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Mode change design for capacity modulation in reciprocating compressor

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Abstract

Due to environmental issues, the development of low energy consumption products has become one of the main topics in the home appliance industry. The energy consumption of a refrigerator depends on the efficiency of its compressor as well as on the refrigerator cycle design, such as the capacity modulation. This study features the design of a novel capacity modulation reciprocating compressor, i.e., two-step capacity modulation (TCM). In a TCM compressor, capacity modulation is achieved by changing the dead volume in the cylinder. Instead of a concentric sleeve, an eccentric sleeve is used to change the dead volume for the clockwise and counterclockwise rotation of a motor. For stable capacity modulation, a new latching system with a key, a spring, and an eccentric sleeve is introduced, and the mode change reliability is verified by dynamic analysis.

Keywords: Capacity modulation; Reciprocating compressor; Energy saving

1. Introduction

After remarkable improvement of the compressor efficiency, the compressor technology is nearly saturated. Hence, in an effort to reduce energy consumption, many researchers in the refrigeration and air-conditioning industry have focused on the use of capacity modulation techniques. Capacity control reduces the on/off cycling losses of the equipment and improves compressor reliability due to the lower pressure difference at partial capacity operation [1]. There are many capacity control techniques, namely, the bypass method [2, 3], multiple compressor usage [4], idle rotation of the cylinder [5] and variable speed control [6-8]. The characteristics of these capacity control techniques are well described in a review paper by Qureshi and Tassou [9].

Capacity control methods are mainly applied to large-capacity systems because, compared to the overall costs, the additional compressor costs for capacity control are insignificant. In small-capacity systems, especially in a refrigerator, the implementation of capacity control techniques is difficult because the additional costs are a higher portion of the overall costs. For large-capacity systems, capacity modulation can be easily implemented by using multiple compressors [4]. Therefore, most capacity control methods have been implemented in the air-conditioning area, especially in large-capacity system air-conditioners.

Variable speed compressors, such as the inverter compressor, are mainly used for capacity control in refrigerators [10, 11], though the cost of the inverter system is too high. Many compressor researchers are therefore trying to develop a mechanical capacity modulation compressor. In fact, the mechanical capacity modulation compressor was studied earlier than the inverter compressor by Westinghouse Electric Corp. [12]. After the development of inverter

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compressors in the 1980s, mechanical capacity modulation compressors lost their merits. These days, however, due to the high costs of the inverter compressor, the interest in the mechanical capacity modulation is increasing. Powers [5] developed a capacity modulation reciprocating compressor (TS compressor) for air-conditioning systems. The TS compressor has two cylinders, and both cylinders compress the refrigerant at full capacity operation, though only one cylinder compresses the refrigerant at the partial capacity mode of operation.

In the design of a mechanical capacity modulation compressor, dynamic analysis is needed to get a stable mode change performance. To date, however, there have been no reports on dynamic analysis of capacity modulation. Instead, many dynamic analyses have been performed to investigate the vibration characteristics and performance improvement. Kim [12, 13] did a dynamic analysis of a reciprocating compression mechanism. He considered the viscous frictional force between the piston and the cylinder wall to determine the coupled dynamic behavior of the piston and the crankshaft. Furthermore, Kim et al. [15] and Park et al. [16] analyzed the dynamic behavior of the scroll compressor and the rotary compressor, respectively.

We developed a novel capacity modulation reciprocating compressor (TCM) to reduce energy consumption in a refrigerator. The TCM compressor controls the cooling capacity with only one cylinder, by varying the piston stroke at the forward and reverse rotation of the motor. We designed a new latching system with a key, a spring and an eccentric sleeve, and we verified the latching reliability by conducting dynamic analysis of the eccentric sleeve.

