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# Experimental Study on Effects of Lubricant Oil in a Domestic Refrigerator-Freezer

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# EXPERIMENTAL STUDY ON EFFECTS OF LUBRICANT OIL IN A DOMESTIC REFRIGERATOR-FREEZER

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## ABSTRACT

Lubricant oil is essential for lubricating moving parts and cooling the components in a refrigerant compressor. However, the oil deteriorates heat transfer performance in the heat exchangers, and increases pressure drop in a refrigeration circuit. In this study, investigation of the effects of lubricant oil circulating in heat exchangers on the performance of the domestic refrigerator-freezer was carried out by performing some experiments.

The experiments were conducted for conventional and oil-less systems with using a linear compressor in an environmental chamber to compare the cooling capacity, power consumption, and operating characteristics such as temperatures and pressures at inlets and outlets of each component, mass flow rate, and cooling time. The cooling capacity and power consumption of two systems were measured during the on-off cyclic tests.

Comparing the experimental data for the conventional and the oil-less systems, the discharge temperature of the oil-less system is higher than that of the conventional system more than 20℃. However, due to the oil removal, the heat transfer performance was improved and the condenser outlet temperature was measured lower than the conventional about 1℃. The power consumption of the oil-less system was reduced by about 4.0% compared to the system with the lubricant oil.

## 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 lubricant 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 refrigeration cycle. The presence of oil causes increase of pressure drop and deterioration of heat transfer performance in the heat exchangers.

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. Han et al. (2017) experimentally studied on flow boiling heat transfer characteristics of R161/oil mixture in the micro-fin tube. Keqiao et al. (2018) studied on heat transfer and pressure drop characteristics of R290/oil mixture in the horizontal tube. 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 on the refrigerator performance and characteristics. The objective of this study is experimentally to investigate changes in the performance and characteristics of the domestic refrigerator system with and without oil.

## 2. EXPERIMENTAL APPARATUS & PROCEDURE

### 2.1 Experimental apparatus

The experimental apparatus consists of refrigerator case, refrigeration cycle and some data measurement devices. Figure 1 shows the schematic diagram of experimental apparatus. In this study, side-by-side single evaporator refrigerator/freezer using R600a is adopted with internal volume of 413 L for refrigerator compartment and 246 L for freezer compartment. In this refrigerator cycle, the dual operation and the F operation are conducted sequentially. When the temperatures of refrigerator compartment and freezer compartment are higher than each reference temperature, the Dual operation is conducted by opening the damper between the two compartments. If the refrigerator compartment sufficiently cooled, the damper is closed, and the F operation is conducted.

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 flowmeter is installed to measure refrigerant flow.

