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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
a : 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_θ : Tangential force
h : Wrap height
L : Bearing Length
L_{10h} : Bearing life
M : Shaft mobility
m_{sLeak} Leakage refrigerant mass
m_{sPump} : Refrigerant mass
n : Operating frequency
N : Scroll involute turns
$P_{adiabatic}$: Adiabatic compression power
$P_{bearing}$: Bearing Power Loss
P_{comp} : Compression power
$P_{friction}$: Friction Power Loss
P_{mech} : Mechanical Power
P_{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
η_c : Compression efficiency
$\eta_{Leakage}$: Leakage volumetric efficiency
η_{Heat} : Superheat volumetric efficiency
η_{mech} : Mechanical efficiency
η_{motor} : Motor efficiency
η_v : Volumetric efficiency
δ : Leakage clearance
ρ : Oil density
ρ_s : Compressor inlet refrigerant density
ρ_{sPump} : Suction refrigerant density
κ : Isentropic Index
ω : Rotating speed
μ : Friction coefficient
υ : 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 Figure1, 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 completive 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℃ lower by the theoretical calculations. The machining errors of these two scroll curves are both 6.3μm and 20.3 μm, 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： $V_s = (2N-1)\pi p(p-2t)h$ （1）

Crank radius： $r = p/2 - t$ （2）

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： $V_i = ((2i-1)\text{-}\theta/\pi)\pi p(p-2t)h$ （3）

Pressure change： $P_i = P_s * (V_s/V)^K$ （4）

Temperature change： $T_i = T_s * (T\ /T_s)^{(K-1)/K}$ （5）

Calculation of Forces

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

Radial force： $F_r = 2ah * (P_d - P_s)$ （6）

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

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

Centrifugal force： $F_c = m * r * \omega^2$ （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: $L_{10\text{h}} = (10^6 / 60n) * (C / P)^c$ （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",

$$\frac{d}{dt}\begin{bmatrix} e_x \\ e_y \end{bmatrix} = \frac{\left|\overrightarrow{F_B}\right|\left(\frac{C_B}{R}\right)}{LD\left(\frac{\mu}{C_B}\right)}\begin{bmatrix} M_x \\ M_y \end{bmatrix} + \bar{\omega}\begin{bmatrix} 0 & -1 \\ 1 & 0 \end{bmatrix}\begin{bmatrix} e_x \\ e_y \end{bmatrix} \quad （11）$$

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

$$\Delta p = \rho * r^2 * \omega^2 \quad （12）$$

Volumetric Efficiency

The volumetric efficiency can be defined as follows:

$$\eta_v = \eta_{Heat} * \eta_{Leakage} \quad （13）$$

where η_{Heat} is caused by the internal superheat and can be defined as $\eta_{Heat} = \rho_{sPump} / \rho_s$ （14）

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

$$\eta_{Leakage} = 1 - m_{sLeak} / m_{sPump} \quad （15）$$

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

$$\frac{dm}{dt} = \frac{\pi\delta^3 (P_i - P_o)}{6\upsilon \ln(r_o / r_i)} \quad \text{for scroll tip leakage.} \quad （16）$$

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

Compression Efficiency

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

$$\eta_c = \frac{P_{adiabatic}}{P_{comp}} \quad （18）$$

$$P_{adiabatic} = \eta_v \left\{ \left(\frac{\kappa}{\kappa - 1} \right) * P_{sPump} * V_{sPump} \left[\left(\frac{P_{dPump}}{P_{sPump}} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] * \omega_m \right\} \quad (19)$$

$$P_{comp} = \int_{\theta=0}^{2\pi} (P(\theta) - P_s) dv \quad (20)$$

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:

$$\eta_{mech} = \frac{P_{comp}}{P_{mech}} \quad (21)$$

where $P_{mech} = P_{comp} + P_{friction} + P_{bearing} + P_{oilpump}$ (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:

$$\eta_{motor} = \frac{P_{mech}}{P_{motor}} \quad (23)$$

where P_{mech} means the power output of motor and P_{motor} means the power input.