2. Capacity modulation design

Capacity modulation in the TCM compressor is achieved by changing the dead volume of the cylinder for the clockwise and counterclockwise rotation of a crankshaft. To change the dead volume, we used an eccentric sleeve instead of a concentric sleeve. Fig. 1 shows a schematic illustration of the change of dead volume in the clockwise and counterclockwise rotation, where r represents the eccentricity between the crankshaft pin and the crankshaft axes and e represents the eccentricity of the eccentric sleeve. With the parameters, the piston stroke is 2 $(r+e)$ for the clockwise rotation and 2 $(r-e)$ for the counterclockwise rotation. Thus, the top clearance difference between the clockwise rotation and the counterclockwise rotation is $2e$. When the crankshaft rotates clockwise, the top clearance is small and the flow rate of refrigerant is large; hence, the cooling capacity is large (as shown in Fig. 2(a) for the full capacity mode of operation). When the crankshaft rotates counterclockwise, the top clearance increases by $2e$ more than in the clockwise rotation, consequently leading to a decrease in the flow rate of the refrigerant as well as the cooling capacity (for a partial capacity operation). The cooling capacity can therefore be modulated by changing the rotating direction of the crankshaft. Furthermore, the modulation ratio depends on the eccentricity of the eccentric sleeve. Fig. 2(b) shows the refrigerator operating period at the full and the partial capacity mode. When a refrigerator operates at the partial capacity mode, lower power is consumed than that at the full capacity mode. And, to maintain the cooling temperature, the compressor has to be operated longer at the partial capacity mode than at the full capacity mode. But because the cycle efficiency of a refrigerator is higher at the partial capacity mode, the power consumption is lower at the partial capacity mode of operation than at the full capacity mode of operation.

For stable capacity modulation, the eccentric sleeve should be fixed in a given position at the full and the partial capacity mode of operation, respectively, as shown in Fig. 1. We designed a new latching system

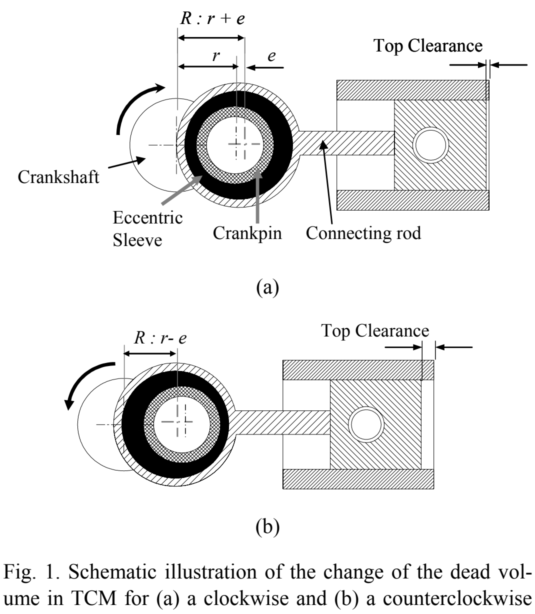
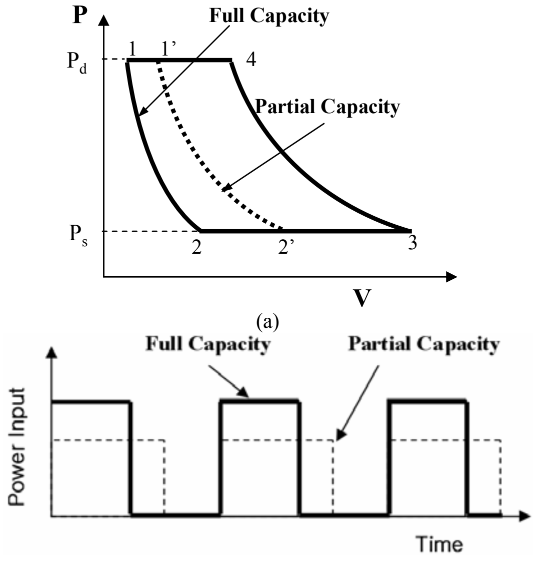


Fig. 1. Schematic illustration of the change of the dead volume in TCM for (a) a clockwise and (b) a counterclockwise rotation of the crankshaft



(a)

(b)

Fig. 2. (a) Pressure-volume diagram and (b) refrigerator operating time at full and partial capacity operations