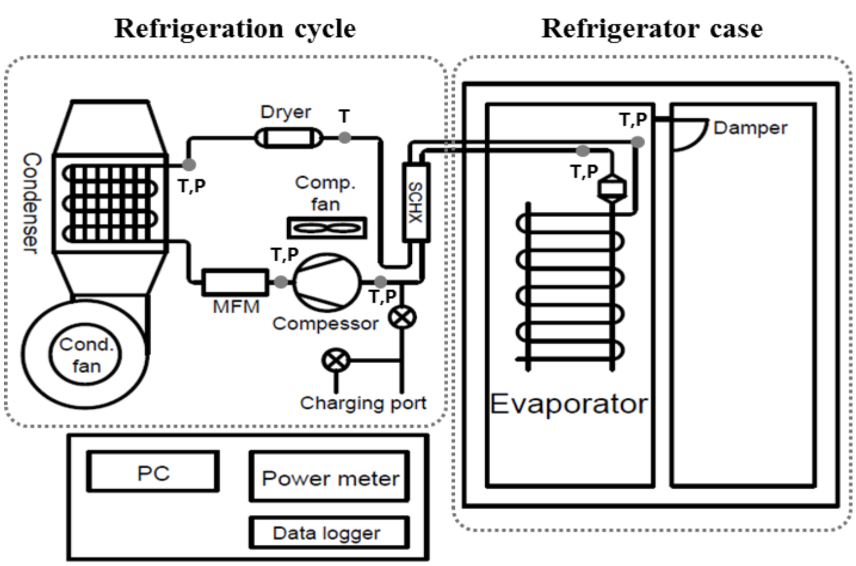


**Figure 1 :** Schematic diagram of experimental apparatus

**Table 1 :** Specification of each component

| | Specification | | | |
|---|---|---|---|---|
| Compressor | Inverter linear compressor (165cc) | | | |
| Condenser | Spiral 8(R) | Width (mm) | Depth (mm) | Height (mm) |
| | | 140 | 170 | 160 |
| Evaporator | Fin & tube 2(R)18(C) | 290 | 60 | 610 |
| Capillary tube | D : 0.75mm L : 1800mm | | | |

The brief specification of each component shown in table 1. The inlets and outlets of all components were instrumented with T type probe thermocouples and pressure transducers as shown in figure 1. Ambient temperature and the room temperature were measured by using T type thermocouples. The data measurement devices include power meter, data logger, flow and pressure transmitter. These devices measure compressor power, energy consumption, mass flow rate, temperature and pressure of each component in real time.

### 2.2 Experimental procedure

Experimental procedure consists of refrigerant charge optimization and on-off cyclic energy test. Refrigerant charge optimization was proceeded in two steps. The first step is to find out specified charging zone in 5g increment. It refers to the section where the superheat of the evaporator outlet is within ±1℃ and the suction temperature is not excessively reduced when the compressor is continuously operated with maximum capacity. This tests were performed by continuously driving the compressor at ambient temperature of 32 ℃ 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. During the tests, the evaporator inlet, outlet temperature and suction temperature were measured to find out the specified charging zone. The second step is to determine the optimal refrigerant charge through the ISO energy tests in 2g increment. The refrigerant charge amount with the lowest monthly energy consumption was determined as the optimum charge amount.

After determining the optimum refrigerant charge amount for each system, experiments were conducted to compare the performance and characteristics of the two 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 32℃ and relative humidity of 50%. The power consumption, on-time ratio, temperature, pressure, and mass flow rate of each system were measured.

## 3. RESULTS AND DISCUSSION

### 3.1 Refrigerant charge optimization

For conventional system, the specified charging zone was found to be from 75g to 85g. Figure 2 and 3 shows the refrigerant charge optimization test results for conventional system. As more refrigerant is added to the system, more refrigerant tends to be accumulated in heat exchangers. Therefore, superheat degree is decreased with increase of refrigerant charge amount. However, when the refrigerant charge amount is excessively increased, the suction temperature was excessively decreased about 15℃ and the evaporator outlet temperature was increased again. This phenomenon occurred when 95g of refrigerant is charged in both conventional and oil-less systems.

