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Performance Comparison of Various Types of CO_2 Compressors for Heat Pump Water Heater Application

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Abstract

Numerical simulations for scroll, two-stage twin rotary, and two-cylinder reciprocating compressors have been carried out to understand the effectiveness of each type compressor for heat pump water heater application using CO_2 as refrigerant. For suction pressure of 3.5 MPa and discharge pressure of 9 MPa, clearance volume ratio of the reciprocating compressor needs to be about 5% or less to have the volumetric efficiency comparable to that of the scroll compressor with tip clearance of 5 μm . Volumetric efficiency of the scroll compressor is quite sensitive to tip clearance. Adiabatic efficiency of the twin rotary compressor was calculated to be the lowest among the three types, and the most severe drawback of the CO_2 scroll compressor was a significant increase in the mechanical loss at the thrust surface supporting the orbiting scroll member. While the scroll compressor showed very smooth torque load variation, peak-to-peak torque variations of the twin rotary and two-cylinder reciprocating compressors were about 50% and 250%, respectively.

Key words: Heat pump, Scroll, Twin rotary, Reciprocating, CO_2

Nomenclature

- A : area [m^2]
- c_v : flow coefficient
- f : friction coefficient
- L : loss [W]
- M : mass [kg]
- $\dot{m}$: mass flow rate [kg/s]
- n : polytrophic index
- P : pressure [Pa]
- R : gas constant, radius of curvature [m]
- T : temperature [℃]
- t : thickness [m]
- V : control volume [m^3]
- W : useful work [W]
- Z_c : compressibility factor

Greeks

- ε : leakage clearance [m]
- γ : clearance volume ratio [%]
- η : efficiency

Subscripts

- a : actual
- ad : adiabatic
- c : compression
- cb : suction chamber into compression chamber
- dv : discharge valve
- eb : suction chamber into through vane slot
- ec : compression chamber into through vane slot
- i : ideal
- lk : leakage
- v : volumetric
- m : intermediate, mean
- $mech$: mechanical
- mt : motor
- pr : piston ring
- r : flank leakage
- rb : suction chamber into roller
- rc : compression chamber into roller
- s : suction

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sv : suction valve
th : thrust bearing
vb : leakage through vane side
vt : leakage through vane tip
z : tip leakage
1, 2 : Suction and discharge, respectively

1. Introduction

Among natural refrigerants, CO_2 is one of the most promising alternatives for HCFC and HFC, especially in applications where safety is a concern: It is inflammable and nontoxic. It has been more than 10 years since CO_2 was revived as a refrigerant, and research efforts have been increased to develop CO_2 systems for several applications. Recent studies indicate that CO_2 has promise in particular applications including heat pumps for water heating.[1-3]

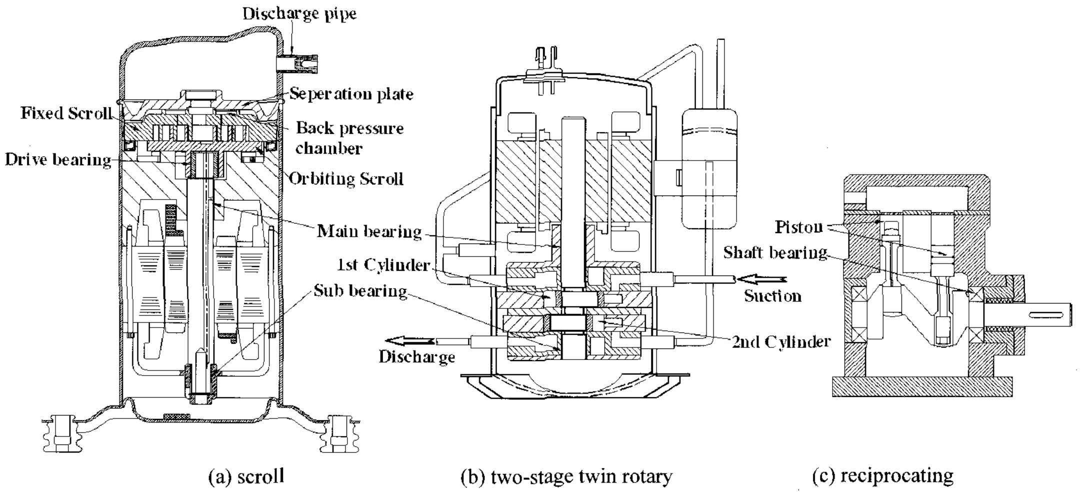
One of the major issues that relates to the CO_2 cycle for heat pump application is selection of compressor type. Several cases of CO_2 compressor development have been reported. Among them are semi-hermetic or hermetic reciprocating[4, 5], swing rotary[6], rolling piston rotary[7], vane[8] and scroll compressors.[9, 10]