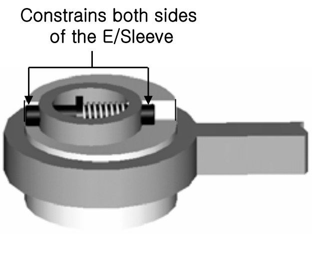
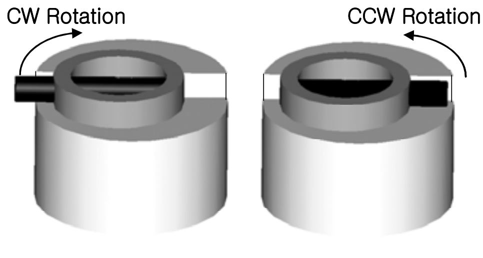


Fig. 3. A new latching system design with an eccentric sleeve, a key and a spring

with a latching key, a spring, and an eccentric sleeve (Fig. 3) for stable capacity modulation from partial mode to full mode or from full mode to partial mode. At first, the spring is not used and only the key and the eccentric sleeve (E/sleeve) are used, as shown in Fig. 4, for productivity. To investigate the stability of the key position we carried out static force analysis. The pressure in the cylinder is calculated as follows:

$$P = P_s (V_s / V)^k \,, \tag{1}$$

where P_s is the suction pressure, V_s is the refrigerant volume in the cylinder after the suction is completed, and k is the compression coefficient. Fig. 5 shows the calculated pressure. The gas force by the compressed refrigerant is calculated by multiplying the piston area by the pressure. Fig. 6 shows a free body diagram for



(a) (b)

Fig. 4. The first design of a latching system without a spring for (a) the partial and (b) the full capacity mode of operation.

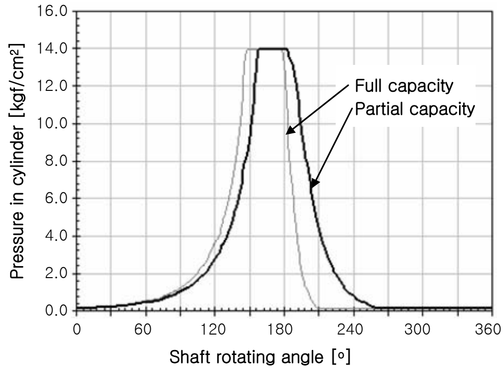
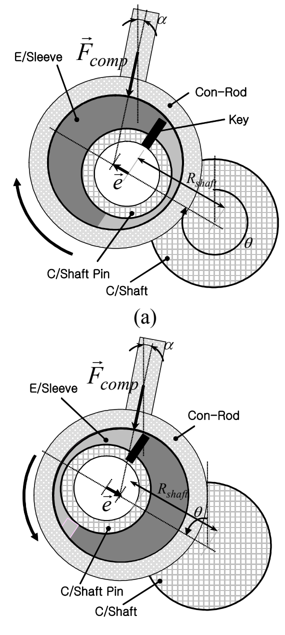


Fig. 5. Diagram of the Pressure-shaft rotating angle.



(b)

Fig. 6. Schematic illustration of the force acting on the eccentric sleeve for (a) the full and (b) the partial capacity mode of operation.

the eccentric sleeve. If the friction force between the eccentric sleeve and the connecting rod is neglected, we can calculate the moment about the center of crankshaft pin as follows:

$$\vec{M} = \vec{e} \times \vec{F}_{comp} , \qquad (2)$$

where $\vec{e}$ is the position vector of the center of the eccentric sleeve from the center of the crankshaft pin and $\vec{F}_{comp}$ is the force induced by the compressed refrigerant which is calculated from the equilibrium equation for the connecting rod. Fig. 7 shows the calculated moment that is applied on the eccentric sleeve. When the moment is greater than or equal to zero, the eccentric sleeve keeps in contact with the key. When the moment is less than zero, however, the eccentric sleeve cannot keep in contact with the key anymore and the cooling power of the refrigerator fluctuates. In the full capacity mode of operation, the moment is greater than zero in most ranges of the rotating angle of the crankshaft. In the partial capacity mode of operation, however, the moment is less than zero in most ranges of the rotating angle of the crankshaft. This results means that the eccentric sleeve tends to come apart from the key and the cooling capacity is not maintained at the partial capacity mode of operation. Moreover, the eccentric sleeve may collide with the key when the moment becomes positive from negative; the collision can then induce a serious reliability problem.

