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Hyun-Jin Kim
Incheon National University, Korea, Republic of (South Korea), kimhj@incheon.ac.kr

Yong-Hee Kim
Incheon National University, Korea, Republic of (South Korea), dydgml007@incheon.ac.kr

Young-Sung Lee
Incheon National University, Korea, Republic of (South Korea), ys5941@incheon.ac.kr

Un-Seop Lee
Compressor R&D Group, Comp. & Motor Team, Digital Appliances, Samsung Electronics Co. Ltd, unseop.lee@samsung.com

Jeong-Bae Lee
Compressor R&D Group, Comp. & Motor Team, Digital Appliances, Samsung Electronics Co. Ltd, jb_lee@samsung.com

See next page for additional authors

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Authors

Hyun-Jin Kim, Yong-Hee Kim, Young-Sung Lee, Un-Seop Lee, Jeong-Bae Lee, and Yang-Hee Cho

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Hyun-Jin KIM[1*], Yong-Hee KIM[1], Young-Sung LEE[1],
Un-Seop LEE[2], Jeong-Bae LEE[2], Yang-Hee CHO[2]

[1]Department of Mechanical Engineering, University of Incheon,
Incheon, Korea
(+82-32-835-8419, +82-32-835-8410, kimhj@incheon.ac.kr)

[2]Compressor R&D Group, Comp. & Motor Team, Digital Appliances, Samsung Electronics Co. Ltd.,
Suwon, Kyunggido, Korea
(+82-31-200-6967, +82-31-200-6466, yangxi.cho@samsung.com)

* Corresponding Author

ABSTRACT

Performance analysis has been carried out on a high side scroll compressor having a fixed scroll equipped with a circular oil groove on its thrust surface. Oil is supplied to the oil groove through an intermittent opening from a high pressure oil reservoir formed inside the orbiting scroll hub. Oil in the groove is then delivered to both suction and back pressure chambers by pressure differentials and viscous pumping action of the orbiting scroll base plate. Mathematical modeling of this oil groove system has been incorporated into main compressor performance simulation program for optimum oil groove design. Pressure in the oil groove can be controlled by changing the oil passage area and oil groove configuration. With an enlarged oil passage, pressure in the oil groove increases due to increased flow rate, but pressure increase in the back pressure chamber is not that large, resulting in reduced friction loss at the thrust surface between the two scrolls. On the other hand, by increasing the oil passage area, oil content in the refrigerant flow increases, and the orbiting scroll stability could be negatively affected by oil groove pressure increase. Considering all these factors, EER could be improved by about 3.6% at ARI condition by optimum oil groove design.

1. INTRODUCTION

For axial compliance in a high-side scroll compressor, a back pressure chamber with combined high and intermediate pressure zones is widely employed to produce thrust force on the rear side of the orbiting scroll base plate. A seal-ring is often used to separate intermediate the pressure zone outside from the high pressure zone inside. Pressure in the intermediate chamber is determined by the balance between the in and out flows of the oil/refrigerant mixture. Out-gassing of the refrigerant dissolved in the oil also affects the pressure level.

To increase stability of the orbiting scroll, back pressure must be increased. But this increases friction loss in the thrust surface, requiring some trade-off between the stability and reduced friction in determining the back pressure level.

Some studies have suggested how to increase orbiting scroll stability, while producing little extra friction. Introduction of a full circular oil groove filled with high pressure oil has been suggested for the thrust surface of the orbiting scroll base plate to produce additional recovering force against the overturning moment (Furusho *et al.*, 2005). Several sections of circular grooves on the fixed scroll side have also been suggested to expand the compressor operating conditions (Chikano *et al.*, 2009). In this case, fluid in the back pressure chamber fills the grooves to reduce loss of injected fluid expansion loss. Hiwata *et al.* (2004) introduced a grooved-thrust bearing filled with suction gas to pull the orbiting scroll toward the fixed scroll in order to improve the overturning limit. A new mechanism for a compliant frame can also reduce thrust friction loss, while maintaining stability of the orbiting scroll (Fushiki *et al.*, 2002).