Energy Efficiency Ratio

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

$$E.E.R. = \frac{Q_c}{P_{motor}} \quad (24)$$

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

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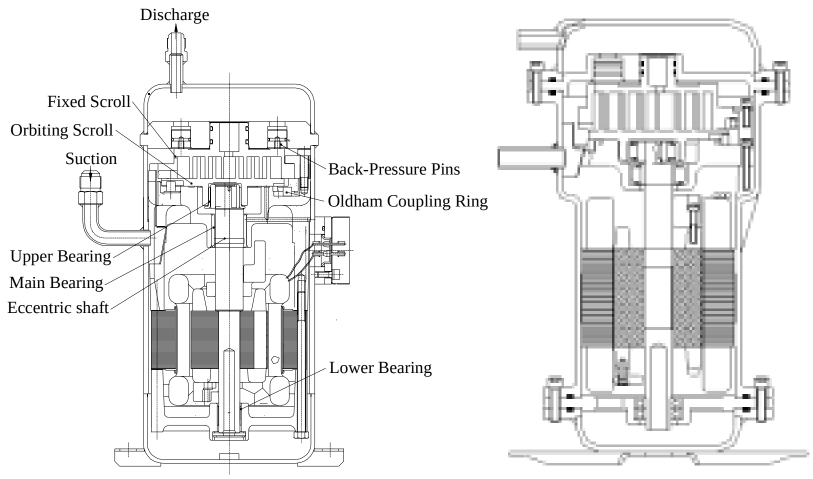
Table 1 Compressor working conditions

Condensing Temp.	Evaporating Temp.	Sub-cooling Temp.	Superheating Temp.	Room Temp.
54.4℃	7.2℃	8.3℃	27.8℃	35℃

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 (℃)	12	7.9

Motor Speed (rpm)	3485	3536
Scroll Curve Error (μm)	6.3	20.3
Lubrication Oil	4GS	4GS



(a) 3RT (b) 35RT

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

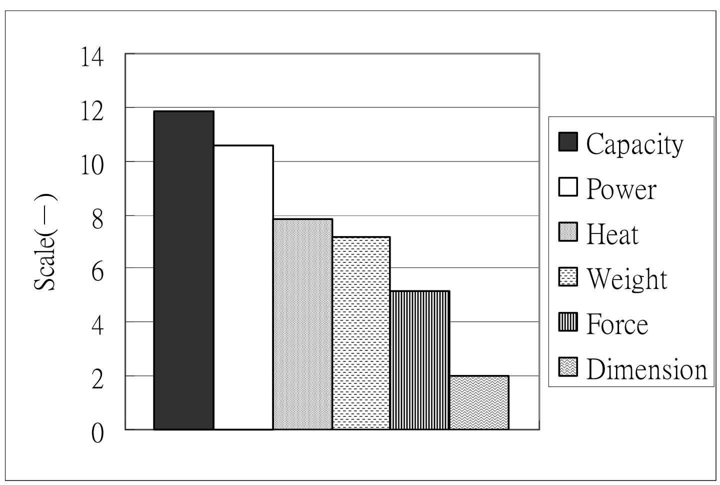


Figure 2: Comparison of property scale of 3RT and 35RT.

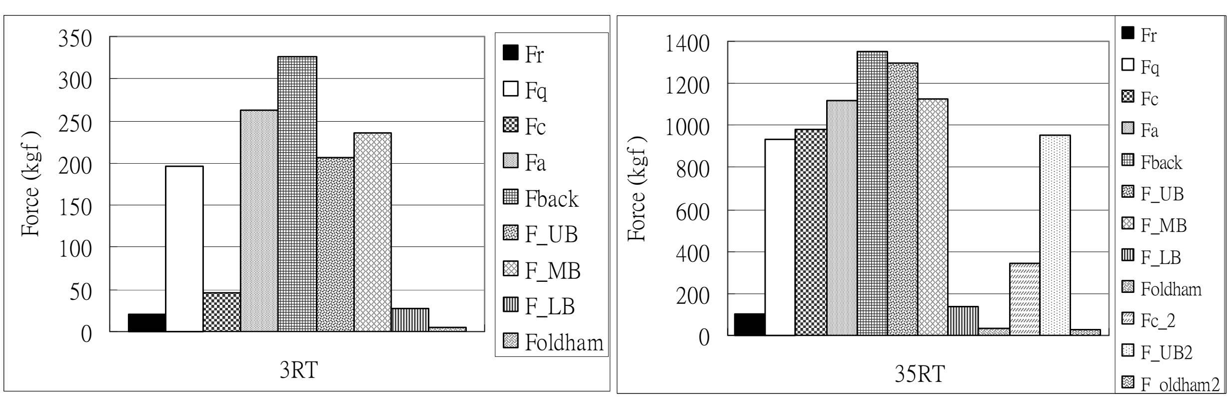


Figure 3: Comparison of the reaction forces of 3RT and 35RT scroll compressors.