To overcome this problem, we used a spring as shown in Fig. 3 and Fig. 8. Fig. 8 shows the mode change procedures. Fig. 8(a) shows the initial position of the key and the eccentric sleeve when the compressor is at rest. The key is located inside a hole in the crankshaft pin, and therefore rotates at the same speed as the crankshaft. When the compressor is at rest, the key is always positioned by a spring at the inner most center of the crankshaft rotation. The eccentric sleeve can therefore be positioned anywhere because, as shown in Fig. 8(a), there is no constraint in one side of the eccentric sleeve. When the crankshaft starts to rotate, as shown in Fig. 8(b), the key rotates with the crankshaft. The eccentric sleeve, however, nearly keeps its rotational position due to inertia. The only driving moment for the eccentric sleeve rotation is

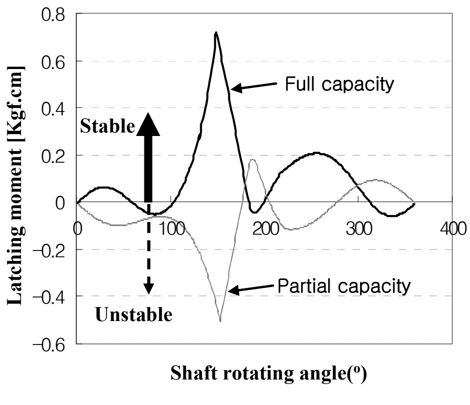


Fig. 7. Contacting moment between the eccentric sleeve and the key.

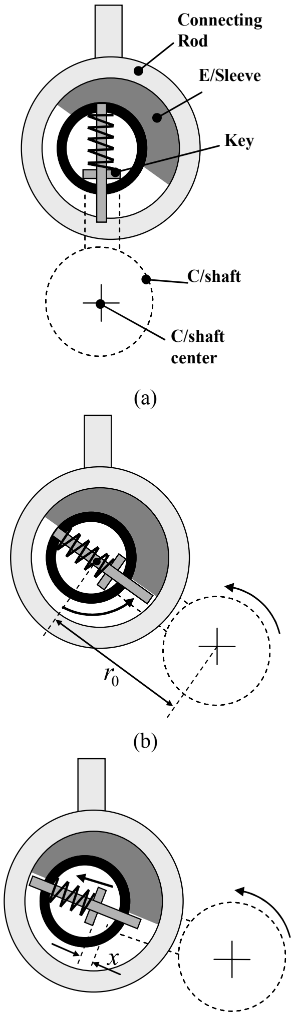


Fig. 8. Schematic explanation of the mode change processes for (a) the at rest (stop) position; (b) the movement of the key to the eccentric sleeve; the start of the rotation; (c) the outward movement of the key by centrifugal force.

given by the friction force that is induced by the relative movement between the eccentric sleeve and the crankshaft pin and between the eccentric sleeve and the connecting rod. These effects are described in detail in Chapter 3. As the crankshaft rotates, the key comes into contact with the eccentric sleeve (Fig. 8(b)). And, as the rotating speed increases, the key moves outwards due to the centrifugal force (Fig. 8(c)). These three-step procedures of mode change are the main ideas of this study.

3. Dynamic analysis for the eccentric sleeve

3.1 Eccentric sleeve dynamics

As shown in Fig. 8(c), the key must be in contact with the eccentric sleeve for a while to ensure outward movement of the key. Due to the centrifugal force of the eccentric sleeve and the gas force by the compressed refrigerant, however, the key sometimes does not stay in contact with the eccentric sleeve. Consequently, the key does not move outwards and the mode change cannot be completed. To confirm reliability of the mode change, we therefore did a dynamic analysis for the eccentric sleeve and the key.