In this paper, we have performed a study on the effects of the oil groove design parameters on the compressor performance for a high-side scroll compressor.

2. STRUCTURE OF A HIGH-SIDE SCROLL COMPRESSOR WITH THRUST OIL GROOVE
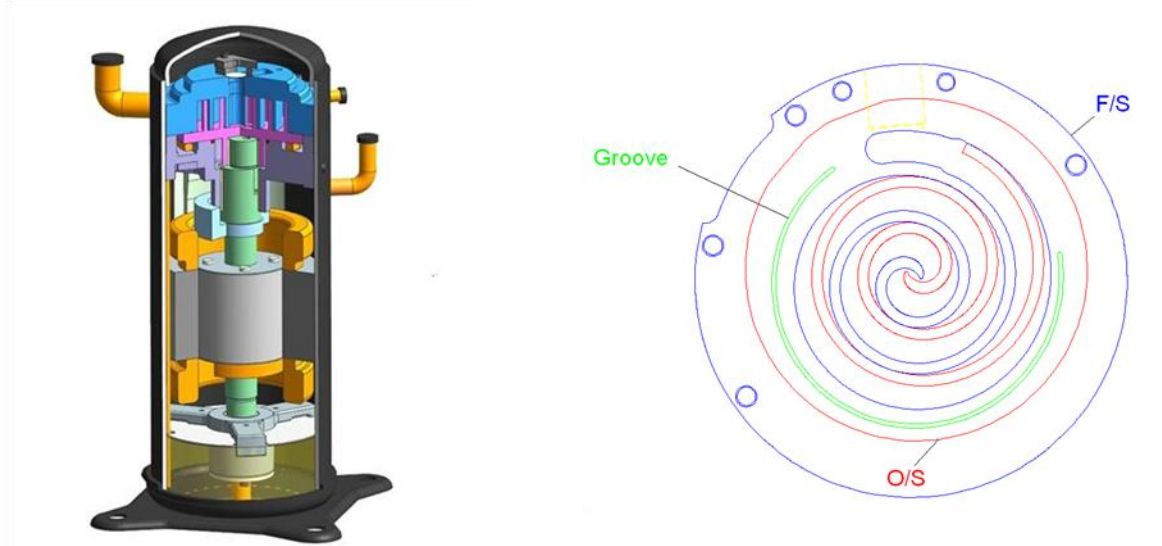

Figure 1: Cross sectional view of scroll compressor   **Figure 2:** Oil groove on thrust surface of the fixed scroll

Figure 1 shows a cut-away view of a high-side scroll compressor. A back pressure chamber is provided on the rear side of the orbiting scroll base plate for axial compliance. To improve friction characteristics on the thrust surface between the orbiting and fixed scrolls, a circular-shaped oil groove was fabricated as shown in Figure 2. A seal-ring divides the back pressure chamber into two sub-chambers: the central portion inside the seal-ring is filled with a high pressure oil/gas mixture and a circumferential portion outside the seal-ring has a gas/oil mixture of intermediate pressure.

Inside the hub of the orbiting scroll, a small region is formed between the hub flat area and the top surface of the crank pin (Figure 2). This small chamber, called the HIC (hub-inside chamber), is filled with oil pumped from sump at the shell bottom through an oil gallery inside the crankshaft.
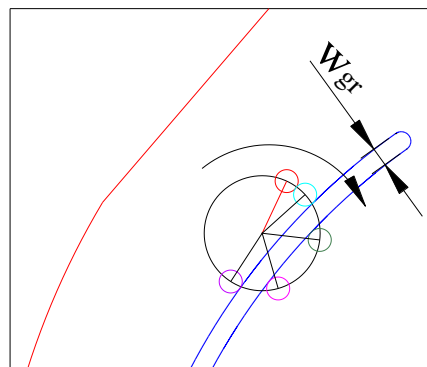

Figure 3: Intermittent connection to oil groove

Oil to the thrust surface oil groove (Figure 2) is supplied through a passage inside the base plate of the orbiting scroll as seen in Figure 3. An inlet hole to the oil passage is connected to the HIC, and an outlet hole to the oil groove. At the outlet of the oil supply passage, a connection to the oil groove is made intermittently due to the motion of the orbiting scroll itself (Figure 3).

