Performance Analysis of Refrigerant Centrifugal Compressor

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Abstract: ***In this work, a performance analysis of the compression cycle of a centrifugal compressor for domestic refrigeration was carried out. A numerical simulation model of a centrifugal compressor has been developed by using MATLAB Simulink R2009a. An advantage of numerical simulation model of a centrifugal compressor is for the design, development, improvement and optimization of the elements constituting the compressor circuit. The actual working condition of a centrifugal compressor, different losses such as in incidence loss, friction loss, pressure drop, leakage losses etc. is also incorporated in the simulation model of the compressor. Result presented show the influence of different aspects (speed, geometry, working conditions) on the basis of meaningful non-dimensional parameters, which describe the compressor behavior. The refrigerants taken for the analysis in this paper is R134a, but any refrigerants would be considered for the analysis. The idea of this paper is to give a better understanding of the performance behavior under different aspects to improve the design of these equipment.***

Keywords: Centrifugal Compressor; Numerical Simulation Model; Calculation; Parameter; Performance Analysis; Geometry

1. Introduction

A vapour compression refrigeration system comprised of four components: compressor, evaporator, condenser and expansion device. Out of which, the most power consumption device is a compressor. Centrifugal compressors are widely used in refrigerators, residential air conditioners etc. The advantage of centrifugal compressor over the reciprocating compressor are high efficiency over a large range of a load and a large volume of the suction vapour and hence a larger capacity for its size. The centrifugal compressor has also many other advantageous features. The most important is the flat head capacity characteristic as compared to that of a reciprocating compressor.

Analytical performance prediction method plays an important role in designing a centrifugal compressor by way of predicting the overall dimensions and performance curve of the stage. First of all, the analytical method can be used to perform parametric studies to demonstrate the influence of changes in geometry on the performance under both design and off-design conditions. Therefore, the availability of reliable analytic tool saves expensive experimental development. In addition, analytic models for predicting the overall performance of the compressor can be effectively used in predicting the overall performance of a combination system where the compressor is a component of such a system.

In order to derive the analytical tools, the losses occurring throughout the stage must be specified. The origin and effects of loss mechanism were discussed in details in references [4–6]. These papers presented prediction methods based on modeling various losses throughout the stage. According to references [1,2,7], the losses in the centrifugal compressor are classified into incidence loss, friction loss, clearance loss, backward loss and volute loss. Reference [3], presents comparative study of the existing modeling techniques and their limitations for predict-ing incidence loss. There are two widely used models: one is the constant pressure incidence model and the other is the NASA shock loss theory as described in references [2, 3]. In the current study, the NASA shock loss theory is used. It can be mentioned here that very little difference exists between the two models for centrifugal compressor as expounded in reference [1]. As to the friction loss, references [2, 3] apply the energy and momentum equations to pipe flow with surface frictions to get the loss coefficient related with Reynolds number. For this study, we used the approach of references [2, 3] to predict the friction loss.

In this paper, a specific numerical simulation model of the centrifugal compressor has been developed. With the help of different references mentioned in this paper and by considering different losses a better simulation model has been developed. And a closed analysis has been done by drawing the graphs between different parameters.

Nomenclature	
m	Mass flow rate (Kg/s)
Tc	Condenser Temperature (^{0}C)
Te	Evaporator Temperature (^{0}C)
Pe	Evaporator Pressure (bar)
Pc	Condenser Pressure (bar)
Q	Refrigerating effect (KW)
D	Mean diameter (m)
L	Chenal length (m)
u	Mean blade velocity (m/s)
u1	blade velocity at inlet (m/s)
u2	blade velocity at outlet (m/s)
Δh_f	Friction loss(KJ/Kg)
H	Enthalpy of suction vapour
Wc	Compressor work (KW)
N	Speed of the compressor (r.p.m)
v	Specific volume (m³/Kg)
hi	Incidence loss
β_2	Blade angle
η_p	Polytropic efficiency
γ	Specific heat ratio
ρ	Density (kg/m3)
τ	Torque(Nm)
H	Head in (kg.m/s2)

2. Mathematical Model

The mathematical model of a centrifugal compressor, which consists of various thermodynamic relations. For the analysis following assumptions are made:

1. Mean diameter (D) = .4 m
2. Chenal length of (L) = .3 m
3. Evaporator temperature (Te) = 6^0C
4. Condenser temperature (Tc) = 40^0C
5. Blade angle β_2 = 30^0,32^0,34^0,36^0
6. Isentropic index (γ) = 1.118 (for R-134a)
7. Speed of the compressor (N) = 2000 r.p.m, 2500 r.p.m, 3000 r.p.m, and 3500 r.p.m.

