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A Study of Designing 35RT Aluminum Alloy Scroll Compressor

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ABSTRACT

The 35RT scroll type compressor has gradually drawn attention to the designer and already has been established the practical cast iron scroll mockup in the compressor industry. However, the aluminum alloy scroll has not been widespread in this field for the strength and force balance issue. The main objective of this study is to build up the scroll model under the theoretical simulation result. Thus, the FEA method based on the calculated parameters (the pressure and temperature effects) has been used to check the design proposal. Combined the results, there are several findings should be brought out, and the developed Aluminum-made scroll could greatly reduce the bearing load, the weight size and the power loss with advantage.

1. INTRODUCTION

The scroll type is one popular compressor type in air-conditioning application with its advantages which are low noise, low vibration, highly mechanical balance and high energy efficiency. The small compressor using scroll pump is more and more popular, however, the application of commercialized compressor with capacity over 15RT is still a challenge today. In ITRI, the 35RT cast iron scroll compressor applied for air-conditioners has been well developed and exhibited with an excellent performance, but the new model of 35RT compressor with aluminum made scroll has not be published for the cooling system. In year 2002, it was proposed that according to the previous experience and theoretical studying, the 35RT scroll compressor could be developed. The former suggestion has using cast iron to be the scroll material for weight balance effect but increase the bearing loading and low operation life-circle. In this paper, base on the theoretical analysis result, it is utilized to design an aluminum scroll type compressor with highly performance. Because of the giant size scale, the characteristics of 35RT compressor are quite different than the smaller one. The scroll model has been developed based on the calculated datum concerning with the pressure and temperature effects has been checked by simulation software which have be developed in ITRI. The simulated results are most helpful to promote the old design suggestion and shows great different then the former cast-iron scroll type compressors.

2. DESIGN MODEL OUTLINE AND SPECIFICATIONS

In Figure 1, the sketch of a 35RT scroll compressor section shoes the key components which are scroll set (cast iron fixed scroll and aluminum orbiting scroll), small size bearing set, an eccentric shaft. In this study, the investigation structure of the big compressor is a low-pressure motor chamber half-hermetic case design. The operation environment conditions have been shown in Table 1. The initial design data of this 35RT scroll compressor using R22 refrigerant has been listed in Table 2, however, the three model type shows the different volume efficiency results. The suction temperature raise is lower then the theoretical calculations. The machining errors of these scroll curve is 20.3 μm which makes them as references for leakage clearance setup.
3. DESCRIPTION OF DESIGN PROCESS

- **Basic Scroll Parameter Design**

There are four basic parameters of the scroll design that must be defined first which are: (1) the scroll involutes’ pitch, (2) the scroll wrap height, (3) the wrap width, and (4) the scroll extended angle (see Table 2). By using these parameters, the suction volume and the crank radius of the compressor can be determined as follows:

**Suction volume** \( (V_s) \):
\[
V_s = (2N - 1) \pi p (p - 2t) h
\]  
(1)

Where N is defined as following \( N = (\phi_E / 360) - 0.25 \) and \( \phi_E \) is the involute extended angle.

Then, we defined the scroll set orbit radius as equation (2).

**Orbit radius** \( (r) \):
\[
r = p / 2 - t
\]  
(2)

- **Calculate the Scroll Forces**

The variations of volume, temperature and pressure during the adiabatic compression process of refrigerant properties could be simulated. The result would be the input data for calculating scroll chamber pressure and obtain the operation forces worked on scroll set.

**Volume change** :
\[
V_i = ((2i-1) - \theta / \pi) \pi p (p - 2t) h
\]  
(3)

**Pressure change** :
\[
P_i = P_s \times \left( V_s / V \right)^K
\]  
(4)

**Temperature change** :
\[
T_i = T_s \times \left( T_i / T_s \right)^{(K-1)/K}
\]  
(5)

The forces can be calculated by the following equations:

**Radial force** :
\[
F_r = 2ah \times (P_d - P_s)
\]  
(6)

**Tangential force** :
\[
F_\theta = \sum_{i=1}^{N} ph \times \left( 2i - \frac{\theta}{\pi} \right) \times (P_i - P_{i+1})
\]  
(7)

**Axial force** :
\[
F_a = \sum_{i=1}^{N} (P_i - P_s) \times A_i
\]  
(8)

**Centrifugal force** :
\[
F_c = m \times r \times \omega^2
\]  
(9)

The other detail dynamic balance calculations (Oldham-coupling and bearings) can be obtained by further analysis.

- **Bearing Life Prediction**

There are two major types of bearings which will influence on the performance of 35RT model. The bearing life models can be express as:

\[
L_{10h} = \left( 10^6 / 60n \right) \times \left( C / P \right)^v \text{ (Hours)}
\]  
(10)
Where, \( c=3 \) for ball bearing and \( c=10/3 \) for roller bearing.