Since the oil in the HIC is of high pressure, while the oil groove is in a somewhat lower pressure environment, oil supply is achieved due to the pressure difference between the two regions. A cylindrical throttling pin is inserted inside the main oil passage so that the supply rate can be changed by altering the pin diameter.

3. MODELING OF THE THRUST OIL GROOVE

The oil flow network around the oil groove of the thrust bearing is illustrated in Figure 4. As described above, supply from the HIC to the oil groove ($\dot{m}_H$) is intermittent as the outlet hole moves according to the orbiting motion of the orbiting scroll. The flow rate into the oil groove is determined by the pressure difference and resistance in the passage. There are two outflows from the oil groove: one is to the suction chamber ($\dot{m}_{SC}$) and the other to the outer back pressure chamber ($\dot{m}_{BPC}$). The outer back pressure chamber communicates with compression chambers (2B or 3B) through a back pressure hole. Gas/oil mixture flow ($\dot{m}_{BPH}$) takes place into or out of the outer back pressure chamber depending on the pressure difference across the hole. The hole opening is also intermittent. Across the sealing there occurs a leak flow from the inner back pressure chamber to the outer chamber.
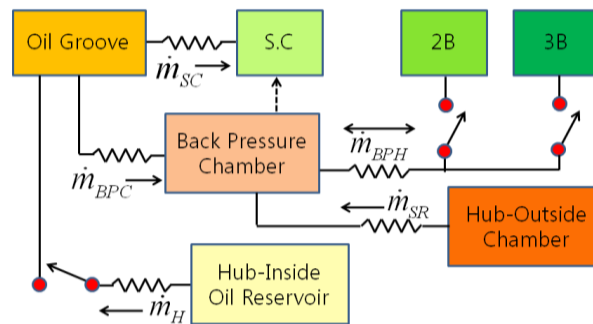

Figure 4: Oil flow network around oil groove system

If the oil fraction in the oil supply flow ($\dot{m}_H$) is x_H, gas and oil content in the oil supply flow can be expressed respectively by equations (1a) and (1b),.

$$\dot{m}_{H,\,gas} = (1 - x_H)\dot{m}_H \, , \, \dot{m}_{H,\,oil} = x_H \, \dot{m}_H \tag{1a)(1b}$$

As the oil flows from higher pressure and temperature region to lower region, some of the gas dissolved in the oil is separated from the oil due to change of solubility. The rate of gas flashing can be written by equation (2)

$$\dot{m}_{H,\,flash} = [\alpha(P_H, t_H) - \alpha(P_{gr}, t_{gr})]\dot{m}_{H,\,oil} \tag{2}$$

If the gas portion of outflow from the oil groove to the suction and back pressure chambers are denoted by $\dot{m}_{SC,\,gas}$ and $\dot{m}_{BPC,\,gas}$, respectively, the rate of change of gas mass in the oil groove is obtained by equation (3).

$$\frac{dM_{gr,\,gas}}{dt} = \dot{m}_{H,\,gas} + \dot{m}_{H,\,flash} - (\dot{m}_{BPC,\,gas} + \dot{m}_{suc,\,gas}) \tag{3}$$

Gas density and pressure in the oil groove can be obtained by equations (4a) and (4b), respectively.

$$\rho_{gr,\,gas} = \frac{M_{gr,\,gas}}{V_{gr}} \, , \, P_{gr} = P_s \, (\frac{\rho_{gr,\,gas}}{\rho_s})^n \tag{4a)(4b}$$

Pressure in the back pressure chamber Pm can also be calculated in the same way.