2.1 Thermodynamic relations used

2.1.1 Losses Considered

Following are the losses which had considered in the simulation model of compressor

2.1.2

a Leakage loss: The effect of leakage elements is normally accounted for by allowing 1.5 per cent leakage per unit of the compression ratio r, which is equal to P_2/P_1. Old worn out compressors tend to have more leakage and hence they lose their cooling capacity.

b Incidence loss: Incidence loss comes from the off-design velocity triangles at the impeller eye(inducer) causing flow separation and are therefore at the design point of view this loss is zero.

c Friction loss: According to the reference [2], the friction loss in the impeller can be define as $\Delta h_f = km^2$ Where k is the constant $k = \frac{u_2 l}{2D\rho^2 A^2 sin^2 \beta_2}$. Where l is the mean channel length and D is the mean hydraulic channel diameter.

d Other loss: According to the reference [2], the clearance loss is a function of the clearance to passage width at the tip. For back flow loss no theory or mathematical modal exist at present to describe the backflow loss.

3. Analysis of the Results

Based on the above simulation model, various graphs has been plotted by varying different parameters, the graphs which are plotted are based on the following parameters concerned -:

1. By varying the speed of the compressor at fixed evaporator temperature.(i.e.,2000rpm,2500rpm, 3000 r.p.m,3500r.p.m,)
2. By varying the outlet blade angle at fixed evaporator temperature (i.e.,30^0,32^0,34^0,36^0)

All numerical experiments have been performed assuming a single stage vapour compression low pressure cycle at the following conditions: outlet compressor temperature and inlet condenser temperature of 40^0C

3.1 By varying the speed of the compressor at fixed evaporator temperature (Compressor Speed influence)

Fig.3.1 present outlet pressure versus mass flow rate characteristics obtained from graph under the different rotational speed N (2000rpm, 2500rpm, 3000rpm, and 3500rpm). The observations from the figure are listed as follows;

- The outlet pressure is a function of the mass flow rate and rotational speed N. Therefore, the desired outlet pressure can be achieved by regulating either the mass flow rate or the rotational speed or both of them.
- The rotational speed imposes a much bigger effects on the outlet pressure than that of the mass flow rate. As a result adjustment of the rotational speed works more favorably for the macro adjustment of the outlet pressure while the regulating of the mass flow would be preferable for the micro adjustment of the outlet pressure.

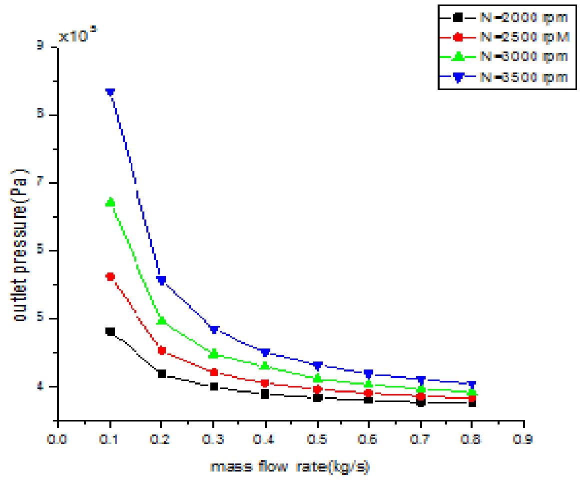

Figure 3.1: outlet pressure versus mass flow rate with variable rotational speed.

3.2 Friction loss versus mass flow rate with variable rotational speed at fixed outlet blade angle.

Fig.5.2 depicts the friction loss versus mass flow rate characteristics obtained from graph under the various rotational speeds N (2000rpm, 2500rpm, 3000rpm, 3500 rpm). The overlapping of all the curves indicates that frictional loss is a function of mass flow rate only and that the change of the rotational speed has no bearing on the friction loss.