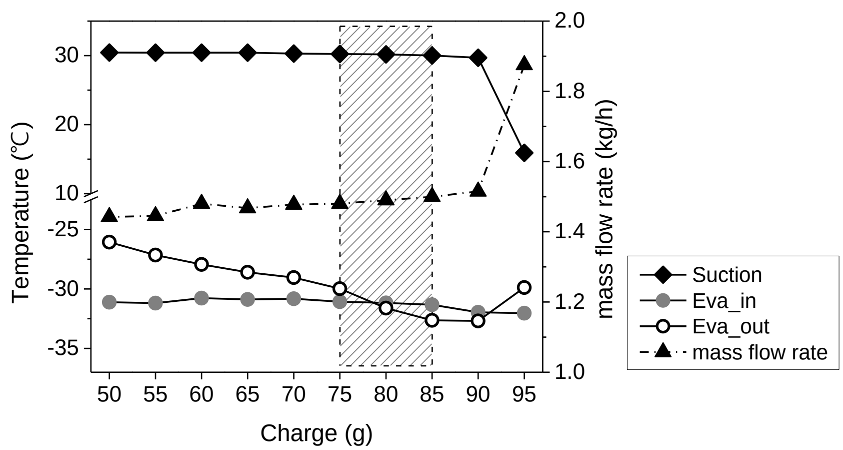


**Figure 2 :** Refrigerant charge amount vs. temperature & mass flow rate (conventional system)

In the specified charging zone as the amount of refrigerant increases, the evaporator superheat decreases. When 81g of refrigerant charged, the evaporator superheat degree was 0.49 ℃ and the monthly energy consumption was the lowest at 41.27 kWh/month. When 83 g of refrigerant was charged, temperature inversion occurred. Also, the energy consumption was increase again about 1 kWh/month. This is because of increased number of cycle repetition result from shorten cycle operation time. Therefore, the optimal charge amount for conventional system was determined to be 81g.

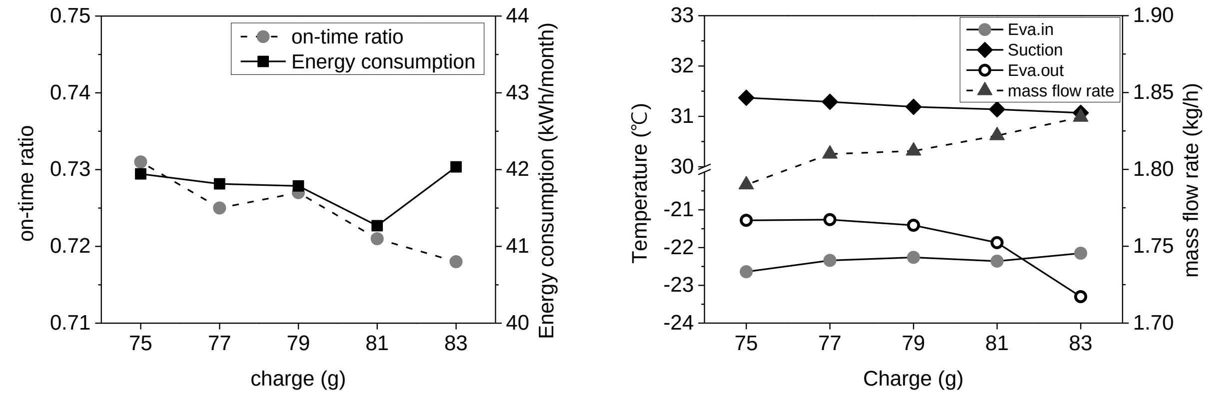


**Figure 3 :** Refrigerant charge amount vs. energy performance & characteristics (Conventional system)

By the way, the specified charging zone of oil-less system was from 70g to 80g. Because of the removal of the oil which degrades heat transfer performance, sufficient cooling capacity was obtained with smaller amount of refrigerant. However, minimum monthly energy consumption is about 5 kWh higher than the conventional system. This is discussed in chapter 3.3. The results of refrigerant optimization test for oil-less system is shown in figure 4 and 5.

