2002

A Case Study Of Theoretical Analysis Comparisons Of 3RT And 35RT Scroll Compressors

C. F. Lai  
*Industrial Technology Research Institute; TAIWAN*

Y. C. Chang  
*Industrial Technology Research Institute; TAIWAN*

B. C. Yang  
*Industrial Technology Research Institute; TAIWAN*

Follow this and additional works at: https://docs.lib.purdue.edu/icec

https://docs.lib.purdue.edu/icec/1607

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.  
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
A Case Study of Theoretical Comparisons of 3RT and 35RT Scroll Compressors

Ching-Feng Lai*, Yu-Choung Chang and Bing-Chwen Yang, Ph.D.
Energy & Resources Laboratories, Industrial Technology Research Institute
Hsinchu 310, Taiwan, ROC; Tel.: 886/3-5913366; Fax: 886/3-5820250
E-Mail: CFLai@itri.org.tw *Author for Correspondence

ABSTRACT

The large capacity of 35RT scroll compressor results more significant of forces and heat than 3RT model through the compression process than 3RT. In order to minimize the size, weight and cost of compressors and optimize the performance and reduce the risk of failure, the proper design of dimensions, material and lubrication is very important. In this paper, some defects of the 35RT model in the original design are found by comparison with the analytical results for both models of compressors. In addition, the better solutions for the worse situations are proposed and the advantages of improvement are also discussed in this study.

NOMENCLATURE

$A_i$: Chamber area
$\alpha$: Base circle radius
$C$: Bearing Load
$C_B$: Bearing clearance
$C_D$: Flow coefficient
c: Life index
$D$: Bearing diameter
e: Eccentricity
$F_a$: Axial force
$F_B$: Bearing force
$F_c$: Centrifugal force
$F_r$: Radial force
$F_\theta$: Tangential force
$h$: Wrap height
$\delta$: Leakage clearance
$L$: Bearing Length
$L_{10h}$: Bearing life
$M$: Shaft mobility
$m_{\text{Leak}}$: Leakage refrigerant mass
$m_{\text{Pump}}$: Refrigerant mass
$n$: Operating frequency
$N$: Scroll involute turns
$P_{\text{ad}}$: Adiabatic compression power
$P_{\text{bearing}}$: Bearing Power Loss
$P_{\text{comp}}$: Compression power
$P_{\text{friction}}$: Friction Power Loss
$P_{\text{mech}}$: Mechanical Power
$P_{\text{motor}}$: Motor input power
$P_i$: Chamber pressure
$P_s$: Suction pressure
$P_d$: Discharge pressure
$p$: Scroll pitch
$Q_c$: Capacity
$R$: Bearing radius
$r$: Crank radius
$r_i$: Equivalent radius
$T_i$: Chamber temperature
$V_i$: Chamber volume
$V_s$: Suction volume
$\eta_c$: Compression efficiency
$\eta_{\text{Leak}}$: Leakage volumetric efficiency
$\eta_{\text{Heat}}$: Superheat volumetric efficiency
$\eta_{\text{mech}}$: Mechanical efficiency
$\eta_{\text{motor}}$: Motor efficiency
$\eta_{\text{mech}}$: Volumetric efficiency
$\delta$: Leakage clearance
$\rho$: Oil density
$\rho_s$: Compressor inlet refrigerant density
$\rho_{\text{Pump}}$: Suction refrigerant density
$\kappa$: Isentropic Index
$\omega$: Rotating speed
$\mu$: Friction coefficient
$\nu$: Viscosity
**INTRODUCTION**

The scroll compressor is a popular type of compressor due to its high efficiency, low vibration and low noise. However, the development of commercialized compressor with capacity over 15RT is still a challenge today. As shown in Figure 1, a 3RT model scroll compressor applied for air-conditioners has been well developed and exhibited with an excellent performance several years ago. A new model of 35RT scroll compressor used for the chiller system was designed according to the previous experience. It is well known that the high-quality design is the most effective way to prove the competitive ability of product in the marketplace. In this paper, the theoretical analysis method is utilized to predict the compressor performance. Because of the different size scales of compressors, the characteristics between 3RT and 35RT are found quite different by the simulation results (see Figure 2). A further discussion is carried out to find the detail relationships that affect the compressor efficiencies.