Fig. 9 shows the forces that have an effect on the rotation of the eccentric sleeve about the center of the crankshaft pin. Four types of force have an effect on the rotation of the eccentric sleeve: the friction force between the eccentric sleeve and the connecting rod; the friction force between the eccentric sleeve and the crankshaft pin; the centrifugal force, and the inertial

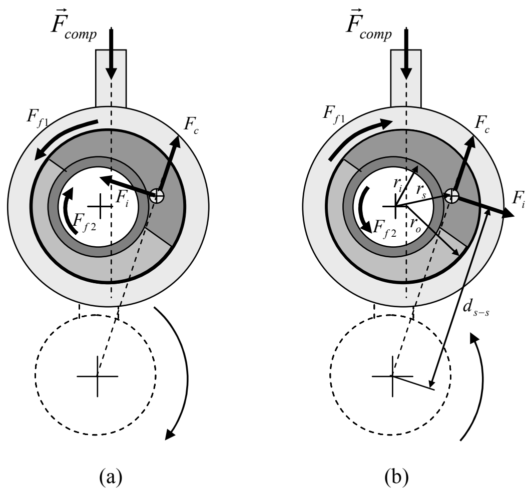


Fig. 9. Free body diagram for the eccentric sleeve for (a) the full and (b) the partial capacity mode of operation.

force of the eccentric sleeve. To calculate the moments by friction force, we used the Coulomb friction law as follows:

$$M_{f1} = -r_o F_{f1} = -r_o \mu_1 F_{comp} \tag{3}$$

$$M_{f2} = r_i F_{f2} = r_i \mu_2 F_{comp}\,, \tag{4}$$

where M_{f1} is the moment by the connecting rod, M_{f2} is the moment by the crank shaft pin, r_o is the outer radius of the eccentric sleeve, r_i is the inner radius of the eccentric sleeve, F_{comp} is the gas force by the compressed refrigerant, μ_1 is the friction coefficient between the eccentric sleeve and the connecting rod, and μ_2 is the friction coefficient between the eccentric sleeve and the crankshaft pin. A positive value of the moment means a clockwise direction and a counter-clockwise direction in a full capacity mode of operation and a partial capacity mode of operation, respectively. The friction coefficients μ_1 and μ_2 are calculated from a finite difference analysis considering shaft dynamics [17]. In the calculation of friction coefficient, we used the temperature-dependent oil viscosity and considered the bearing clearance. Thus, the friction coefficient is dependent on the temperature and the clearance. Moreover, centrifugal force is induced when the eccentric sleeve rotates about the center of the crankshaft. We calculated the centrifugal force as follows:

$$F_c = m_{sleeve} d_{s-s} \varpi^2\,, \tag{5}$$

where m_{sleeve} is the mass of the eccentric sleeve, d_{s-s} is the distance between the mass center of the sleeve and the rotation center of the crank shaft and ϖ is the angular speed of the crankshaft. The direction of the centrifugal force is outward from the rotation center of the crankshaft to the mass center of the eccentric sleeve. The moment by the centrifugal force is then given as

$$M_c = -\vec{r}_s \times \vec{F}_c\,, \tag{6}$$

where $\vec{r}_s$ is the position vector of the mass center of the eccentric sleeve from the center of the crank shaft pin and $\vec{F}_c$ is the force vector of the centrifugal force. When the compressor starts to run, the rotation speed of crankshaft increases until its speed becomes about 60 Hz. Thus, the eccentric sleeve also experiences the accelerating motion. The inertia force by acceleration is calculated as follows:

$$F_a = m_{sleeve} d_{s-s} \alpha \,, \tag{7}$$

where α is the angular acceleration of the crankshaft. The direction of the inertia force is normal to the direction of the centrifugal force and opposite to the acceleration direction. Thus, the moment by the inertia force about the center of the crankshaft pin is given as

$$M_a = -\vec{r}_s \times \vec{F}_a \,. \tag{8}$$

To calculate the rotational acceleration and the speed of the crankshaft, we carried out dynamic analysis for the crankshaft considering the motor torque and the piston force by the compressed refrigerant. The reliability of the crankshaft dynamics is validated by the measured data and is mentioned in Chapter 4.

As a result, we can calculate the total moment that is acting on the eccentric sleeve about the center of the crankshaft pin as follows:

$$M_{sleeve} = M_{f1} + M_{f2} + M_c + M_a \,. \tag{9}$$

Next, we can calculate the dynamic behavior of the eccentric sleeve by using the following forward explicit time integration scheme:

$$\alpha_{sleeve}(t) = M_{sleeve}(t) / J_{sleeve} \tag{10}$$

$$\varpi_{sleeve}(t+\Delta t) = \varpi_{sleeve}(t) + \alpha(t)\cdot\Delta t \tag{11}$$

$$\theta_{sleeve}(t+\Delta t) = \theta_{sleeve}(t) + \varpi(t)\cdot\Delta t + 0.5\alpha(t)\cdot(\Delta t)^2 \,. \tag{12}$$

The forward explicit time integration scheme shows different results for Δt and sometimes gives an oscillating solution. We determined the time step, Δt, to be sufficiently small to give a stable solution. The dynamic equations for the analysis of the partial capacity mode of operation can be obtained in the same way as that of the full capacity mode of operation.