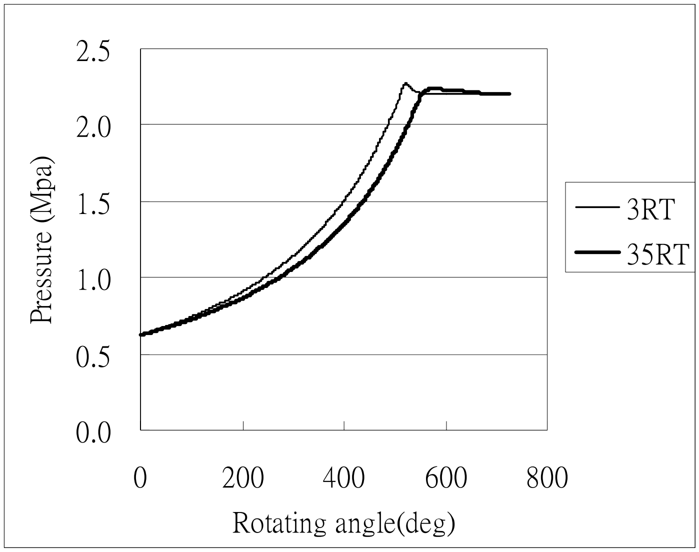


Figure 4: Comparison of compression process of 3RT and 35RT.

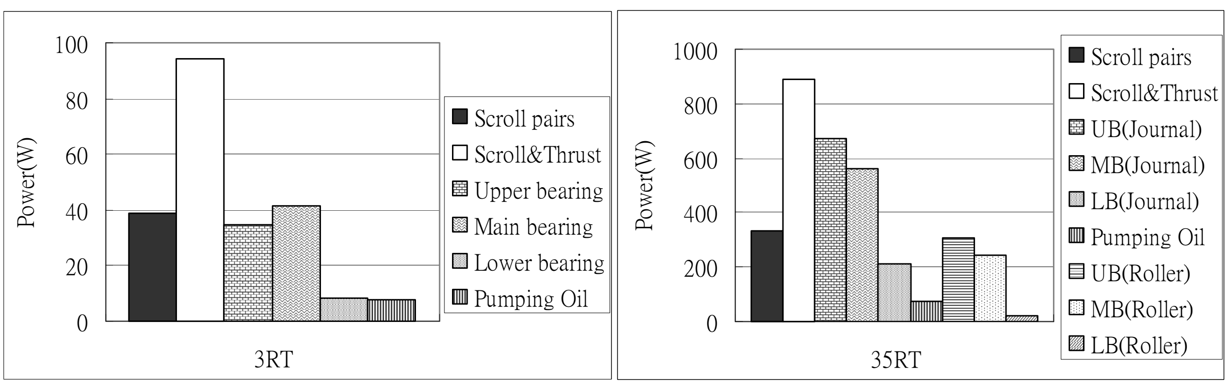


Figure 5: Comparison of mechanical loss of 3RT and 35RT.

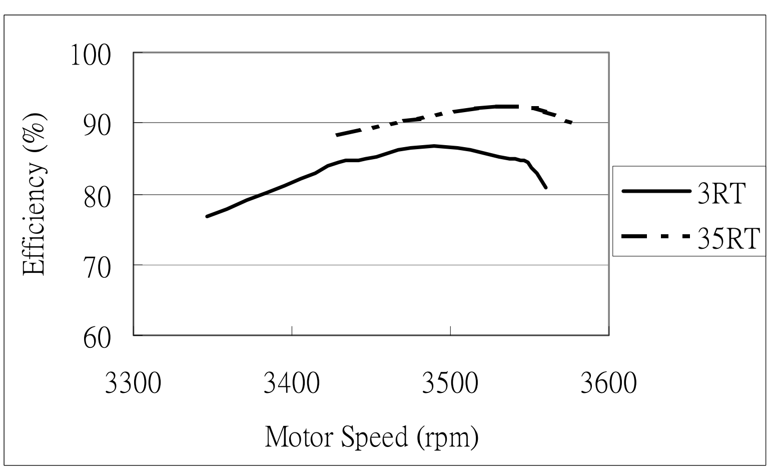
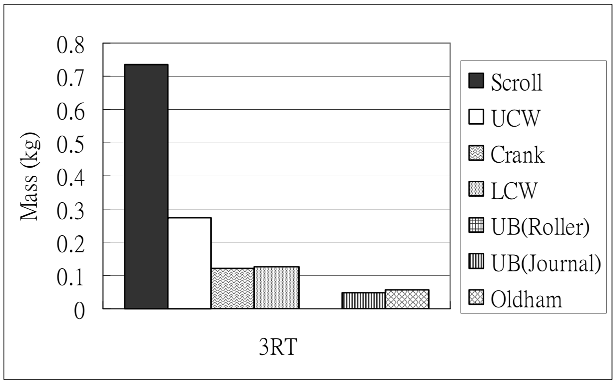
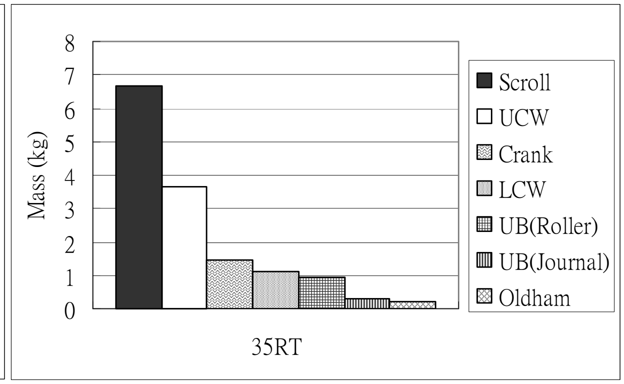