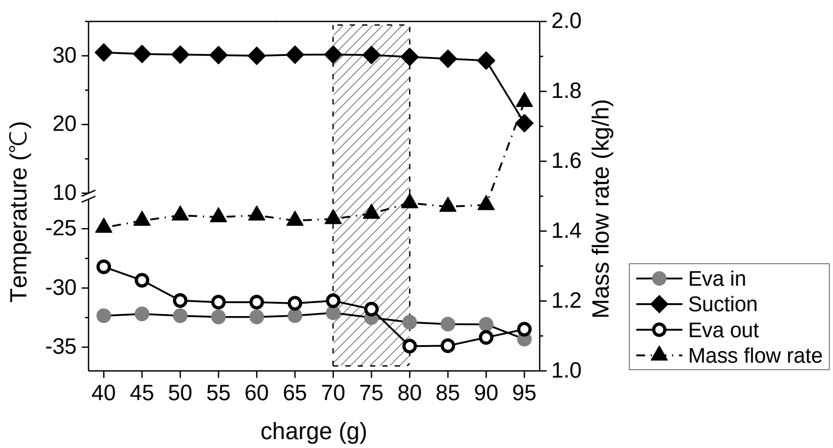


**Figure 4 :** Refrigerant charge amount vs. temperature & mass flow rate (oil-less system)

Like the results of the conventional system, as the refrigerant charge amount is increased, the mass flow rate was increased, and the evaporator superheat degree and the suction temperature were decreased. In oil-less system, the difference in superheat is not clear compared to conventional system. When the oil-less system charged with 76g of refrigerant, monthly energy consumption was the lowest at 46.51 kWh/month. However, when the additional refrigerant was charged, the energy consumption and on-time ratio were increased. As a result, the optimum charge of the oil-less system was determined to be 76g, 5g less than that of the conventional system.

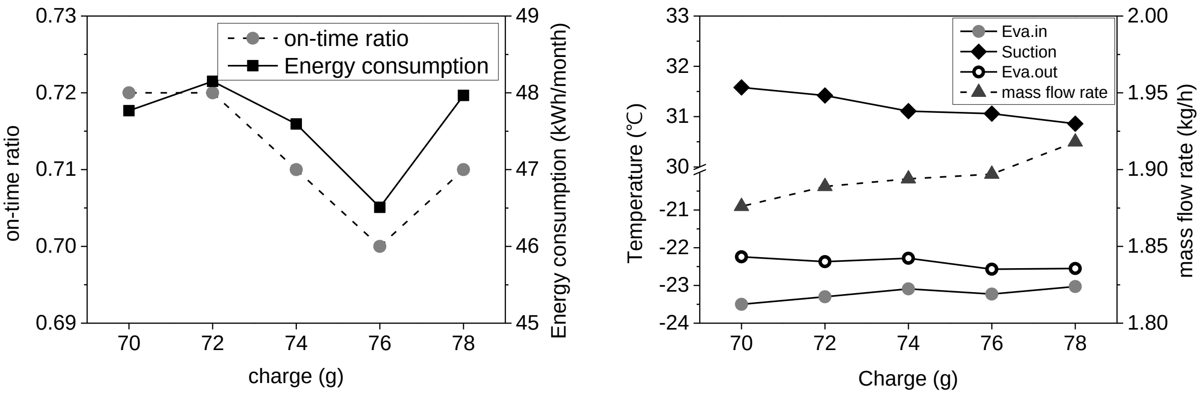


**Figure 5 :** Refrigerant charge amount vs. energy performance & characteristics (oil-less system)

## 3.2 Effect of changing the compressor capacity

For both system, cyclic on-off tests were performed with changing the compressor capacity from 50 to 100% in ambient temperature of 32℃, relative humidity of 50% condition. Figure 6 shows the effect of the compressor capacity to the on-time ratio and the energy consumption for both system. The on-time ratio was increased with decrease of the compressor capacity, while the monthly energy consumption was decreased.

