Performance comparison of oil and oil less system with changing evaporator shape in domestic refrigerator

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Abstract. Generally, lubricating oil has the role of reducing the friction of the compressor. However, due to the presence of lubricating oil, the heat transfer performance in the evaporator is decreased and the pressure drop of the pipe in the system is increased. In this paper, performance of the oil absence was experimentally evaluated in the domestic refrigerator with different types of evaporator. The experiments were conducted in oil and oil-less systems. The on-off cyclic test and steady state energy test were performed to analyze the system performance such as power consumption, cooling capacity, coefficient of performance and operating characteristics. The grooved tube evaporator and the plain tube evaporator were also compared to investigate the influence of the lubricating oil with the evaporator tube type. The power consumption of the oil-less system was decreased compared to the oil system. This is because the pressure drop in the capillary tube was decreased and the heat transfer performance from the compressor discharge to the condenser outlet is improved. On the evaporator side, the grooved tube evaporator was improved compared to the plain tube evaporator in all systems. The inner fin on the grooved evaporator caused a larger heat transfer area and heat transfer coefficient of the refrigerant side. The oil that exists between the inner fins of the grooved evaporator influenced decreasing the heat transfer area.

1. Introduction
Generally, compressors account for more than 80% of power consumption in household refrigerators. For this reason, various research and development have been carried out to improve the efficiency of the compressor. As a result, the linear compressor had been developed. This compressor has greatly improved mechanical efficiency compared to the conventional reciprocation compressor. Meanwhile, a small amount of lubricating oil is filled in the compressor for lubrication and sealing of moving parts. When the compressor is driven, the oil is discharged together with the refrigerant and flows through the system. The presence of oil causes increase of pressure drop and deterioration of heat transfer performance in the heat exchangers. And these effects are different depending on the internal shape of the heat exchanger.
Hughes et al. (1984) performed some experiments with R12 and oil and reported that the oil has a significant effect on the refrigerant flow and pressure drop [1]. Han et al. (2017) experimentally studied on flow boiling heat transfer characteristics of R161/oil mixture in the micro-fin tube [2]. The results showed that at low vapor quality region, the local heat transfer coefficient increased slightly with increase of the oil concentration and at the high vapor quality region, it decreased significantly with increase of the oil concentration. Keqiao et al. (2018) studied on heat transfer and pressure drop characteristics of R290/oil mixture in the horizontal smooth tube [3]. They reported that pressure drop of the boiling flow with increase of the mass flux and decrease of the inner diameter. Momenifal et al. (2015) analyzed the effect of oil concentration ratio and mass velocity on boiling heat transfer coefficient and pressure drop [4]. The results showed that the boiling heat transfer coefficient of R-600a/oil mixture tend to increase at low vapor quality and decrease at the middle and high vapor qualities rather than pure R-600a.

In many previous studies, researchers did not discuss about the changes in system performance characteristics but were concerned with characteristics in the evaporator or tube. Ultimately, it is necessary to understand the effect of oil in the refrigerator performance and characteristics. The objective of this study is to investigate the performance characteristics and difference of the domestic refrigerator system experimentally with and without oil, by changing the inner fin shape of evaporator.

2. Experimental set up

Figure 1 shows the schematic diagram of the experimental apparatus. This apparatus consist of refrigerator case, refrigeration cycle, oil-separator and some data acquisition devices. The refrigerant oil contained in the refrigerant gas discharged from the compressor is separated from the refrigerant through the oil separator, and the oil is circulated to the compressor suction port. In this cycle, the environmental temperature and humidity were maintained 25±2℃ and 50±5%, respectively. Two-evaporator refrigerator using R600a (iso-butane) is adopted with internal volume of 413 L for fresh-food compartment and 246 L for freezer compartment. This refrigeration system is controlled in the following order: R operation (fresh-food) - F operation (freezer) - Pump down - Off. When the temperature of the fresh-food compartment is satisfied, the flow path to the R-evaporator direction is locked and the F-evaporator direction is opened. Then, if the temperature of the freezer compartment is satisfied, the pump down operation is started. And the system is turned off until the specified temperature is met. The refrigerator case was processed some holes to inserting piping, thermocouples, and pressure transducers. The refrigerator cycle was placed on a self-fabricated base for easy replacement of each component. The refrigeration cycle consists of a compressor, a condenser, a capillary tube, and an evaporator. At the compressor outlet, a coriolis flow meter (±0.5% uncertainty band) is installed to measure refrigerant flow.