When the eccentric sleeve collides with the key, the eccentric sleeve bounces out of the key. The speed of the eccentric sleeve after the collision is obtained by using the coefficient of restitution as follows:

$$v_{sleeve}^2 = v_{key}^2 + e_c(v_{key}^1 - v_{sleeve}^1) \,, \tag{13}$$

where the superscripts '1' and '2' are the status of 'before' and 'after' the collision, respectively. For the restitution coefficient, we used a value of 0.7.

3.2 Spring design

As shown in Fig. 8, the key must move outwards to constrain the eccentric sleeve on both sides. The motion of the key depends on the spring constant and the centrifugal force of the key. The average displacement of the key is determined from the force equilibrium as follows:

$$k(x_0 + x) = m_{key}\varpi^2(r_0 + x) \tag{14}$$

$$x = \frac{kx_0 - m_{key}\varpi^2 r_0}{m_{key}\varpi^2 - k} \,, \tag{15}$$

where x is the key displacement, x_0 is the initial spring compression, k is the spring constant, m_{key} is the mass of the key, and r_0 is the initial distance between the mass center of the key and the rotation center. The mass of the key is given from the geometry and density. From the empirical investigation, the design objectives for the spring are given as follows:

1) The maximum key displacement should be 3 mm to constrain the eccentric sleeve on both sides.

2) The key must move to the desired position (x=3 mm) when the crankshaft has an angular velocity of 20Hz.

It is very difficult to satisfy both of the design objectives. The second objective can be satisfied by deciding the proper spring constant. If we choose a spring constant to satisfy the second objective, the maximum key displacement will be far greater than 3.0mm when the crankshaft rotates at an angular velocity of 60Hz. To satisfy the first objective, therefore, we should design the fully compressed spring length to have a given value. The number of turns, the wire diameter and the spring diameter are determined to satisfy both objectives.

4. Design of the eccentric sleeve through dynamic analysis

The motion of the eccentric sleeve and the latching behaviors are investigated by dynamic analysis. Fig. 10 shows the rotational velocity of the crankshaft when the compressor starts to run. The analysis results were obtained from the crankshaft dynamics

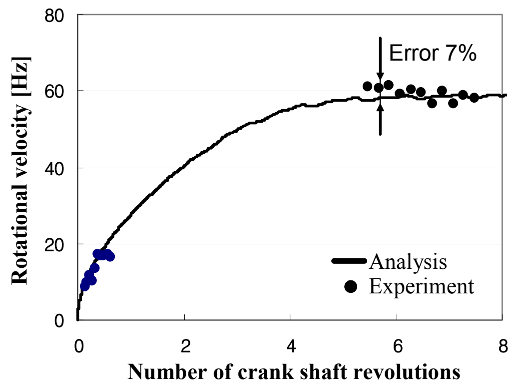


Fig. 10. Rotational velocity of the crank shaft when a compressor starts to run.

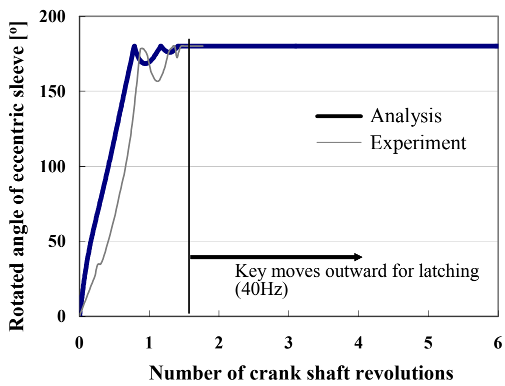
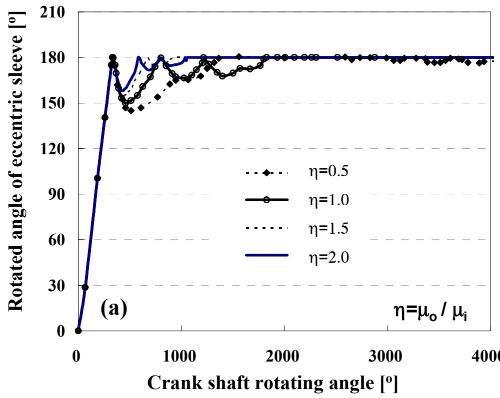


Fig. 11. Rotated angle of the eccentric sleeve for full capacity operation.