**CASE STUDY DESCRIPTION**

The working conditions, compressor specifications and basic characteristics for 3RT and 35RT scroll compressors employed in this case study are shown in Table 1 and Table 2. The same refrigerant and lubrication oil are used for both models of compressors. However, the motor speed of 35RT model is 1.5% higher than 3RT model according to the dynamometer test results. The suction temperature raise of 35RT model is about 4°C lower by the theoretical calculations. The machining errors of these two scroll curves are both 6.3µm and 20.3µm, which are used as references for leakage clearance setup.

**PERFORMANCE ANALYSIS**

**Initial Design**

**Scroll Parameter Design**

Four basic parameters for the scroll design are: (1) the scroll pitch, (2) the wrap height, (3) the wrap width, and (4) the number of involute turns [1]. By using these parameters, the suction volume and the crank radius of the compressor can be determined as follows:

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

**Calculation of Refrigerant Properties**

With the geometry defined from above, the variations of volume, temperature and pressure during the adiabatic compression process can be defined,

- **Volume change**: 
  \[ V_i = ((2i - 1) \cdot \theta/\pi)\pi p(p - 2t)h \]  
- **Pressure change**: 
  \[ P_i = P_x \ast \left(\frac{V_s}{V}\right)^K \]  
- **Temperature change**: 
  \[ T_i = T_x \ast \left(\frac{T_i}{T_s}\right)^{(K-1)/K} \]

**Calculation of Forces**

The forces caused by pressure and mass can be calculated by the following equations:

- **Radial force**: 
  \[ F_r = 2ah \ast \left( P_i - P_x \right) \]  
- **Tangential force**: 
  \[ F_\theta = \sum_{i=1}^{N} ph \ast \left( 2i - \frac{\theta}{\pi} \right) \ast (P_i - P_{i+1}) \]
Axial force: \[ F_a = \sum_{i=1}^{N} (P_i - P_o) \times A_i \]  
(8)

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

Also, the other detail calculations of dynamic balance for Oldham-coupling and bearings can be obtained by further analysis [2]. The simulation results are shown in Figure 3.

**Bearing Model**

Two major types of bearings are adopted to evaluate their influence on the performance of 35RT model. These two bearing models can be express as:

1. Ball bearing: \[ L_{10h} = \left(\frac{10^6}{60n}\right) \times \frac{(C/P)}{C_B} \text{ (Hours)} \]  
(10)

   Where, \( c=3 \) for ball bearing and \( c=10/3 \) for roller bearing.

2. Journal bearing [3-5]:

   While the oil supplement is sufficient, the behavior of journal bearing can be analyzed by the “mobility method”,

   \[
   \frac{d}{dt}\begin{bmatrix} e_x \\ e_y \end{bmatrix} = \begin{bmatrix} \frac{C_B}{R} \\ C_B \\ M_x \\ M_y \end{bmatrix} + \begin{bmatrix} 0 & -1 \\ 1 & 0 \end{bmatrix} \begin{bmatrix} e_s \\ e_y \end{bmatrix}
   \]  
(11)

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

   \[ \Delta p = \rho \times \omega^2 \]  
(12)

**Volumetric Efficiency**

The volumetric efficiency can be defined as follows:

\[ \eta_v = \eta_{Heat} \times \eta_{Leakage} \]  
(13)

where \( \eta_{Heat} \) is caused by the internal superheat and can be defined as \( \eta_{Heat} = \frac{\rho_{Pump}}{\rho_s} \)  
(14)