In this study, a theoretical investigation has been carried out to assess the performance of scroll, two-stage rotary, and two-cylinder reciprocating compressors for the application of CO_2 heat pump water heater, and relative advantages and disadvantages for each compressor have been examined.

2. Computer simulation program and compressor models

Fig. 1 shows compressor models. Scroll compressor has a back pressure chamber behind the fixed scroll, and the orbiting scroll is supported by the thrust bearing on its backside. For radial compliance, slider bush is used. Rotary compressor is a two-stage twin cylinder type, having two oppositely situated eccentrics on the crankshaft. The pressure inside the shell is intermediate pressure, which is about the same as the discharge pressure for the first stage and as the suction pressure for the second stage. Reciprocating compressor has two one-stage cylinders. Displacement volumes are about 14 cc, 13 cc, and 42 cc for the scroll, twin rotary and reciprocating compressors, respectively. For CO_2 reciprocating compressors, the displacement of 42 cc was the smallest one currently available for heat pump use. Typical operating condition of a CO_2 heat pump water heater system is shown in Fig. 2. Compressor runs between points 1 and 2. For the following calculations, these points are fixed: $P_1 = 3.5$ MPa and $P_2 = 9$ MPa, and compressor speed is also held at 3500 rpm.

Typical measure of a refrigeration compressor performance is given by COP, which is proportional to $\eta_v, \eta_{ad}, \eta_{mech}, \eta_{mt}$. Since the three compressor models are quite different in structure and compression method, compressor performance will be evaluated in terms of separate individual efficiencies. Computer simulation programs developed for each type of compressor[11] have been used to calculate all the



(a) scroll (b) two-stage twin rotary (c) reciprocating

Fig. 1. Schematics of various hermetic compressors.

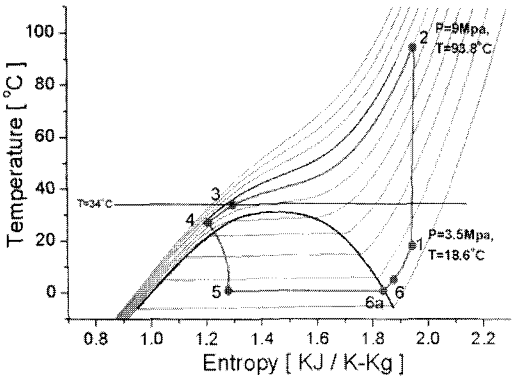


Fig. 2. Heat pump water heater cycle with CO_2 in a T-s diagram.

efficiencies. Two main calculation modules of the computer simulation programs are for the calculation of the gas state in the compression chamber and solving of the dynamics of moving elements. Gas pressure is obtained by Eq. (1)

$$P = P_s\left(\frac{M}{\rho_s V}\right)^n, \quad M = M(0) - \int^t \sum \dot{m}_{lk}\, dt \tag{1}$$

where $\sum \dot{m}_{lk}$ is the sum of leakage mass flow rates in various leakage paths. When leakage clearance is very small compared to leakage length, quasi 1-D model can be applied to calculate leakage flow rate for compressible and viscous flows with quite accuracy[12]. For a computer simulation for a whole compressor, direct solving of quasi 1-D model at every time step requires enormous calculation time. Hence, the flow coefficient c_v defined by Eq. (2) was first calculated over various pressure ratios P_2/P_1 and clearance height to length ratios ε/t or ε/R.

$$c_v(\varepsilon/t,\, P_2/P_1) = \dot{m}_a/\dot{m}_i \tag{2}$$

where $\dot{m}_a$ is the mass flow rate calculated by quasi 1-D model and $\dot{m}_i$ is the ideal orifice flow described by Eq. (3).

$$\dot{m}_i = \frac{AP_1}{Z_c}\sqrt{\frac{2n}{n-1}\frac{1}{RT_1}} \cdot \sqrt{(P_2/P_1)^{n/2} - (P_2/P_1)^{(n+1)/n}} \tag{3}$$

For every time step in the simulation program, ideal orifice flow rate, $\dot{m}_i$, is first calculated according to Eq. (3), and actual mass flow rate $\dot{m}_a$ is then obtained by the product of $\dot{m}_i$ and c_v according to Eq. (2), while c_v being taken from pre-calculated data set with given pressure ratio and clearance.