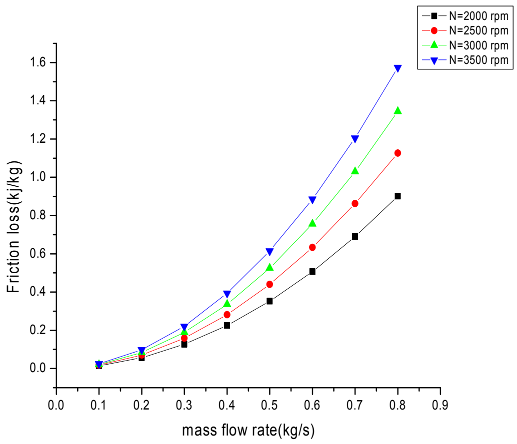

Figure 3.2: Friction loss versus mass flow rate with variable rotational speed at fixed outlet blade angle.

3.3 Friction loss versus mass flow rate with fixed rotational speed by varying outlet blade angle.

Fig.3.3 depicts the friction loss versus mass flow rate characteristics obtained from graph under the fixed rotational speeds N (2000rpm) by varying outlet blade angle. The overlapping of all the curves indicates that frictional loss is a function of mass flow rate only and that the change of the outlet blade angle has no bearing on the friction loss.

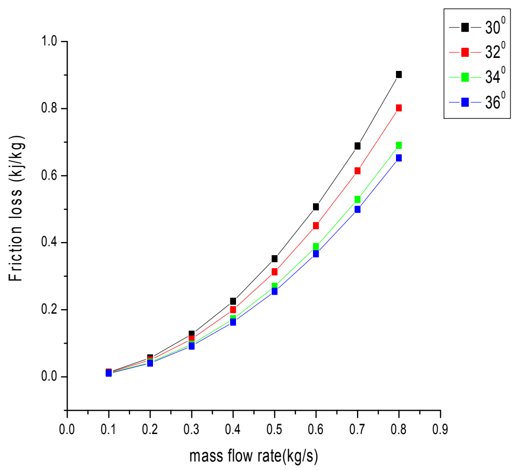

Figure 3.3: Friction loss versus mass flow rate with variable rotational speed at fixed outlet blade angle.

3.4 Calculating head by varying outlet blade angle at constant rotational speed (2000 rpm).

The head versus mass flow rate characteristic obtained from graph under the various outlet blade angle at constant rotational speed. The graph shows that, the head of compressor decreases by increasing the mass flow rate but b varying the outlet blade angle at same rotational speed N(2000 rpm) initially head become high but slowly they are decreases. From the graph at 30^0 blade angle the head is low with compare to the other angles i.e. 32^0, $34^0$$36^0$. At same rotational speed head increases with increasing the outlet blade angle.

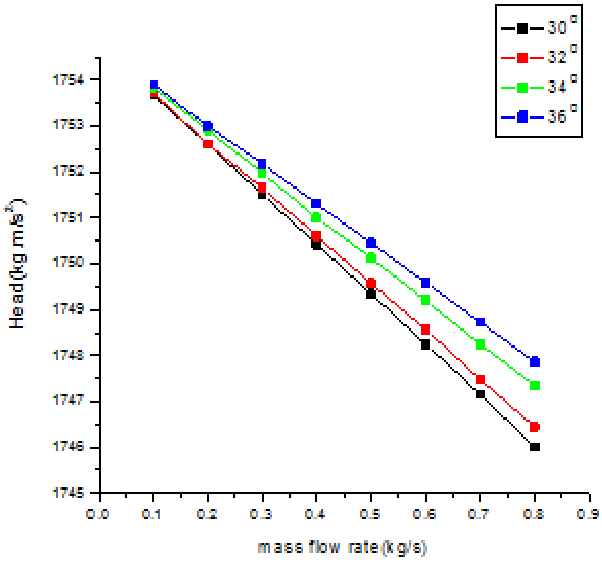

Figure 3.4: Head versus mass flow rate with variable outlet blade angle with constant rotational speed.