4. SIMULATION RESULTS OF THRUST OIL GROOVE PRESSURE

Figure 5 and Figure 6 show comparisons of the calculated groove pressure with experimental data and of the back pressure, respectively. Capital letters in the figures stand for the compressor operating conditions as indicated in Figure 7. While calculated back pressures compare well with experimental data within ±5% error range, (except under low temperature heating conditions [A and D]), scatters in the oil groove pressure are large. In general, however, the calculation fits marginally to measurements.
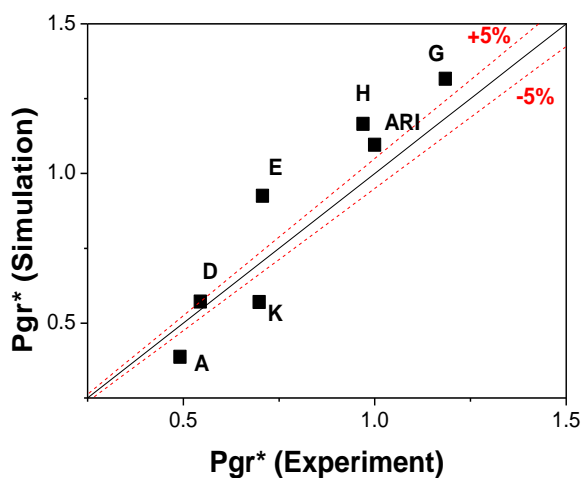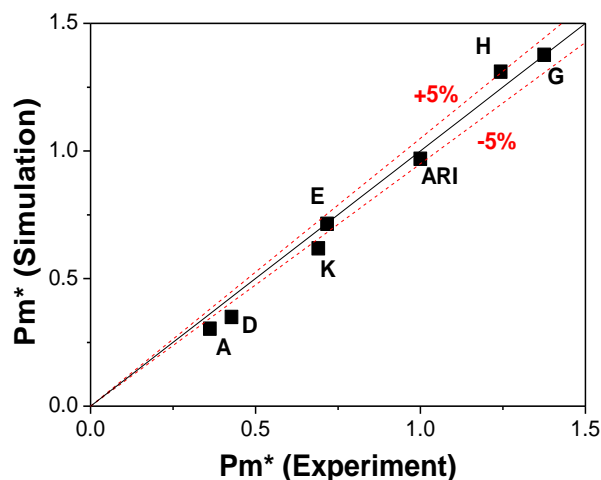

Figure 5: Groove pressure
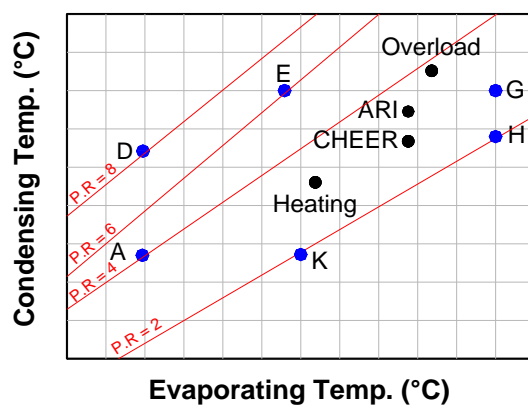

Figure 6: Back pressure

Figure 7: Compressor operating conditions

5. EFFECTS OF THRUST OIL GROOVE ON COMPRESSOR PERFORMANCE

5.1 Compressor performance with oil groove of base configuration

To help with optimum oil groove design, mathematical modeling of an oil groove system has been incorporated into a main compressor performance simulation program. Calculated compressor input and cooling capacity were compared to measurement data in Figure 8 and Figure 9, respectively. Most of the calculated values compared well to measurements with an error range of ±5%. For conditions A and D of low temperature heating, calculations were slightly larger than measurements. This difference may be ascribed to some suction gas heating from the environment, since the gas temperature in the suction line is around -20 to -25°C.