Reaction forces among the moving elements are obtained by solving the dynamics of the moving elements. For journal bearings, the friction coefficient is obtained as a function of Sommerfeld number and bearing slenderness. For boundary lubrication, it varies from 0.013 to around 0.1, depending on the sliding condition. Formulation of the dynamic equations for the moving elements and calculation method for reaction forces and frictional losses are found in Kim.[11]

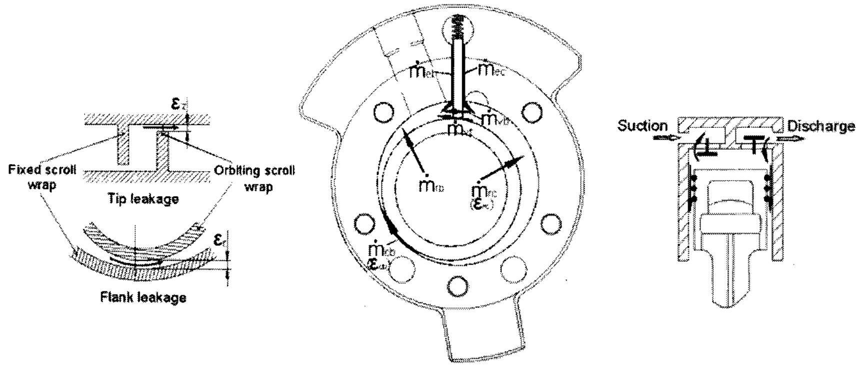
The rotary compressor has only discharge valve, and the reciprocating compressor has both of suction and discharge valves. Present scroll compressor uses no valve.

3. Volumetric efficiency

Fig. 3 shows leakage paths for each compressor. In order to obtain reference values for leakage characteristics for each compressor, the computer simulations were first run at zero leakage clearance for the three compressors. Volumetric efficiencies at zero clearance were 100%, 94.15%, and 94.37% for the scroll, twin rotary and reciprocating compressors, respectively. Clearance volume ratio of the present reciprocating compressor was γ =5%. From now on, all the volumetric efficiencies will be normalized by these zero clearance volumetric efficiencies, and the normalized efficiencies will be indicated by the superscript '*'. The adiabatic efficiency will also be normalized by the corresponding value at zero clearance for each compressor.

Fig. 4 shows typical leakage pattern from a compression pocket through the tip clearance in the scroll compressor. Leakage flow rate was normalized by theoretical suction mass flow rate. Instantaneous value of the leakage reaches about 14% at $\varepsilon_z = 15\,\mu$m. Fig. 5 shows the influence of the clearance on the volumetric efficiency for the scroll compressor. The volumetric efficiency η_v^* decreases rapidly with ε_z, but it is slightly affected by ε_r. It is simply due to the fact that the flank leakage area is smaller than the tip leakage area.

Fig. 6 shows leakage flows through various leakage paths of the two-stage twin rotary compressor. For the first stage in Fig. 6(a), leakage from the roller inside to the suction chamber $\dot{m}_{rb}$ increases continuously with the crank angle, since the leakage area increases



(a) (b) (c)

Fig. 3. Leakage paths : (a) scroll; (b) twin rotary; (c) reciprocating.