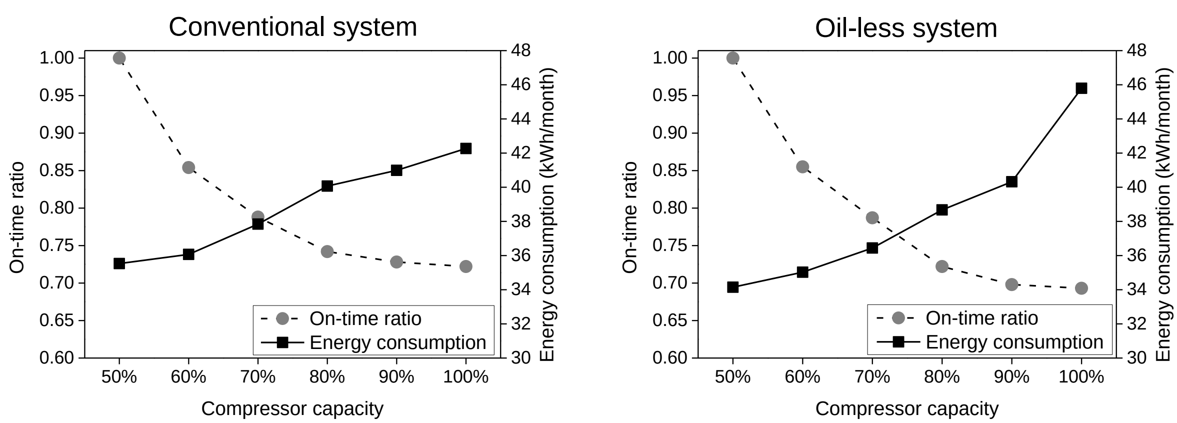


**Figure 6 :** Effect of the compressor capacity to energy consumption for both system

Figure 7 shows the changes of the input power and the compressor running time for some representative compressor capacity conditions. The reason why the output is intermittently high at the beginning of the cycle is that the Dual operation is proceeded at this time. During the Dual operation, high temperature return air flows from the R compartment into the evaporator. As a result, the evaporation temperature is increased, and the specific volume of suction gas is increased, so that the mass flow rate and the compressor input power are temporarily increased. After the Dual operation, the compressor 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. On the other hand, the compressor running time and the on-time ratio were increased with decrease of the compressor capacity. The increase of the on-time ratio 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.

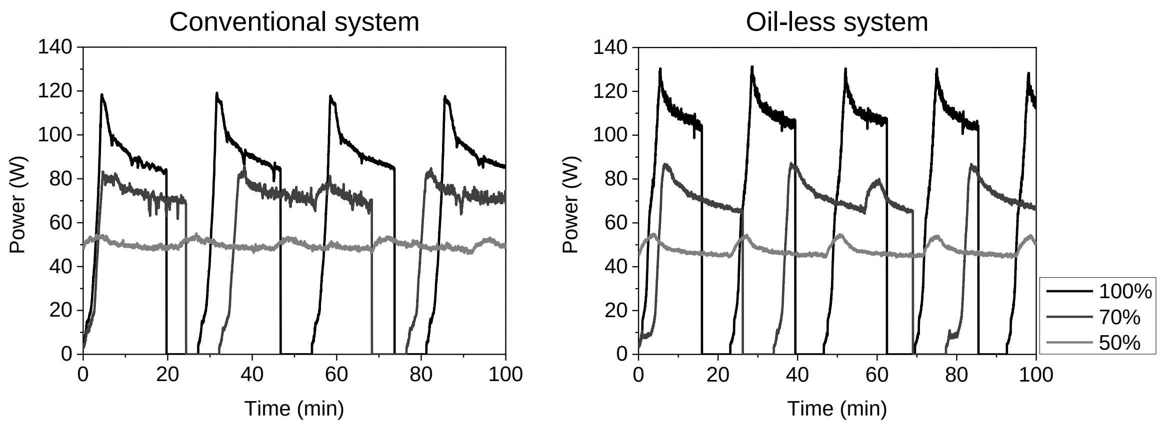


**Figure 7 :** Effect of the Compressor capacity to input power & compressor running time for both system

Figure 8 shows the variation of discharge temperature and mass flow rate with changes in compressor capacity. As mentioned above, reduction of the compressor capacity means a decrease of the stroke length of the compressor. Therefore, when the compressor capacity is reduced, a smaller amount of refrigerant is discharged and flows. At the same time, since the friction of the compressor moving parts is reduced, the discharge temperature is decreased. Also, the average discharge temperature of the oil-less system was about 20 ℃ higher than that of the conventional system. This is the result of the absence of cooling and lubrication due to oil elimination.