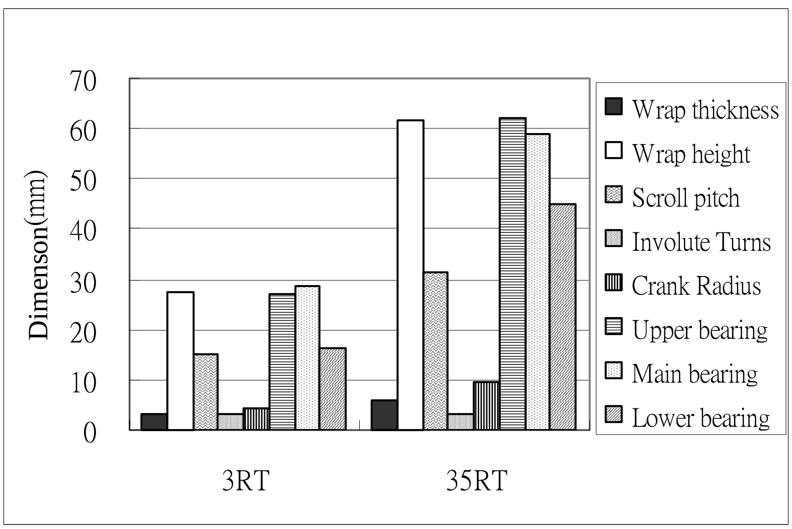
Figure 6: Comparison of motor efficiency of 3RT and 35RT.



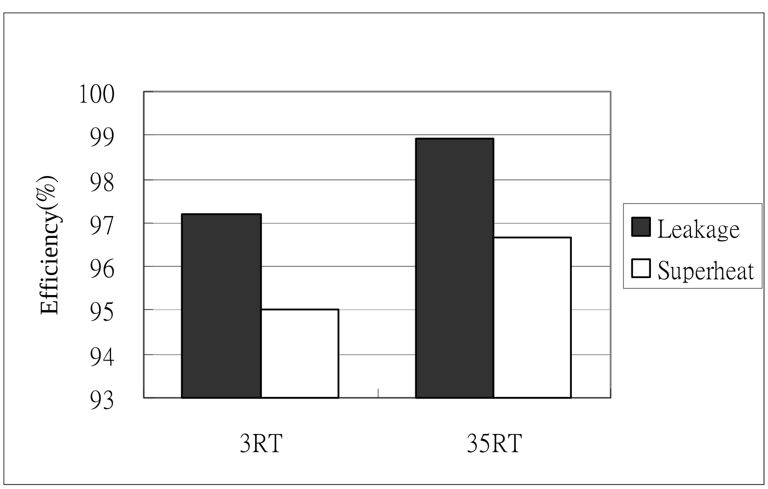
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
Mass (kg)
3RT
Scroll
UCW
Crank
LCW
UB(Roller)
UB(Journal)
Oldham



8
7
6
5
4
3
2
1
0
Mass (kg)
35RT
Scroll
UCW
Crank
LCW
UB(Roller)
UB(Journal)
Oldham



