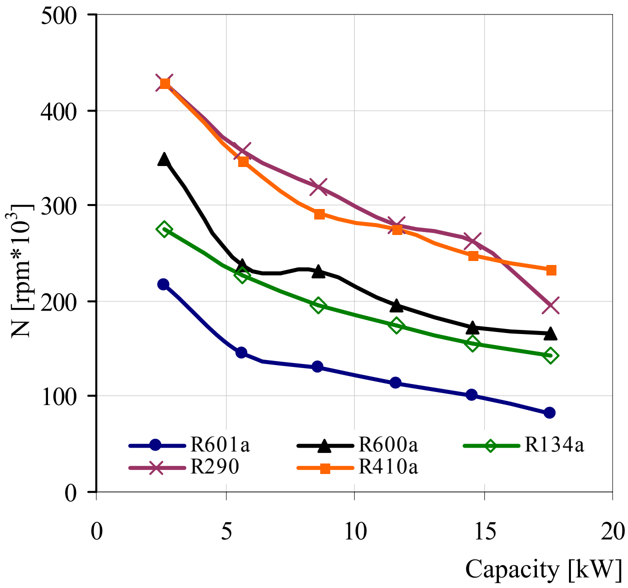


Figure 4. Rotational speed requirements versus capacity for different refrigerants fluids

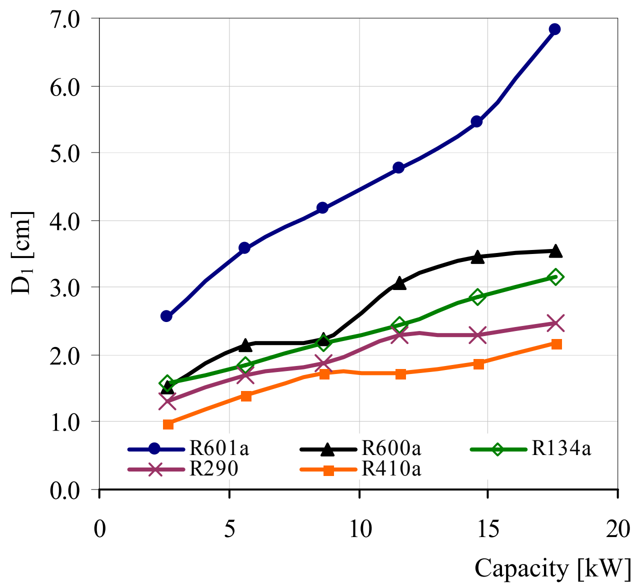


Figure 5. Diameter at the stage 1 outlet versus capacity

Table 2. Coefficient of Thermodynamic Performance (COPpv), for different compressors and fluid refrigerant, considering a cooling capacity of 17.6 kW.

Mechanism	Refrigerant		
	R290	R410a	R601a
Reciprocating	3.63	3.82	-
Scroll	3.75	3.24	-
Rollin Piston	3.80	3.13	-
Centrifugal	3.67	3.03	4.57

It is also possible to compare results for the overall efficiency (COP) of the centrifugal compressor with other types of compressor commercially available, considering two refrigerating capacities: 1.3 and 14.6 kW. For the smaller capacity, the choice for comparison was a reciprocating compressor operating with R134a. The centrifugal compressor was simulated with R601a as the refrigerant and the resulting COP was 3.43, approximately 24% higher than the value of 2.77 obtained for the reciprocating compressor. In the case of the 14.6 kW, the most adequate compression technology for the centrifugal compressor is the scroll compressor. From data made available by scroll manufacturers, it has been verified that a R22 scroll compressor had a COP of 3.46. However, even for this case, an R601a centrifugal compressor can offer a COP of 3.82, a value 10% higher than that of the scroll compressor.

5. CONCLUSIONS

The present investigation considered an analysis of centrifugal compressor applied to small refrigerating capacities. The use of centrifugal compressors offers several advantages over positive displacement compressors, such as higher energy efficiency, smaller size and weight, reduced levels of mechanical vibrations and flow pulsations, oil-free operation and great flexibility for capacity control. The main challenge of designing a centrifugal compressor for small refrigerating capacity is associated with the high rotational speed and small impeller required, leading to small efficiency and mechanical difficulties. However, if efficient electric motors and bearings can be supplied to high speed compressors at affordable costs, centrifugal compressors may prove to be competitive even for small capacities.

The results obtained in the present investigation show that the compressor speed is greatly affected by the refrigerant fluid adopted, with the most moderate speed being achieved with R601a and R134a. The investigation also demonstrated that the efficiency of small centrifugal compressors is significantly affected by gas leakage through the compressor clearances. For the conditions investigated, the centrifugal compressor operating with R601a was found to be thermodynamically more efficient than any other type of positive displacement compressor.

6. ACKNOWLEDGEMENTS

This study is part of a technical-scientific program between the Federal University of Santa Catarina and EMBRACO. Support from FINEP (Federal Agency of Research and Projects Financing) and CAPES (Coordination for the Improvement of High Level Personnel) is also acknowledged.

7. REFERENCES

Casey, M.V., 1985, “The Effects of Reynolds Number on the Efficiency of Centrifugal Compressor Stages”, ASME, J. of Engineering for Gas Turbines and Power, Vol. 107, pp. 541-544.

Conry, R., Whelan, L., Ostman, J., 2002, “Magnetic Bearings, Variable Speed Centrifugal Compression and Digital Controls Applied in a Small Tonnage Refrigerant Compressor Desing”, Proc. Int. Compressor Engineering Conf. at Purdue, USA, Paper C2-1.

Cumpsty. N.A., 2004, “Compressor Aerodynamics”, Ed. Krieger, USA, 517 p.

Daily, J.W., Nece, R.E., 1960, “Chamber Dimension Effects on Inducer Flow and Frictional Resistance of Enclosed Rotating Disks”, ASME J. Basic Engineering, Vol. 59, pp. 217-232.

Japikse, D., 1996, “Centrifugal Compressor Design and Performance”, Ed. Concepts ETI, USA, 424 p.

modeFRONTIER, ESTECO, Version 3.1.0, 2005.

Pandy, D.R., Brondum, D., 1998, “Innovative, Small, High-Speed Centrifugal Compressor Technologies”, Proc. Int. Compressor Engineering Conf. at Purdue, USA, pp. 913-918.

Pfleiderer, C., 1960, “Bombas centrífugas y turbocompresores”, Editorial Labor, S. A. 452 p.

Schiffmann, J., Favrat, D., 2006, “High-Speed Low Power Radial Turbo Compressor for Oil-Free Heat Pumps”, Proc. International Compressor Engineering Conf. at Purdue, USA, Paper C-114.

Senoo, Y., Ishida, M., 1987, “Deterioration of Compressor Performance Due to Tip Clearance of Centrifugal Impellers”, ASME, Journal of Turbomachinery, Vol. 109, pp. 55-61.

Wiesner, F.J., 1967, “A Review of Slip Factors for Centrifugal Compressors”, ASME J. Engineering for Power, Vol. 89, No 4, pp. 558-572.

Wiesner, F.J., 1979, “A New Appraisal of Reynolds Number Effects on Centrifugal Compressor Performance”, ASME J. Engineering for Power, Vol. 101, pp. 384-396.

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