3.5 Calculating head by varying outlet blade angle at constant rotational speed (2500 rpm).

Again the head versus mass flow rate characteristic obtained from graph under the various outlet blade angle at constant rotational speed. The graph shows that , the head of compressor decreases by increasing the mass flow rate but b varying the outlet blade angle at same rotational speed N(2500 rpm) initially head become high but slowly they are decreases. From the graph at 30^0 blade angle the head is low with compare to the other angles i.e. 32^0 ,$34^0$$36^0$. At same rotational speed head increases with increasing the oulet blade angle.

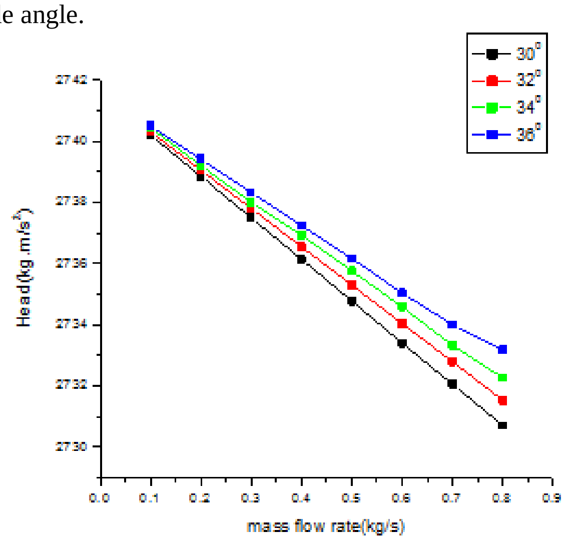

Figure 3.5: Head versus mass flow rate with variable outlet blade angle with constant rotational speed.

3.6 Observation of Efficiency by varying rotational speed at fixed outlet blade angle.

The another analysis of the paper is efficiency versus mass flow rate characteristics obtained the analysis under the different rotational speed N (2000rpm, 2500rpm, 3000rpm, 3500rpm). The observations are listed as follows;

- The optimum mass flow rate i.e best efficiency of the compressor change a function of rotational speed.
- The optimum mass flow rate increases as the rotational speed goes up.
- Under the specific rotational speed in case that the mass flow rate is beyond the optimum value, the efficiency of the compressor begins to decrease as the mass flow rate increases, and this phenomenon gets more pronounced at lower rotational speed.

4. Conclusions

An advanced numerical simulation model of refrigerant centrifugal compressor has been used in a wide range of geometries and working conditions. A parametric study of centrifugal compressors behavior has been presented based on the numerical simulation mentioned above. The simulation model was able to evaluate the mass flow rate (m), head (H), friction loss, incidence loss, polytropic efficiency (η_p) and outlet pressure of the whole refrigeration system based on the specific input parameters and varying. By varying the mass flow rate outlet pressure of compressor decreases, tme mass flow rate and rotational speed much affected the outlet pressure. By varying the mass flow rate with different rotational speed the friction loss increases in quadratic form, and again the mass flow rate varying with different blade angle , friction loss again in that same manner. The head of the compressor decreases with increasing the mass flow rate by varying either rotational speed or outlet blade angle it means for optimum head the mass flow rate will be optimum. In the another analysis ,efficiency versus mass flow rate characteristics obtained the analysis under the different rotational speed ,The optimum mass flow rate i.e best efficiency of the compressor change a function of rotational speed, The optimum mass flow rate increases as the rotational speed goes up,Under the specific rotational speed in case that the mass flow rate is beyond the optimum value, the efficiency of the compressor begins to decrease as the mass flow rate increases, and this phenomenon gets more pronounced at lower rotational speed. The simulation model can be easily adapted to different compressor geometry and different operative fluids; therefore, it can be useful tool for the analysis, the design and the development of the centrifugal compressors. Any new parameters can be introduced without affecting the whole model. The design of highly efficient reciprocating compressors increasingly requires the adaption of simulation methodologies to reduce the costs associated with the design of new products. The present study also made evident the impact of different parameters which need to be optimize so as to increase the performance and to reduce the various losses incurred. The simulation model of the reciprocating compressor offers flexibility in handling complex compressor circuitry and quick adaptability to new arrangements by addition or removal of the required elements.

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