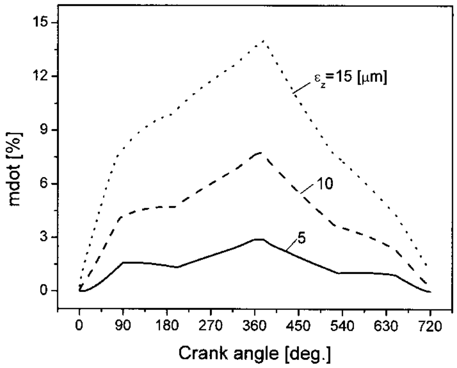


Fig. 4. Tip leakage in the scroll compressor: $\varepsilon_r = 10\ \mu m$

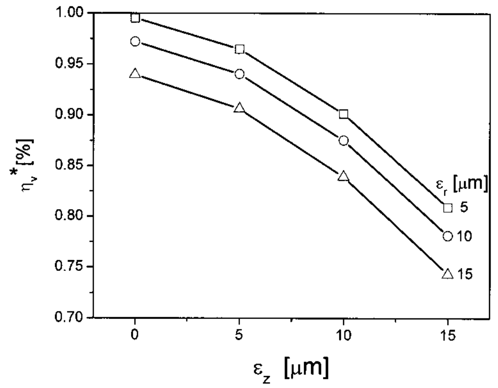


Fig. 5. Volumetric efficiency for the scroll compressor.

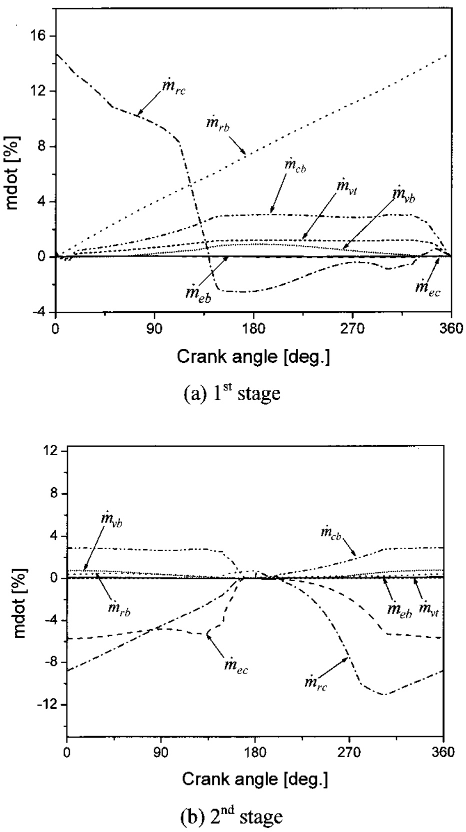
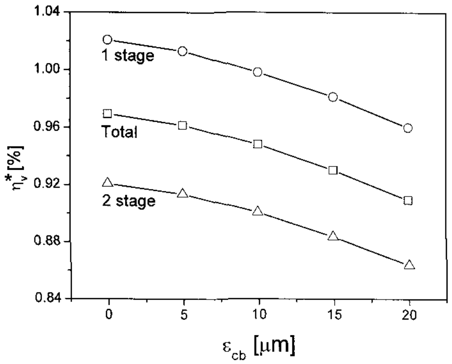


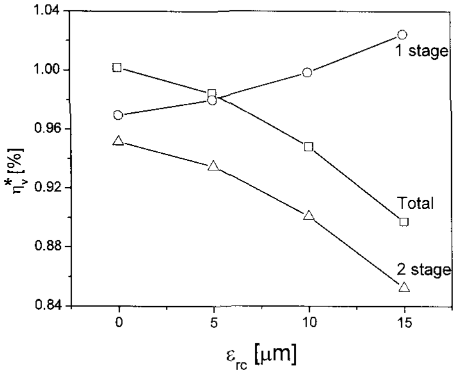
Fig. 6. Various leakages in the twin rotary compressor.

with θ, and leakage from the roller inside to the compression chamber $\dot{m}_{rc}$ is positive and large until the gas pressure in the compression chamber reaches the shell pressure P_m. Leakages around the vane, $\dot{m}_{vt}$, $\dot{m}_{vb}$, $\dot{m}_{eb}$, and $\dot{m}_{ec}$ are relatively small. For the second stage in Fig. 6(b), $\dot{m}_{rc}$, the leakage through ε_{rc}, is large and always negative, indicating that it always flows from the compression chamber to the roller inside since the pressure in the second cylinder is always larger than the shell pressure. Also $\dot{m}_{ec}$, the leakage through ε_{ec} becomes large and negative in the second stage.