70
60
50
40
30
20
10
0
Dimenson(mm)
3RT
35RT
Wrap thickness
Wrap height
Scroll pitch
Involute Turns
Crank Radius
Upper bearing
Main bearing
Lower bearing



100
99
98
97
96
95
94
93
Efficiency(%)
3RT
35RT
Leakage
Superheat

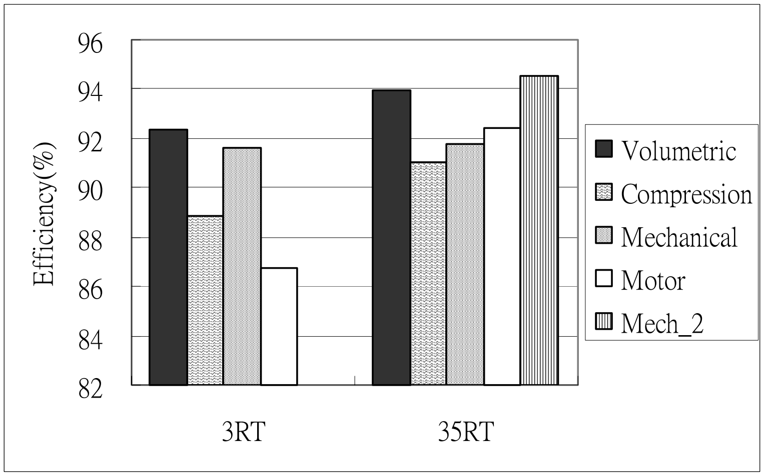


Figure 10: Comparison of efficiencies of 3RT and 35RT.

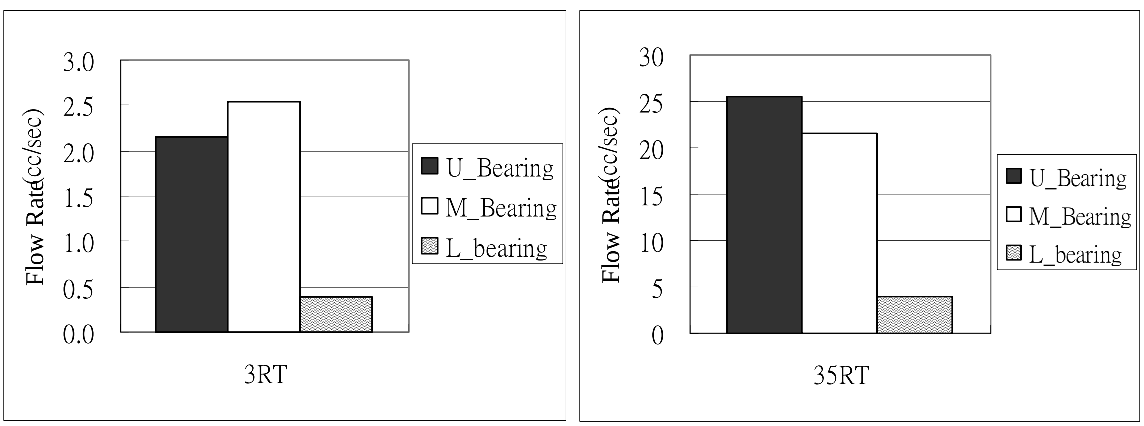


Figure 11: Comparison of oil flow rate need of 3RT and 35RT.

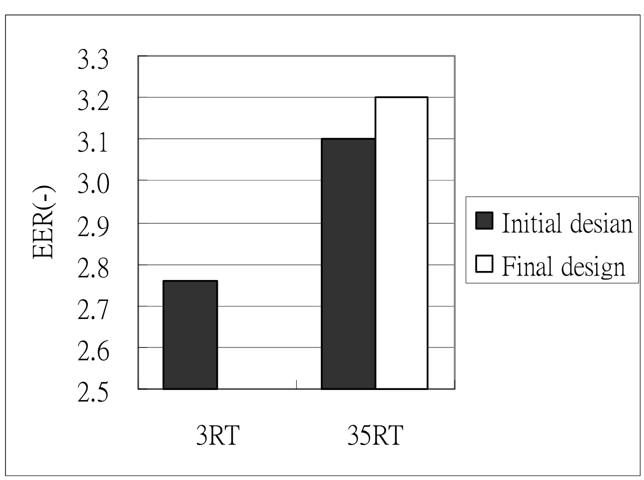


Figure 12: Comparison of energy efficiency ratio of 3RT and 35RT.