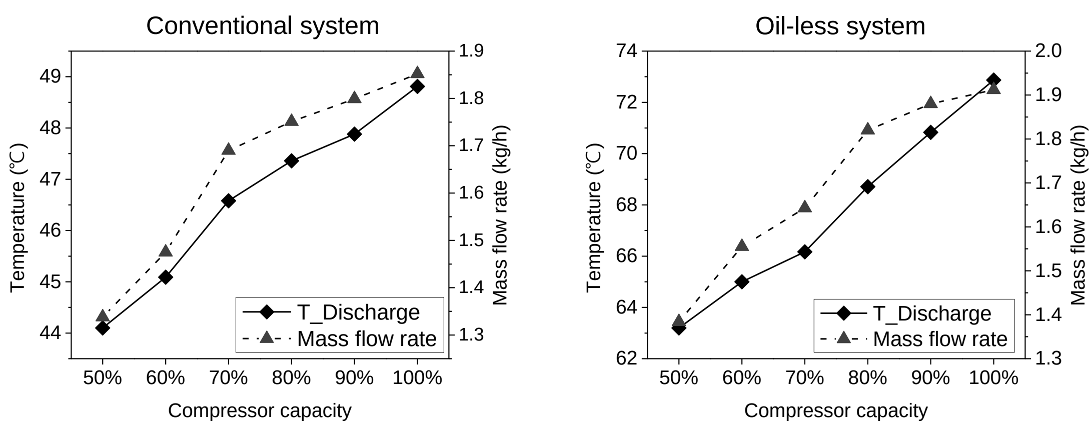


**Figure 8 :** Compressor capacity vs. discharge temperature & mass flow rate for both system

### 3.3 Performance & characteristics comparison

The summary of measured data for both system is shown in table 2 and 3. 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. The temperature and energy consumption correction were performed based on the reference literature (Matej et al., 2014)

Based on measured ambient and room temperatures, actual thermal load was calculated (1).

$$\mathrm{Q} = UA_R(T_{amb} - T_R) + UA_F(T_{amb} - T_F) \quad (1)$$

Overall heat transfer coefficients of refrigerator and freezer compartment were determined by reverse heat leakage test. ($UA_R = 3.34$ W/K and $UA_F = 2.48\ W/K$)

COP can be calculated by energy consumption and the thermal load (2).

$$\mathrm{COP} = \frac{Q \times 24h}{E} \tag{2}$$

Correction of the energy consumption was performed for the refrigerator and freezer compartment temperatures at 5 and -18℃ respectively (3).

$$\mathrm{E} = \frac{Q_{corrected} \times 24h}{COP} \tag{3}$$

Figure 9 shows the condenser outlet and suction temperatures of two systems with changes in compressor capacity. Comparing the temperature measurement data at all experimental conditions, the condenser outlet temperature of the oil-less system was measured to be lower than the minimum value of the conventional system. This is because the heat transfer performance in the heat exchanger is improved due to the oil being removed. In addition, the suction temperature for the oil-less system was also reduced compared to the conventional system. This is because heat transfer occurred in the SCHX (suction line-capillary tube heat exchanger) with the colder refrigerant from the condenser outlet.

**Table 2 :** Summary of on-off cyclic test results of conventional system

| | Temperature [℃] | | | | | | | | Mass flow rate [kg/h] | On-time ratio | Energy consumption [kWh/month] |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Case | Eva. in | Eva. out | Suc. | Dis. | Cond. out | R room | F room | Ambient | | | |
| TDC | -22.3 | -21.49 | 31.36 | 48.81 | 36.7 | 4.61 | -17.28 | 31.23 | 1.852 | 0.722 | 42.27 |
| 90% | -21.98 | -21.2 | 31.41 | 47.88 | 36.69 | 4.78 | -17.1 | 31.18 | 1.799 | 0.728 | 40.99 |
| 80% | -21.78 | -20.86 | 31.41 | 47.36 | 36.7 | 5.01 | -16.88 | 31.17 | 1.751 | 0.742 | 40.94 |
| 70% | -22.13 | -21.17 | 31.32 | 46.58 | 36.59 | 4.84 | -17.35 | 31.2 | 1.69 | 0.788 | 40.07 |
| 60% | -21.81 | -20.89 | 31.34 | 45.09 | 36.15 | 4.85 | -17.63 | 31.19 | 1.475 | 0.854 | 37.84 |
| 50% | -21.92 | -21.07 | 31.24 | 44.1 | 36.14 | 4.82 | -18.25 | 31.13 | 1.338 | 1 | 36.07 |