Fig. 7 shows the influence of the clearance on the volumetric efficiency for the twin rotary. With



(a) $\varepsilon_{rc} = 10\,\mu m$



(b) $\varepsilon_{c} = 10\,\mu m$

Fig. 7. Volumetric efficiency for the twin rotary compressor.

increasing ε_{cd}, η_v^* decreases by about 5% for both stages. For the first stage, η_v^* becomes larger than 1 at ε_{cb}=0. It is because there is an incoming leakage flow to the compression chamber through ε_{rc}. Influence of ε_{rc} on η_v^* is rather complicated: As ε_{rc} increases, η_v^* decreases in the second stage, but increases in the first stage. While the leakage through ε_{cb} always reduces the mass of the gas in the compression chamber, the mass in the compression chamber is reduced by the leakage through ε_{rc} in the second stage, but increased in the first stage as indicated by $\dot{m}_{rc}$ in Fig. 6(a).

Fig. 8 shows the influence of the clearance on the volumetric efficiency for the reciprocating compressor. Effect of clearance volume ratio γ on η_v^* is less for CO_2 than for other refrigerants, since the pressure ratio is small for CO_2. According to $\eta_v = 1-\gamma[(P_d/P_s)^{1/n}-1]$, η_v^* decreases by 4.3% by changing γ from 3% to 7%. The volumetric efficiency η_v^* is sensitive to the leakage clearance such as ε_{sv}, ε_{dv}, and ε_{pr}. At γ=5%, η_v^* decreases by 11% by increasing leakage clearances by 5 μm.

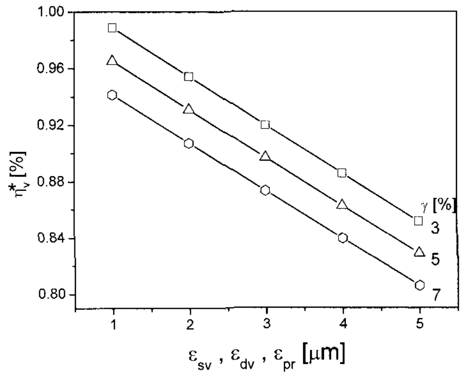


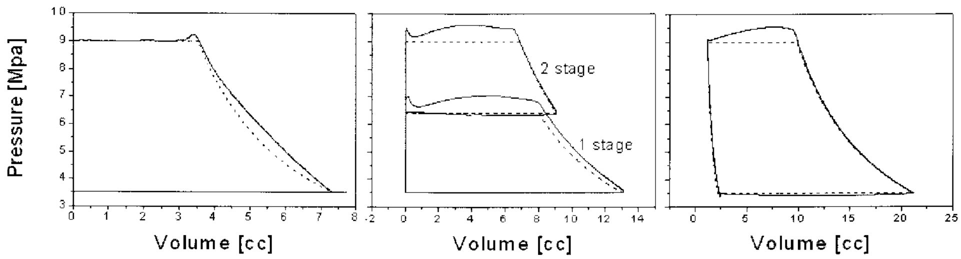
Fig. 8. Volumetric efficiency for the reciprocating compressor.

The volumetric efficiency is also affected by thermal effect: CO_2 is sensible to superheat, and increase in the specific volume due to suction gas superheat is considerable[(13)]. This thermal effect, however, could not be quantified in the present study, since estimation of the suction gas superheating requires heat transfer modeling for the whole compressor. Qualitatively speaking, direct suction would help reduce suction gas heating. In this sense, a rotary type compressor would have an advantage.

4. Adiabatic efficiency

Fig. 9 shows P-V diagrams for the scroll, two-stage twin rotary, and reciprocating compressors. The scroll compressor has small over-compression loss for design condition, and some loss occurs during compression process. The two-stage twin rotary shows relatively large over-compression loss, and re-expansion loss for the first stage is noticeable. For the reciprocating compressor, certain amount of over-compression loss and small suction loss are found. Actually these losses are very much dependent on the valve design. Here ideal ring plate type valves are assumed: Large enough flow passages are provided for the suction and discharge, resulting in relatively small throttling losses.