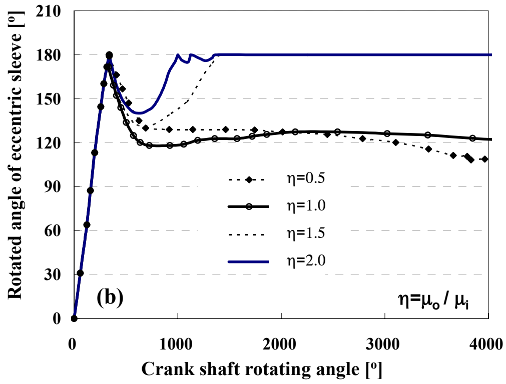
considering gas force and motor torque. The dynamic analysis was validated by experimental data. It was very difficult to measure the dynamic behaviors of the crankshaft and the eccentric sleeve in normal refrigerant condition. Therefore, the experiment was performed in the air at room temperature and, then, the top cover of the test compressor was made of transparent glass. The discharge pressure is controlled by using a capillary tube. Fig. 15 shows the prototype used in the test. The dynamic behavior of the crankshaft and the eccentric sleeve was captured with a high-speed camera and the angular velocity was calculated by using the captured image. In Fig. 10, it is shown that the analysis predicts the experimental results accurately. The calculated angular acceleration and velocity can therefore be reliably used in the dynamic analysis of the eccentric sleeve.

Fig. 11 shows the motion of the eccentric sleeve at the full capacity mode operation. The ordinate refers to the relative rotating angle of the eccentric sleeve to the key, that is, 180^{o}- (angle between the key and the eccentric sleeve). The eccentric sleeve comes into contact with the key when the angle becomes 180^{o}. Furthermore, the eccentric sleeve bounces out of the key after the first collision. We measured the motion of the eccentric sleeve with a high-speed camera and compared the measurements with the analysis results. The dynamic behavior of the eccentric sleeve predicted in the analysis is in good agreement with the measured data. When the crankshaft rotates about 1.5 turns, the eccentric sleeve keeps in contact with the key and the key moves outwards. When the rotation angle of the eccentric sleeve is less than 180^{o}, the key cannot move outwards because the high wall of the eccentric sleeve constrains the motion. Similarly, when the eccentric sleeve first collides with the key, the key cannot move outwards while the rotating angle is 180^{o}. This inability to move outwards occurs because the duration is too short to produce sufficient movement. For the outward movement of the key, two conditions must be satisfied. First, the eccentric sleeve must keep in contact with the key for sufficient time for the movement of the key. Second, the rotational velocity of the crankshaft must be greater than the given value, that is, the centrifugal force must be greater than the spring force. We therefore designed the spring so that the key moves outwards at a rotational velocity of 20Hz.

The inner and outer clearances of the eccentric sleeve are the main parameters that have an effect on the latching behavior because the friction coefficient is dependent on the clearance. The mass center of the eccentric sleeve is also a main design parameter because the centrifugal force and the inertial force are dependent on it. To investigate how the friction coefficient of the inner and outer clearances affects the latching behavior, we analyzed the latching behavior for various values of the friction coefficient, as shown in Fig. 12. The value of the friction coefficient at the inner clearance is fixed to 0.05. This value is obtained by finite difference analysis of bearings at room temperature. In the full capacity mode of operation, as shown in Fig. 12(a), the mode change and the latching are completed successfully in all four cases. In the partial capacity mode of operation, however, the latching is impossible when the coefficient ratio is less than 1.5. This impossibility means that the friction at the outer clearance has positive effects on latching, whereas the friction at the inner clearance has negative effects. The inner and outer clearances



(a)



(b)

Fig. 12. Dynamic behavior of the eccentric sleeve for various values of the friction coefficient for (a) the full and (b) the partial capacity mode operation.