The oil flow rate, the oil pumping pressure generated by the propeller is:

\[
\Delta P = \rho * r^2 * \omega^2
\]  

\[\text{(11)}\]

- **Volumetric Efficiency**

The volumetric efficiency could be calculated by following step:

\[
\eta_v = \eta_{\text{Heat}} * \eta_{\text{Leakage}}
\]

\[\text{(12)}\]

Where \( \eta_{\text{Heat}} \) is caused by the internal superheat

\[
\eta_{\text{Heat}} = \frac{\rho_s \rho_{\text{pump}}}{\rho_s}
\]

\[\text{(13)}\]

\[
\eta_{\text{Leakage}} = 1 - \frac{m_{\text{Leak}}}{m_{\text{pump}}}
\]

\[\text{(14)}\]

The quantity of \( m_{\text{Leak}} \) is the sum of the integral of Equation (15) and (16).

\[
\frac{dm}{dt} = \frac{\pi \delta^3 (P_i - P_o)}{6 \rho \ln (r_o/r_i)} \quad \text{For scroll tip leakage.}
\]

\[\text{(15)}\]

For scroll flank leakage:

\[
\frac{dm}{dt} = C_{D} * A * \sqrt{\frac{2P_i P_t}{k-1}} \left[ \left( \frac{P_i}{P_{t+1}} \right)^{2/k} - \left( \frac{P_t}{P_{t+1}} \right)^{k+1/k} \right]
\]

\[\text{(16)}\]

- **Design Process**

For the requirement of using aluminium orbiting scroll to substitution the original cast-iron one, in this study we should make the design change to be smallest. The design process flow chart is as shown in figure 4. That is to combine the scroll warp height \( h \) and thickness \( t \) to be the critical parameter \( D_s \). Defining \( D_s = h/t \). Based on the 35RT requirement to change the \( D_s \) value then try to fine an optimum \( D_s \) value. According to the performance prediction simulation by the software developed in ITRI, the 35RT scroll compressor model driving shaft should change the dimension design, and the bearing set could be change to the smaller one with the decreasing load. The simulation result is shown in figure 2. After the simulation we build the computer visual model to do compressor motion checking that we could check the design result and avoid the critical parts inner interference.

4. FEM SIMULATION RESOULT AND DISCUSS

Follow the design process as Figure 4 shown, this study use the Finite Element Method (FEM) package software “COSMOS” as the checking tool to verify the displacement, the stresses and the safety coefficient which we called it “factor of safety” (FOS). First, this study select three types of scroll design from about fifty scroll parameter sets which has been input in our performance predict software. From these performance sets we select three studying cases that had been input to the performance predict software for testing the maximum E.E.R. value. Then, we use the selected parameters to create three compressor solid models. According to the FEM simulation, the scroll model displacement and stress distribution of the studying case 2 which has been shown in the Figure 6. In another two study cases the displacement is bigger then case 2 means the flank leakage and tip leakage of refrigerant at model 2 is the better one. The FOS value is shown in Figure 7. From the CAE motion simulation, the disequilibrium produce
of centrifugal force and the radial of strength of the orbiting scroll would cause leakage. Increase quality of orbiting scroll (separate strength and balance) and radial of strength to let the compressor operation smooth.

5. PERFORMANCE RESULT AND DISCUSS

From the comparison of these three design types of 35RT scroll compressor, some important results would be concluded.

1. Changing the material of the orbiting scroll from cast iron to aluminium, the worse force balance situation can be completely improved that means the decreasing of scroll mass and crank radius, an obvious reducing on the centrifugal force “Fc” could decrease the bearing loading. The improved centrifugal force “F_{c_2}” is reduced to about 62% of previous design “Fc”. On the other hand, the upper bearing force “F_{UB}” also can be reduced at the same time. In addition, the material changing provides some other advantages: (a) minimize compressor size and the power loss of the upper bearing; (b) minimize counter weight size; (c) make the machining tools has longer service life.

2. As shown in Figure 4, the design process of the new 35RT model make the shorter design cycle time.

3. The required flow rate of lubricating oil for 35RT alumina scroll model is almost the same as the Cast Iron One. The displacement (see figure 8), therefore, the displacement of both type 35RT scroll are almost the same. The maximum displacement of aluminium scroll is 0.04477 mm and the cast iron one is 0.04663. There is a 4% improve of scroll rigidity.

4. The aluminum scroll set model shows that the EER value upper than 3.15 is possible to fabrication.

5. As shown in figure 7, the mechanical FOS value is about 1 for 6061-T6 aluminum alloy model. But, the Cast iron scroll increases more than 2 times due to the larger material property effect. However, a better performance can be obtained by replacing the scroll from the cast iron type to aluminium alloy type. The total bearing loss can be reduced about 30%. The size of the roller-type bearings is related to the bearing load and design life cycles (at least 37000 hours are needed here). However, it was found that the weight and the cost of bearing are lower than the Cast iron ones in 35RT application.