**Table 3 :** Summary of on-off cyclic test results of oil-less system

| | Temperature [℃] | | | | | | | | Mass flow rate [kg/h] | On-time ratio | Energy consumption [kWh/month] |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Case | Eva in | Eva out | Suc | Dis | Cond out | R room | F room | Ambient | | | |
| TDC | -22.99 | -22.44 | 30.56 | 72.87 | 35.97 | 4.3 | -17.47 | 31.74 | 1.912 | 0.693 | 45.8 |
| 90% | -23 | -22.45 | 30.59 | 70.83 | 35.91 | 4.26 | -17.61 | 31.62 | 1.88 | 0.698 | 40.32 |
| 80% | -22.83 | -22.13 | 30.62 | 68.71 | 35.85 | 4.42 | -17.61 | 31.61 | 1.82 | 0.722 | 38.67 |
| 70% | -22.67 | -21.9 | 30.78 | 66.17 | 35.74 | 4.49 | -17.9 | 31.54 | 1.643 | 0.787 | 36.44 |
| 60% | -22.77 | -22.05 | 30.58 | 65 | 35.5 | 4.46 | -18.24 | 31.65 | 1.555 | 0.855 | 35.02 |
| 50% | -23.23 | -23.26 | 30.55 | 63.2 | 35.23 | 4.38 | -19.38 | 31.62 | 1.384 | 1 | 34.15 |

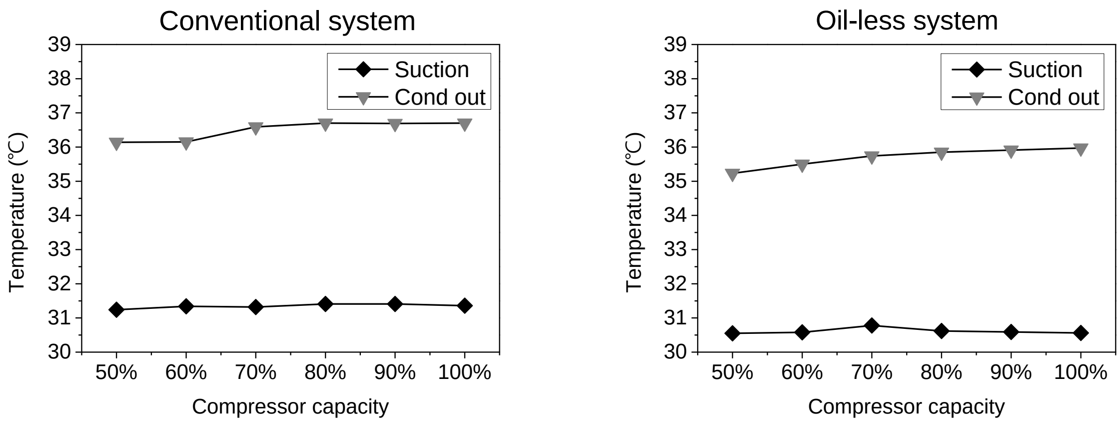


**Figure 9 :** Compressor capacity vs. Suction & Cond. outlet temperature

According to the Table 1, 2 and Figure 10, the energy consumption of the oil-less system is lower than that of the conventional system except for one case. When the compressor is operated with full capacity, the energy consumption of the oil-less system is 45.8 kWh/month which is about 8.4% higher than that of the conventional system. This is due to the on-time ratio of the oil-less system is about 0.69 when the compressor is driven at maximum capacity. A low on-time ratio implies short cycle length, which means an increase in the number of cycle repetition. Once the compressor is stopped, the internal pressure of the cycle is immediately balanced. Therefore, when the compressor is restarted, more work must be done to achieve the pressure difference.