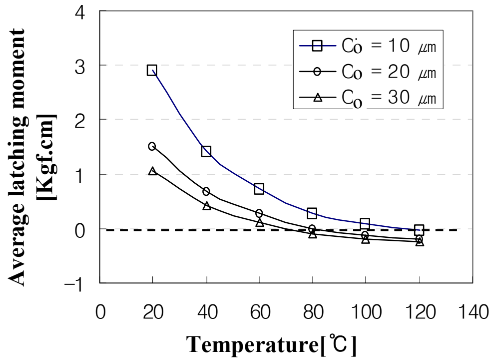


Fig. 13. Average latching moment for various values of the outer clearance and temperature.

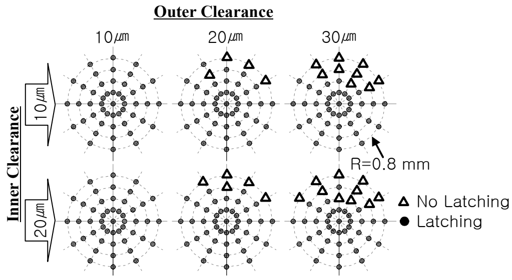
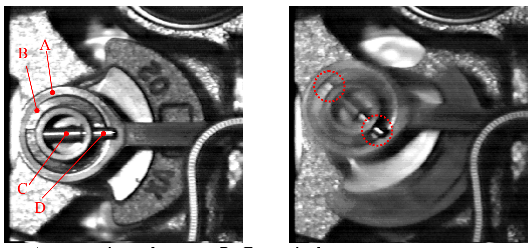


Fig. 14. Latching behavior for the mass center of the eccentric sleeve.



A : connecting rod B : Eccentric sleeve
C : spring D : key

(a) (b)

Fig. 15. Photograph of the prototype: (a) at rest and (b) after latching.

are therefore determined considering these results and the effect of the clearance on the efficiency of the compressor.

Another factor that affects the latching behavior, as well as the inner and outer clearances, is oil temperature. Fig. 13 shows the average moment on the eccentric sleeve for various values of clearance and temperature for the partial capacity mode of operation. The positive values of the moment mean that the latching has a greater possibility of success than failure. The inner clearance is fixed at 20μm. As shown in Fig. 13, when the temperature is greater than 100 °C, the mode change to the partial capacity mode is difficult. The outer clearance cannot be less than 15 μm due to a manufacturing problem, and usually the compressor operates at a temperature higher than 100 °C. We also investigated how the mass center of the eccentric sleeve affects the latching behavior. Fig. 14, which shows the analyzed results for various positions of the mass center, indicates that the mass center of the eccentric sleeve must be located in the lower left quadrant. Hence, we designed the eccentric sleeve so that the mass center was in the lower left quadrant and we confirmed the reliability of the mode change.

Fig. 15 shows the prototype of the new latching system and it was captured with a high-speed camera. In Fig. 15(a), the compressor is at rest and the key is in contact with only one side of the eccentric sleeve.

In Fig. 15(b), the compressor is rotating at 60 rpm and the key is moved outward by the centrifugal force. Therefore, the eccentric sleeve is latched by the key on both sides. With the newly developed TCM compressor, energy consumption is reduced by 10.1% at refrigerator condition.

5. Conclusion

To improve the efficiency of refrigerators, we developed a two-step capacity modulation method for a reciprocation compressor. The capacity modulation is achieved by changing the dead volume in the cylinder. We designed a new latching system with a key, a spring and an eccentric sleeve to change the dead volume. To confirm the reliability of the mode change, we carried out dynamic analysis for the eccentric sleeve while considering all of the forces acting on the eccentric sleeve. A comparison of the analysis results with the experimental results confirms reliability of the analysis. Furthermore, we investigated how temperature and the friction coefficient affect the latching behavior, and we found that the latching behavior in the partial capacity mode of operation is not good. Through dynamic analysis, we were able to design the inner and outer clearances and the mass center of the eccentric sleeve, and we confirmed the reliability of the mode change.

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