6. CONCLUSIONS

As the result, there are several major findings should be brought out. First, the distributions of stress and deformation of the orbiting scroll due to the pressure and temperature effects are obtained. Second, under operating conditions, some gas forces exert on scrolls in different directions, therefore, the dynamic behaviors of the orbiting scroll members are simulated by choosing the 6061-T6 aluminum alloy as the material for the orbiting scroll. The simulated results are most helpful to promote the current design and shows great different then the former cast iron orbiting scroll compressors. The Aluminum-made scroll could reduce the bearing loading, the weight size and the power loss with advantage. Finally, the test result has showed the E.E.R. of this 35RT compressor design is almost 3.0 or even higher.

NOMENCLATURE

\[ A_i : \text{Chamber area} \]
\[ a : \text{Base circle radius} \]
\[ C : \text{Bearing Load} \]
\[ C_B : \text{Bearing clearance} \]
\[ C_D : \text{Flow coefficient} \]
\[ c : \text{Life index} \]
\[ D : \text{Bearing diameter} \]
\[ e : \text{Eccentricity} \]
\[ N : \text{Scroll involutes’ turns} \]
\[ P_i : \text{Chamber pressure} \]
\[ P_s : \text{Suction pressure} \]
\[ P_d : \text{Discharge pressure} \]
\[ p : \text{Scroll pitch} \]
\[ Q_c : \text{Capacity} \]
\[ R : \text{Bearing radius} \]
\[ r : \text{Crank radius} \]

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\( F_a \): Axial force  \( r_i \): Equivalent radius
\( F_B \): Bearing force  \( T_i \): Chamber temperature
\( F_c \): Centrifugal force  \( V_i \): Chamber volume
\( F_r \): Radial force  \( V_s \): Suction volume
\( F_t \): Tangential force  \( \delta \): Leakage clearance
\( h \): Wrap height  \( \rho \): Oil density
\( L \): Bearing Length  \( \rho_s \): Compressor inlet refrigerant density
\( L_{10b} \): Bearing life  \( \rho_{sPump} \): Suction refrigerant density
\( M \): Shaft mobility  \( \kappa \): Isentropic Index
\( D_s \): Rigidity of scroll warp  \( \omega \): Rotating speed
\( m_{sPump} \): Refrigerant mass  \( \mu \): Friction coefficient
\( n \): Operating frequency  \( \eta_c \): Compression efficiency
\( \eta_{\text{Heat}} \): Superheat volumetric efficiency  \( \eta_{\text{Leakage}} \): Leakage volumetric efficiency

REFERENCES


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Figure 1 the schematic outline of the developed compressor

<table>
<thead>
<tr>
<th>Table 1 Compressor working conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>54.4°C</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2 the Characteristics of 35RT scroll compressors</th>
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</thead>
<tbody>
<tr>
<td><strong>Model</strong></td>
</tr>
<tr>
<td>Refrigerant</td>
</tr>
<tr>
<td>Capacity target (kcal/h)</td>
</tr>
<tr>
<td>DESIGN CASE</td>
</tr>
<tr>
<td>Suction Volume (c.c.)</td>
</tr>
<tr>
<td>Temperature Rise (°C)</td>
</tr>
<tr>
<td>Motor Speed (rpm)</td>
</tr>
<tr>
<td>Scroll Curve Error (m)</td>
</tr>
<tr>
<td>Lubrication Oil</td>
</tr>
<tr>
<td>Crank Radius</td>
</tr>
<tr>
<td>Disk Thickness</td>
</tr>
<tr>
<td>Base plate thickness</td>
</tr>
<tr>
<td>Shaft diameter</td>
</tr>
<tr>
<td>Upper bearing</td>
</tr>
<tr>
<td>Trust bearing</td>
</tr>
<tr>
<td>Lower bearing</td>
</tr>
<tr>
<td>Wrap thickness</td>
</tr>
<tr>
<td>Wrap height</td>
</tr>
<tr>
<td>Scroll pitch</td>
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<tr>
<td>Involutes’ Turns</td>
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<tr>
<td>EER</td>
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</tbody>
</table>
Figure 2 Comparison of the reaction forces of different scroll material.

(Comparison with theoretical calculation)

Definition:
1. 35RT Refrigeration Capacity
2. Design variables: p, t, h, $\phi_e$
3. Efficiency
4. E.E.R. more than 3.0

Collect R22 STC System Design Data
1. Friction coefficient of each STC critical parts
2. Experiment Data of Cast-iron Type
3. Motor performance Data

Set Up Initial Design Model
Evaluate the performance On ITRI Simulation Software
Mechanical Design Check by Finite Element Method

Combination

Check the
1. E.E.R. Value
2. Scroll Wrap

NO
Change variable initial value

YES

1. 3D Model
2. Tolerance
3. Graphic

Stop

Figure 3. Design Process in this investigation
Figure 4. Scroll Design variables

Figure 5. Scroll design variables: FOS

Figure 6. Scroll displacement