5.2 Design parameter changes for the oil groove.

To investigate the effects of the oil groove design parameters on compressor performance, compressor performance calculation has been carried out by changing the oil groove width and the insert throttling pin diameter. Groove
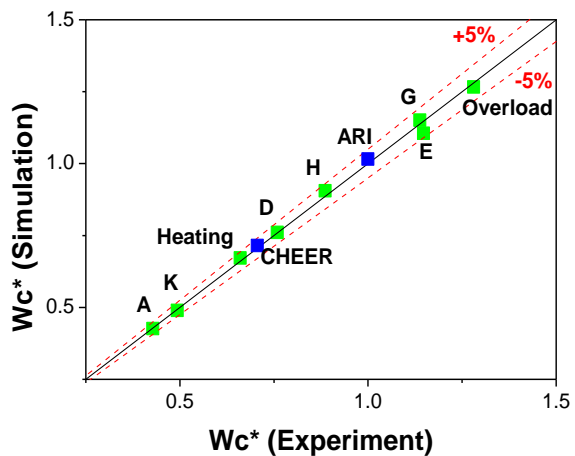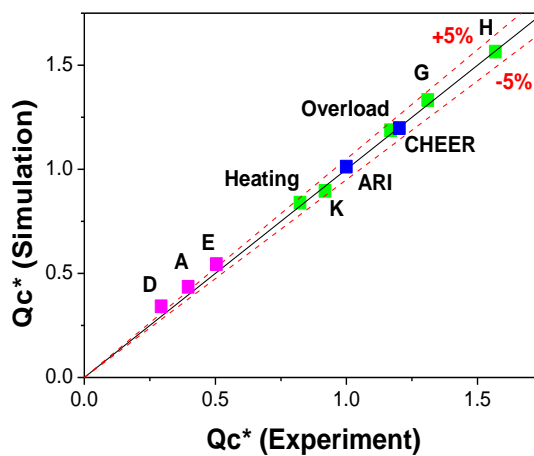