![Figure 1. Schematic diagram of experimental apparatus](image-url)
Table 1. Specification of each component

| Specification                  | Compressor | Inverter linear compressor (165cc) |
|-------------------------------|------------|-----------------------------------|
| Condenser                     | Spiral 8(R)| Width (mm) 140, Depth (mm) 170, Height (mm) 160 |
| Evaporator                    | F-room     | Width (mm) 707, Depth (mm) 60, Height (mm) 180 |
|                               | 3(R) 6(C)  | Width (mm) 736, Depth (mm) 90, Height (mm) 120 |
|                               | R-room     | Width (mm) 736, Depth (mm) 90, Height (mm) 120 |
|                               | 3(R) 4(C)  | Width (mm) 736, Depth (mm) 90, Height (mm) 120 |
| Capillary tube                | Freezer    | Capillary tube 0.75mm (D) 1800mm (L) |
|                               | Refrigerator| Capillary tube 0.85mm (D) 1800mm (L) |

The specifications of each component are shown in table 1. The T-type thermocouples (±2.0% uncertainty band) and pressure transducers (±1.0% uncertainty band) installed at each component inlet and outlet. The ambient temperature and compartment temperature were measured by using T-type thermocouples. The data acquisition system consists of data logger, power meter and pressure transducer. These devices can measure system’s performance and characteristics like energy consumption, mass flow rate, pressure and temperature.

Experimental procedure consists of refrigerant charge optimization, on-off cyclic energy test, reverse heat leakage test and steady state energy test. The refrigerant charge optimization is to determine the optimal refrigerant charge through the ISO energy tests in 5g increment. The refrigerant charge amount with the lowest monthly energy consumption was determined as the optimum charge amount. It refers to the section where the suction temperature is not excessively reduced when the compressor is continuously operated with maximum capacity. This tests were performed by driving the compressor at ambient temperature of 25 ℃ and relative humidity of 50%. The Dual operation was operated so that the refrigerator compartment temperature was maintained at 5 ± 1.5 ℃, and the F operation was operated during the remaining compressor driving time.

After determining the optimum refrigerant charge amount for each system, experiments were conducted to compare the performance and characteristics of the four systems with varying compressor capacity. Experiments were carried out under the condition that the refrigerator and freezer compartments were maintained at a temperature of 5 ± 1.5 ℃, -18 ± 2 ℃ respectively at ambient temperature of 25 ℃ and relative humidity of 50%. The power consumption, on-time ratio, temperature, pressure, and mass flow rate of each system were measured.

The steady state energy test was performed to obtain a cooling capacity. The cabinet temperature was kept constant at -18 ± 1 ℃ using PID-driven heater. The cooling capacity is obtained by the thermal load and the electrical power and to calculate the thermal load, the reverse heat leakage test was conducted.

3. Results and Discussion

3.1. Refrigerant charge optimization

All experiments were conducted after refrigerant optimization by varying the refrigerant charge amount. The refrigerant optimization test was carried out by increasing the amount of refrigerant by 5g. Figure 2 shows energy consumption according to refrigerant charge amount. The optimum refrigerant amounts for the grooved and plain tubes in the oil-less system are 80g and 75g, respectively. And the optimum refrigerant amounts in the oil system are 85g and 70g. The difference in the amount of refrigerant when changing the inner shape of the evaporator tube is due to the increase in volume for
internal threading and the difference in cooling capacity. And the optimum amount of refrigerant differs between the oil and the oil-less system. That is because the removal of the oil which degrades heat transfer performance, sufficient cooling capacity was obtained with smaller amount of refrigerant. This is discussed in chapter 3.3.

Figure 2. Refrigerant charge amount vs. energy performance

Figure 3. Effect of the compressor capacity to energy consumption for each system
3.2. On-off cyclic test

On-off cyclic test was performed with changing the compressor capacity from 50 to 100% in ambient temperature of 25°C, relative humidity of 50% condition. Figure 3 shows the effect of the compressor capacity to the on-time ratio and the energy consumption for each system. The on-time ratio means the ratio between the compressor on time $t_{on}$, and the cycle total time, $t_{on} + t_{off}$. The on-time ratio can be calculated as follows:

$$\tau = \frac{t_{on}}{t_{on} + t_{off}}$$

(1)

The on-time ratio was increased with decrease of the compressor capacity, while the monthly energy consumption was decreased.