For the adiabatic efficiency calculation, again, calculations were made at zero-clearances for each compressor: these are 97.73%, 84.12%, and 93.03% for the scroll, twin rotary and reciprocating compressors, respectively. These values were used to normalize the adiabatic efficiencies for the corresponding compressors. Fig. 10 shows the adiabatic efficiency for the scroll compressor. η_{ad}^* decreases with increasing ε_r but effect of ε_z becomes small at large ε_r. At typical



(a) (b) (c)

Fig. 9. P-V diagrams : (a) scroll ($\varepsilon_r = 10\mu m$, $\varepsilon_z = 10\mu m$) (b) two-stage twin rotary ($\varepsilon_{cb} = 10\mu m$, $\varepsilon_{rc} = 10\mu m$) (c) reciprocating compressor (γ =3%, ε_{sv} , ε_{dv} , $\varepsilon_{pr} = 3\mu m$)

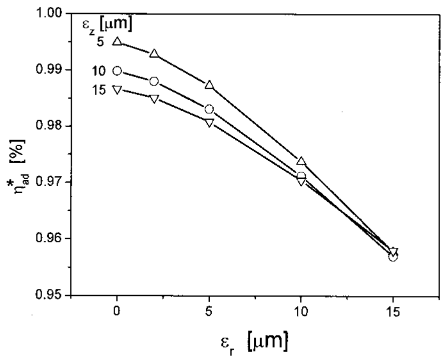
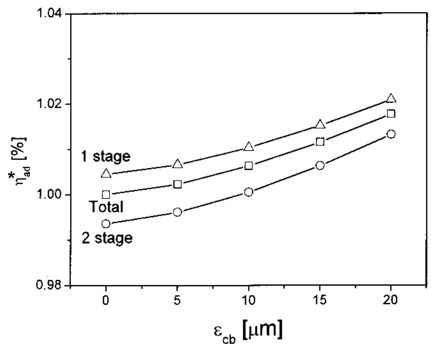
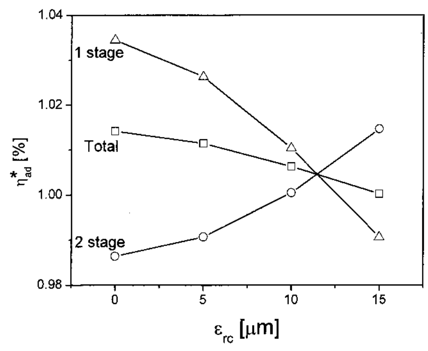


Fig. 10. Adiabatic efficiency for the scroll compressor.



(a) $\varepsilon_{rc} = 10\mu m$



(b) $\varepsilon_{cb} = 10\mu m$

Fig. 11. Adiabatic efficiency for the twin rotary compressor.

clearances such as ε_z = 10 μm and ε_r = 10 μm , η^*_{ad}=97.2%.

Fig. 11 shows the adiabatic efficiency for the twin rotary. The influence of the leakage clearance is rather complicated: Increase in ε_{cb} increases η^*_{ad} both in the first and second stages Increase in ε_{rc} causes η^*_{ad} decreased in the first stage, but it causes η^*_{ad} increased in the second stage. Generally, leakage reduces the gas mass in the compression chamber, resulting in less work to compress the gas. In the first stage, however, there is a significant mass flow rate $\dot{m}_{rc}$ from the roller inside to the compression chamber during the compression process as indicated in Fig. 6(a).

Fig. 12 shows the adiabatic efficiency for reciprocating compressor. Effects of the leakage clearance and clearance volume ratio on η^*_{ad} are small.

5. Mechanical efficiency

Fig. 13 shows breakdown of mechanical loss for the scroll, twin rotary and reciprocating compressors. For reference of this analysis, useful work is taken as 100 for each compressor. For the scroll, total mechanical loss amounts to about 12.71% of the useful

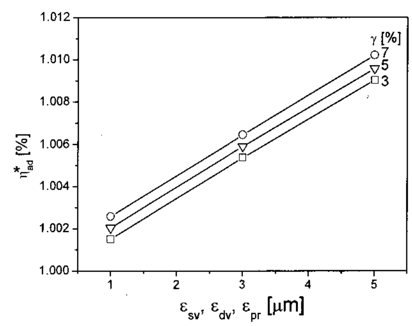


Fig. 12. Adiabatic efficiency for the reciprocating compressor.