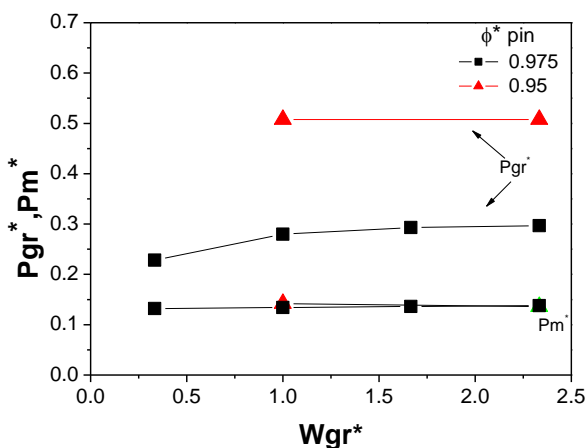**Figure 8:** Compressor input

Figure 9: Cooling capacity

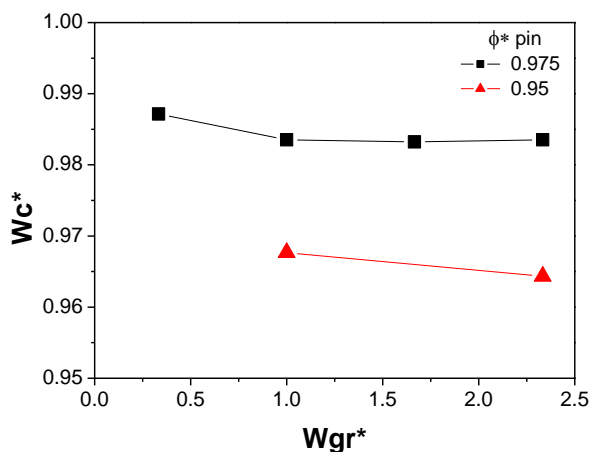**Figure 10:** Thrust oil groove pressure and back pressure
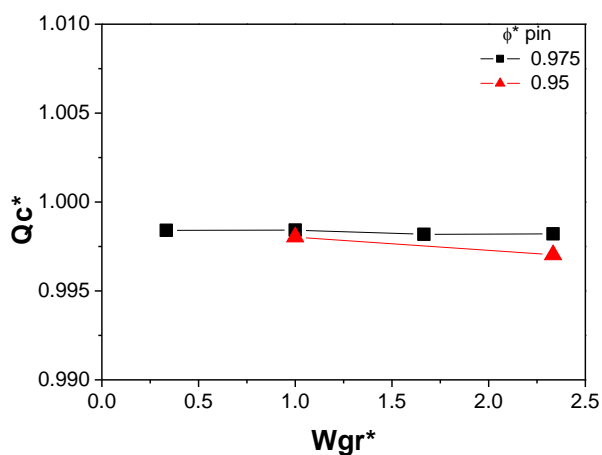

Figure 11: Compressor input

Figure 12: Cooling capacity

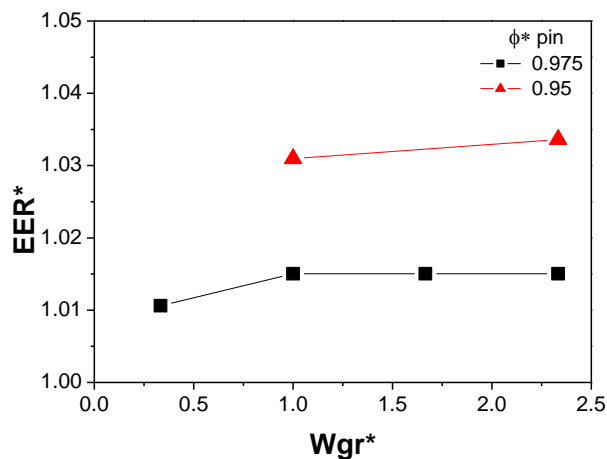**Figure 13:** EER

width was varied by w_{gr}^* =0.3~2.3, and throttling pin diameter by ϕ_{pin}^* =0.95~0.975. Groove width and pin dimensions were normalized by respective base values.

In Figure 10, groove pressure increment was gradual with increasing groove width, but significant increase in the groove pressure was made by decreasing the pin diameter or increasing oil flow passage. Back pressure was only slightly affected by groove design parameters. As a result, both pin diameter reduction and groove width increment achieved a decrease in friction force on the thrust surface. Pin diameter reduction has a much larger effect than groove width change. Compressor input decrease shown in Figure 11 is mainly due to reduced friction force obtained by pin diameter reduction and groove width increase, but cooling capacity is little affected by oil groove design parameters as seen in Figure 12. Reference values for compressor input and cooling capacity are those of a compressor with no oil groove. Further reduction in pin diameter affects orbiting scroll stability and causes increase in oil content of discharge gas. With oil groove design of w_{gr}^* =2.3 and ϕ_{pin}^* =0.95, increase in EER by 3.6% can be obtained (Figure 13).

6. CONCLUSIONS

For a high-side scroll compressor with back pressure chamber on the rear side of the orbiting scroll,
(1) A circular oil groove was employed to improve lubrication and reduce friction loss on the thrust surface between the orbiting and fixed scrolls.
(2) Modeling of the oil groove function incorporated with main compressor simulation program was used to provide optimum groove design.
(3) Compressor performance could be improved by 3.6% at ARI condition with an optimally-designed oil groove system.

NOMENCLATURE

HIC	hub-inside chamber	(-)
$\dot{m}$	mass flow	(kg/s)
α	solubility	(-)
w	width	(mm)
ϕ	diameter	(mm)

Subscript
SC	suction chamber
BPC	back pressure chamber
gr	groove
m	back pressure

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