However, the \( \eta_{Leakage} \) has relation with the leakage effect of compression chambers [6-7]:

\[ \eta_{Leakage} = 1 - \frac{m_{Leak}}{m_{Pump}} \]  
(15)

The quantity of \( m_{Leak} \) is the sum of the integral of Eq.(16) and Eq.(17) w.r.t. time.

\[
\frac{dm}{dt} = \frac{\pi \delta^2 (P_i - P_o)}{6\nu \ln(r_o/r_i)} \quad \text{for scroll tip leakage.}
\]  
(16)

\[
\frac{dm}{dt} = C_D \times A \times \sqrt{\frac{2P_i}{k-1} \left[ \left( \frac{P_i}{P_{i+1}} \right)^{2/k} - \left( \frac{P_i}{P_{i+1}} \right)^{k+1/k} \right]} \quad \text{for scroll flank leakage.}
\]  
(17)

**Compression Efficiency**

The compression efficiency means the ratio of ideal and actual compression power that can be defined as follows:

\[ \eta_c = \frac{P_{adiabatic}}{P_{comp}} \]  
(18)
\[ P_{\text{adiabatic}} = \eta_v \left( \frac{\kappa}{\kappa - 1} \right) \ast P_{sPump} \ast V_{sPump} \left( \frac{P_{dfPump}}{P_{sPump}} \right)^{\frac{\kappa - 1}{\kappa}} \ast \theta_{mi} \]  \hspace{1cm} (19)

\[ P_{\text{comp}} = \int_{\theta=0}^{2\pi} (P(\theta) - P_s) dv \]  \hspace{1cm} (20)

In the above equation, \( P(\theta) \) is a function of leakage effect and port conditions. Figure 4 shows the analytical results of \( P(\theta) \). By the comparison of pressure variation, it is found that the 3RT model possesses more significant over-compression phenomenon.

**Mechanical Efficiency**

The mechanical efficiency can be defined as follows:

\[ \eta_{\text{mech}} = \frac{P_{\text{comp}}}{P_{\text{mech}}} \]  \hspace{1cm} (21)

where \( P_{\text{mech}} = P_{\text{comp}} + P_{\text{friction}} + P_{\text{bearing}} + P_{\text{oilpump}} \)  \hspace{1cm} (22)

The mechanical loss is resulted from friction, bearings and oil pumping loss. The analytical results for mechanical efficiency are given in Figure 5.

**Motor Efficiency**

The motor efficiency, which can be measured by the dynamometer (see Figure 6), defined as follows:

\[ \eta_{\text{motor}} = \frac{P_{\text{mech}}}{P_{\text{motor}}} \]  \hspace{1cm} (23)

where \( P_{\text{mech}} \) means the power output of motor and \( P_{\text{motor}} \) means the power input.

**Energy Efficiency Ratio**

The performance index E.E.R. of compressors is:

\[ E.E.R. = \frac{Q}{P_{\text{motor}}} \]  \hspace{1cm} (24)

**RESULTS AND DISCUSSION**

Some important results can be concluded from the comparison of these two models of scroll compressor.

1. Because of the increasing of scroll mass and crank radius (see figure 7 and figure 8), an obvious growing on the centrifugal force “Fc” about magnification of 21 occurs (see figure 3). By changing the material of the orbiting scroll from cast iron to aluminium, this worse condition can be completely improved. The improved centrifugal force “Fc_2” is reduced to about 35% of previous design “Fc”. Besides, the upper bearing force “F\_UB” and the Oldham-coupling force “F\_oldham” also can be reduced at the same time. In addition, the design change provide some other advantages: (a) minimize the size, cost and power loss of the upper bearing; (b) minimize the size and cost of counter weight; (c) make the service life of machining tools longer.