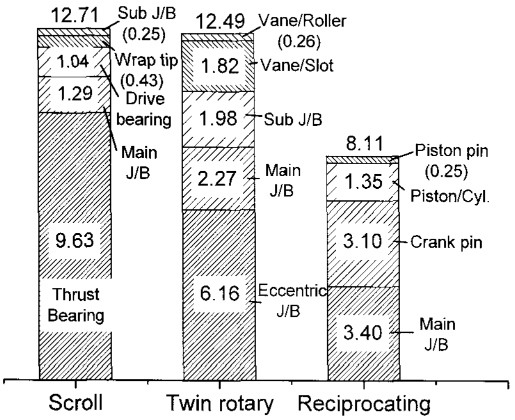


Fig. 13. Mechanical loss breakdown: scroll, two-stage twin rotary, reciprocating.

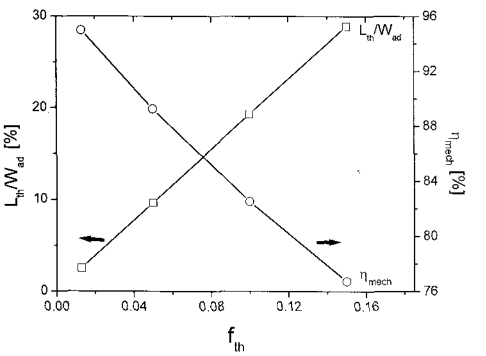


Fig. 14. Effect of lubrication condition on thrust bearing loss for the scroll compressor.

work, substantial portion of which is the thrust bearing loss. For the twin rotary mechanical loss is 12.49%, half of which is consumed at eccentric bearings. Mechanical loss of the reciprocating compressor is relatively small. In fact, thrust bearing loss for the scroll compressor depends on the thrust surface condition. Due to increased axial gas force with CO_2, hydro-dynamic lubrication condition on the thrust surface may not be sustained, and large friction coefficient is expected. Fig. 14 shows change of the thrust surface loss with friction coefficient. For f_{th}=0.15, thrust loss becomes about 29% of the useful work with corresponding mechanical efficiency of η_{mech}=76.5%.

6. Torque variation

Fig. 15 shows torque variation with the crank angle for the scroll, twin rotary, and reciprocating compressors. Torque of the scroll shows very

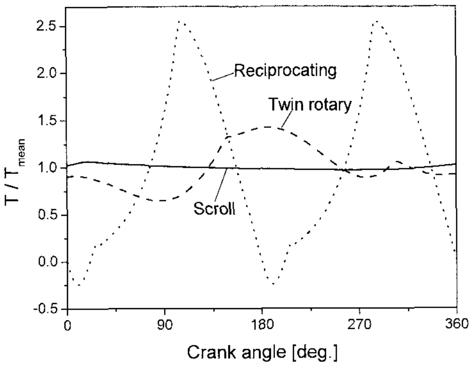


Fig. 15. Comparison of torque fluctuation for scroll, two-stage twin rotary, and reciprocating compressors.

smooth variation, peak-to-peak variation of the reciprocating compressor is about 250%, and that of rotary is in between.

7. Conclusions

In comparing performance of scroll, two-stage twin rotary, and reciprocating compressors whose displacement volumes are 13~21 cc (one cylinder for the reciprocating compressor) for CO_2 heat pump water heater by using computer simulation program,

(1) For zero clearance, the scroll compressor shows η_v =100%, since there is no re-expansion, while it is around 94 % for the twin rotary and reciprocating compressors with clearance volume ratio of 3%, and 5%, respectively.

(2) For the scroll compressor, tip clearance has a significant effect on η_v : Increment of ε_z by 5 μm reduces η_v by 7~9%. For the reciprocating compressor, even small clearance around valves easily reduces η_v : 1 μm valve clearance reduces η_v by 3%. The twin rotary compressor shows the lowest η_v .

(3) Adiabatic efficiencies are about 97.73%, 84.12%, and 93.03% at zero clearances for the scroll, twin rotary, and reciprocating compressors. Effect of clearance on η_{ad} is generally not large for all type compressors.

(4) The reciprocating compressor has better mechanical efficiency than the other two. The most severe drawback of the scroll compressor by using CO_2 is a significant increase in the mechanical loss at the thrust surface supporting the orbiting scroll member.

(5) The scroll compressor has very smooth torque variation, the reciprocating compressor has a

peak-to-peak variation of 250%, and the rotary is in between.

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