Figure 4 shows the changes of the compressor input power and the running time for some of the representative compressor capacity conditions when grooved tubes are applied in oil and oil-less systems. That is why the power is intermittently high at the beginning of the cycle so that the R operation (fresh-food) proceeded at this time. The diameter of the capillary to the fresh-food compartment is larger than freezer compartment. Therefore, the evaporation temperature increases due to the low pressure drop in the R operation. As a result, the specific volume of suction gas is increased, so that the mass flow rate and the compressor’s input power are temporarily increased. After the R operation, the compressor’s input power and mass flow rate remain substantially constant until the compressor turns off during F operation.

The input power was decreased gradually with decrease of the compressor capacity. However, the compressor running time and the on-time ratio were increased with decrease of the compressor capacity. The increase of the on-time ratio in same running time means decrease of the number of cycle repetitions. Therefore, as the on-time ratio is increased, the compressor on-off cycling losses are decreased, thus the energy efficiency is increased.

3.3. Steady state energy test

To calculate the heat transfer coefficient of the cabinet wall, Reverse heat leakage test should be conducted [5]. The cabinet was heated using the PID-driven electric heater. The thermal load calculated Eq. (2) under the steady state temperature condition.

$$Q_t = UA_{wall}(T_{amb} - T_f) + W_{ef}$$

(2)
The steady state energy test was performed to obtain a cooling capacity [6]. The cabinet temperature was kept constant at -18±1 °C using the heater. The cooling capacity is obtained by the thermal load and the electrical power as Eq. (3).

\[ Q_e = Q_t + \text{W}_{\text{heater}} \]  

(3)

Figure 5 shows the compressor input power versus cooling capacity. The cooling capacity for the same output is higher in the grooved tube than in the plain tube, in the oil-less system than in the oil system. When the compressor is operated with full capacity, the cooling capacity of the grooved, oil-less system is 128.51W which is about 3.9% higher than the plain, oil-less system. And the grooved, oil system is 126.53W which is about 2% higher than the plain, oil system. This is because the influence of the inner fin shapes on the grooved tube evaporator. And this characteristic of the shape caused the increasing the heat transfer area and heat transfer coefficient of the refrigerant side.

Generally, the cooling capacity of the oil-less system is reduced compared with the oil system. Figure 6 shows the pressure drop at capillary tube with changes in compressor capacity. The pressure drop in the oil system is 360 kPa in the grooved tube and 353 kPa in the plain tube. 2.3% and 1.1% higher than the oil-less system, respectively. The pressure drop in the oil system is increased because the viscosity of the lubricating oil.

Due to this high pressure drop, the compressor input power of the oil system is higher than the oil-less system under the same condition. In other components, the oil pressure drop is as low as 8 kPa. For this reason, the pressure drop in the capillary tube is the most likely cause of the high compressor input power in oil system.
3.4. Performance & characteristics comparison

The summary of measured data for experiments is shown in figure 8. It is not reasonable to compare the performance of the two systems only through the measurement results since the measurement results of each room temperature and ambient temperature are different. This is because the thermal load and the compressor driving time vary depending on the temperature condition in the cabinet. So the temperature and energy consumption correction were performed based on the reference literature [7]. Based on measured room and ambient temperatures, actual thermal load was calculated (1). Overall heat transfer coefficients of fresh-food and freezer compartment were determined by reverse heat leakage test. ($U_A_{wall} = 1.55 \text{W/K}$)

COP can be calculated by energy consumption and the thermal load (4).

$$\text{COP} = \frac{Q \times 24h}{E} \quad (4)$$

Correction of the energy consumption was performed for the refrigerator for the fresh-food and freezer compartment temperatures at 5 and -18°C respectively (5).

$$E = \frac{Q_{corrected} \times 24h}{\text{COP}} \quad (5)$$
According to the figure 7 and figure 8, the energy consumption of the oil-less system is lower than that of the conventional system. When the compressor is operated with full capacity, the energy consumption of the oil-less system with grooved tube evaporator is 27.31 kW-h/month. And this is about 8.3% higher than that of the oil system. And oil-less system with plain tube evaporator is 28.06 kW-h/month. This is about 7.4% higher than that of the oil system. This is due to the on-time ratio of the oil-less system is about 0.65 and 0.68 when the compressor is driven at maximum capacity, respectively. A low on-time ratio implies short cycle length, which means reduced in compressor running time. Monthly energy consumption has been reduced by the lower compressor input power and the reduced compressor running time.