For the other cases, the energy consumption of the oil-less system is reduced compared with the conventional system. In the oil-less system, due to improved heat transfer performance, the energy consumption was reduced more than the conventional system. Especially, the minimum energy consumption is 34.15 kWh/month which is lower than that of the conventional system by 4% at 50% of compressor capacity. 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. Therefore, if the compressor capacity can be adjusted corresponding to the thermal load, the advantage of oil removal can be maximized.

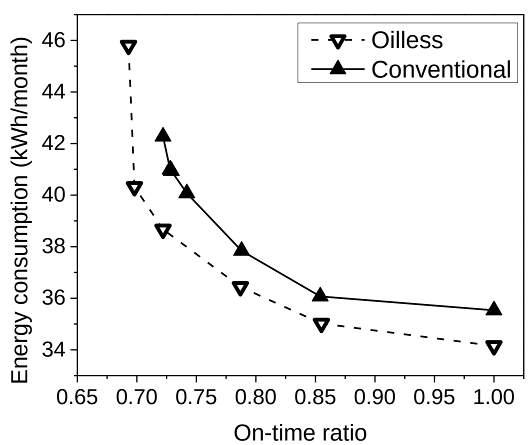


**Figure 10 :** Energy performance comparison between two systems

## 4. CONCLUSIONS

In this study, the comparison between the conventional and the oil-less system was carried out.
The temperature, pressure and power consumption of two systems were analyzed and following conclusions were drawn.

- Due to the compressor capacity variability, the energy consumption of conventional and oil-less systems was reduced by up to 15, 25% respectively.
- The minimum power consumption of the oil-less system is 34.15 kWh/month which is improved by about 4% than that of conventional system.
- In the oil-less system, the discharge temperatures were more than 20℃ higher than the conventional system, However, the condenser outlet and suction temperatures were lower 0.8 and 0.7℃, respectively.

## NOMENCLATURE

| | | |
|---|---|---|
| T | Temperature | (℃) |
| P | Pressure | (kPa) |
| Q | Thermal load | (W) |
| UA | Overall heat transfer coefficient | (W/K) |
| E | Energy consumption | (Wh) |
| SCHX | Suction line-capillary tube heat exchanger | |
| RF | Refrigerator/freezer | |
| Eva. | Evaporator | |
| Cond. | Condenser | |
| Suc. | Suction | |
| Dis. | Discharge | |
| COP | Coefficient of performance | |

**Subscript**

| | |
|---|---|
| R | Refrigerator |
| F | Freezer |
| amb | ambient |

## REFERENCES


Joel Boeng., Claudio Melo. (2014). Mapping the energy consumption of household refrigerators by varying the refrigerant charge and the expansion restriction, Int. J. Refrigeration, 41, 37-44

X. H. Han., Y. B. Fang., M. Wu., X. G. Qiao., & 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

Christian J. L. Hermes., Claudio Melo., Fernando T. Knabben. (2013) Alternative test method to assess the energy performance of frost-free refrigerating appliances, Applied Thermal Engineering, 50, 1029-1034

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

Won-Jong Lee., Joon-Young Seo., Jaedeok Ko., Ji Hwan Jeong. (2016). Non-equilibrium two-phase refrigerant flow at subcooled temperatures in an R600a refrigeration system, Int. J. Refrigeration, 70, 148-156

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

Kim, J. K., Roh, C. G., Kim, H., & Jeong, J. H. (2011). An experimental and numerical study on an inherent capacity modulated linear compressor for home refrigerators, Int. J. Refrigeration, 34, 1415-1423.

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

Won Jae Yoon., Hae Won Jung., Hyun Joon Chung., Yongchan Kim. (2011). Performance optimization of a two-circuit cycle with parallel evaporators for a domestic refrigerator-freezer, Int. J. Refrigeration, 34, 216-224

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