2. As shown in Figure 9, the volumetric efficiency caused by internal superheat and leakage effect for 35 RT model are 1.73% and 1.62% higher, respectively. This results an increasing of the volumetric efficiency by 1.63% as shown in Figure 10. Even though the capacity of 35RT model is 11.67 times of 3RT model, however, the suction volume is only about 11 times higher due to the higher volumetric efficiency.

3. By Consideration of the design requirements of suction volume, structure strength and machining ability, the dimensions of 35RT model is about double of 3RT model as shown in Figure8.

4. As shown in Figure 10, the motor efficiency, compression efficiency and mechanical efficiency of 35RT model are 5.67%, 2.17%, 0.12% higher, respectively.
(5) The required flow rate of lubricating oil for 35RT model is about 10 times more than 3RT model as shown in Figure 11. Therefore, the proportional increasing of oil pumping pressure is also needed for 35RT. According to equation (12), the size of oil-pump propeller requires 3.2 times of 3RT size or larger.

(6) As shown in figure 5, the mechanical loss is about 10-time bigger for 35RT model. But, the journal bearing loss (see figure 5) increases more than 10 times due to the larger bearing forces and bearing sizes (see figure 3 and figure 8). However, a better performance can be obtained by replacing the bearings from journal-type to roller-type (see figure 5). The total bearing loss can be reduced about 60%. This is resulted from the lower friction coefficient of roller bearings. Meanwhile the mechanical efficiency “Mech_2” can be improved about 2.8% for the 35RT model. The size of the roller-type bearings is related to the bearing load and design life cycles (over 37000 hours are needed here). However, it was found that the weight and the cost of roller bearings are higher than the journal-type ones in 35RT application.

(7) From Figure 12, the initial design E.E.R. of 35RT model is about 12.3% higher than 3RT model.

CONCLUSIONS

The Al-made orbiting scroll can reduce the bearing size, counter weight sizes and power loss. Moreover, It will make an additional improvement of compressor performance by using the roller-type bearings, but the weight and the cost of bearings will get higher. The E.E.R. of 35RT final design will be 3.20 or even higher due to the design changes.

ACKNOWLEDGEMENTS

The authors would like to express their thanks for financial support of the Energy R&D Fund provided by the Energy Commission of the Ministry of Economic Affairs in Taiwan, R.O.C..

REFERENCES


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

<p>| Table 2 Comparison of Characteristics between 3RT and 35RT scroll compressors |
|--------------------------|---------|---------|
| Model | 3RT | 35RT |
| Refrigerant | R-22 | R-22 |
| Capacity (kcal/h) | 9130 | 105960 |
| Suction Volume (c.c.) | 50.2 | 553.7 |
| Temperature Rise (°C) | 12 | 7.9 |</p>
<table>
<thead>
<tr>
<th>Motor Speed (rpm)</th>
<th>3485</th>
<th>3536</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scroll Curve Error (µm)</td>
<td>6.3</td>
<td>20.3</td>
</tr>
<tr>
<td>Lubrication Oil</td>
<td>4GS</td>
<td>4GS</td>
</tr>
</tbody>
</table>

(a) 3RT

(b) 35RT

Figure 1: The scheme of two different models of scroll type compressor.

Figure 2: Comparison of property scale of 3RT and 35RT.
Figure 3: Comparison of the reaction forces of 3RT and 35RT scroll compressors.

Figure 4: Comparison of compression process of 3RT and 35RT.

Figure 5: Comparison of mechanical loss of 3RT and 35RT.

Figure 6: Comparison of motor efficiency of 3RT and 35RT.
Figure 9: Comparison of volumetric efficiencies due to leakage and superheat.

Figure 7: Comparison of off-center mass of 3RT and 35RT scroll design.

Figure 8: Comparison of important dimension parameters of scroll design and bearings.
Figure 10: Comparison of efficiencies of 3RT and 35RT.

Figure 11: Comparison of oil flow rate need of 3RT and 35RT.

Figure 12: Comparison of energy efficiency ratio of 3RT and 35RT.