For the other cases, the comparison was made under the condition that the compressor was continuously operated without stopping. In the oil-less system, due to the improved heat transfer performance, the energy consumption was reduced more than the oil system. Especially, the minimum energy consumptions are 24.07 kW-h/month and 25.03 kW-h/month which are lower than that of the oil system by 6% and 7% at 50% of compressor capacity, respectively.

In the same system, when changing from a grooved tube to a plain tube, the performance improved by 1.8% in the oil system and 3.8% in the oil-less system. The performance improvement of the heat exchanger with grooved tube evaporator was higher than plain tube evaporator. In the oil system, it is considered that the advantage obtained by the micro-fin heat exchanger due to the oil film formation is reduced.
As the compressor capacity increased, the refrigeration capacity and power increased. However, COP decreased because the rate of increase in cooling capacity was smaller than the rate of increase in compressor input power. Based on these results, the heat transfer performance in the heat exchangers can be improved through the removal of oil. Furthermore, the total energy consumption of the refrigerator can be reduced. Also, the compressor capacity also has a significant effect on energy consumption.

4. Conclusions
In this study, the comparison between the oil and the oil-less system was carried out by changing evaporator type. The temperature, pressure and power consumption of each system was analyzed and following conclusions were drawn.

- In the oil-less system, the improvement of the internal shape for heat exchanger is about 2.8% higher than oil system.
- The cooling capacity is higher for the oil-less system than for the oil system.
- The pressure drop in the capillary due to the presence of oil appears to have the greatest effect on system performance.
- The compressor capacity also has a significant effect on energy consumption.
- The minimum power consumption of the Grooved tube evaporator, oil-less system is 24.07 kW·h/month which is improved by about 6% than the Grooved tube evaporator, oil system.

Experimental results show better performance in the oil-less system than in the oil system. Therefore, energy consumption can be reduced by removing the compressor lubricating oil.

Reference
[1] D. W. Hughes, J. T. McMullan, K. A. Mawhinney, & R. Morgan 1984 Influence of oil on evaporator heat transfer, Int. J. Refrigeration 7 150-158
[2] X. H. Han, Y. B. Fang, M. Wu, X. G. Qiao and M. Chen 2017 Study on flow boiling heat transfer characteristics of R161/oil mixture inside horizontal micro-fin tube Int. J. Heat and Mass Transfer 104 276-287
[3] Keqiao Li, Guogeng He, Jingkai Jiang, Yang Li, Dehua Cai 2018 Flow boiling heat transfer characteristics and pressure drop of R290/oil solution in smooth horizontal tubes Int. J. Heat and Mass Transfer 119 777-790
[4] M. R. Momenifar, M. A. Akhavan-Behabadi, M Nasr, P. Hanafizadeh 2015 Effect of lubricating oil on flow boiling characteristics of R-600a/oil inside a horizontal smooth tube Appl. Thermal Engineering 91 62-72
[5] Christian J. L. Hermes, Claudio Melo, Fernando T. Knabben 2013 Alternative test method to assess the energy performance of frost-free refrigerating appliances Appl. Thermal Engineering 50 1029-1034
[6] Ji Soo Ha, Jae Seong Sim 2011 Experimental study of heat transfer characteristics for a refrigerator by using reverse heat loss method Int. J. Heat and Mass Transfer 38 572-576
[7] Yong Jin Seo, Ji Hwan Jung 2017 Influence of Internal Shape of Tube and Oil Circulation Rate on the Heat Transfer Performance of Evaporator for Domestic Refrigerator The Society Of Air-Conditioning And Refrigerating Engineers Of Korea 148-151
[8] Matej Visek., Cesare Maria Joppolo., Luca Molinaroli., Andrea Olivani. 2014 Advanced sequential dual evaporator domestic refrigerator/freezer: System energy optimization Int. J. Refrigeration 43 71-79

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