**NOVEL DESIGN AND PERFORMANCE OF DOMESTIC REFRIGERATORS**

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**ABSTRACT**

Domestic refrigerators are significant users of energy due to their continuous operation. The compressor is the single largest energy consumer in a refrigeration appliance and its selection is fundamental to achieve a higher system performance.

This paper focuses on the compressor in the domestic refrigerator and specifically, compressor performance at refrigerator typical operating conditions has been investigated. An analysis of the compressor isentropic efficiency for various compressor displacements (4cc to 11cc) was carried out. A theoretical model for a refrigerator was developed to investigate the influence of compressor displacement, evaporating temperature and ambient temperature on system efficiency. The results have shown that the energy consumption of refrigerators is strongly influenced by ambient temperature and larger compressors are significantly more efficient. An energy reduction of 17% was obtained by replacing the smaller 4cc compressor by an 8cc model in the system considered. The excess cooling capacity produced by large compressors can be stored as thermal cooling energy in a phase change material and used to increase the refrigerator autonomy without power supply.

**1. INTRODUCTION**

Global warming and an increasing demand for energy are among the main challenges faced by humankind today. According to the WorldWatch Institute, today’s greenhouse gas emissions need to be reduced by at least 80% by 2050 to prevent irreversible climate change (WWI, 2009).

The contribution of refrigeration to greenhouse gas emissions results both directly from refrigerant release into the atmosphere and indirectly due to the energy consumption of refrigeration systems. It is estimated that 15% of total worldwide energy consumption results from the use of refrigeration and air-conditioning systems (IIR, 2003). Domestic refrigeration in particular has a major impact on global warming with 1.2 billion domestic refrigerators in use worldwide (UN, 2006). An improvement in the energy performance of these appliances would provide a large contribution towards a more sustainable future. As a result most governments have adopted mandatory energy standards for household appliances and energy labelling programs are widespread throughout the world. These measures have encouraged the industry to develop more energy efficient refrigerators.

The energy consumption of refrigerators is affected by several factors, such as the ambient temperature, product loading and number of door openings, thermostat setting position and refrigerant migration during the compressor off-cycle. It has been demonstrated that about 55% of the total thermal load to the refrigerator comes from conduction through the cabinet walls (ASHRAE, 2006). The best solution available to reduce these heat losses is to integrate vacuum insulated panels (VIPs) in the cabinet, since they offer twice the level of insulation of polyurethane foam (PU) and result in an increased refrigerator net volume. The main reason preventing the widespread use of this technology is related to its high manufacturing and disposable costs (MTP, 2007). Door openings also introduce heat gains into the refrigerator compartment due to heat transfer with interior surfaces and air exchange with the exterior ambient. The energy consumption of a refrigerator-freezer with door openings was found to increase by 10% compared to the same product without door openings (Liu *et al.*, 2004). The influence of thermostat setting position has been experimentally proven with a 1°C reduction in the freezer temperature resulting in a 7.8% increase in energy consumption (Saidur *et al.*, 2002). Whereas, the losses in capacity due to refrigerant charge displacements e.g. due to off-cycle migration and on-cycle redistribution were estimated to be 11% (Björk and Palm, 2006). This energy loss can be prevented by fitting a liquid line solenoid valve to stop refrigerant migration from the condenser to the evaporator during the compressor off-cycle. The compressor is one of key components to optimise a refrigerator, since it consumes about 80% of the total energy consumed by the appliance. Compressor manufacturers have developed variable speed compressors that adjust the refrigeration capacity in relation to the load by controlling the motor speed resulting in energy savings up to 40% (Bansal, 2003). However this technology is still very expensive limiting its use in a market that is particularly sensitive to price.

The paper investigates the performance of single speed compressors at typical refrigerator operating conditions rather than at ASHRAE conditions. Specifically, the aim is investigate the size of compressor in relation to the system efficiency. A theoretical model for a refrigerator was also developed to investigate the influence of compressor displacement on system efficiency. If as expected larger compressors are more efficient than the challenge becomes how to use their extra cooling more efficiently. One promising option is to accumulate the excess cooling capacity as thermal cooling energy in a phase change material (PCM) and use it to increase the refrigerator autonomy without power supply. This paper describes an investigation into compressor size and the impact on system efficiency and considers the possible use of PCMs.

1. **2. COMPRESSOR PERFORMANCE**

The compressor is the single largest energy consumer in a refrigeration appliance; therefore the selection of this component is fundamental in the design of energy efficient appliances. The compressor efficiency is affected by heat transfer between the refrigerant and the cylinder wall, throttling in valve ports, gas leakage through the clearance between the piston and the bore and backflow in the valves (Rovaris and Deschamps, 2006).

Compressor manufacturer datasheets provide information on compressor performance under ASHRAE and CECOMAF conditions, i.e. condensing temperature of 55°C, ambient and suction gas temperatures of 32°C. At CECOMAF conditions the liquid temperature is 55°C and at ASHRAE conditions the liquid is subcooled to 32°C. However these conditions do not reflect how well compressors perform under more realistic ambient conditions found in domestic refrigeration. Since, the yearly average room temperature is well below 32.2°C and the suction temperature during typical operation of a refrigerator is normally lower than 32.2°C. A lower ambient temperature results in lower heat gains to the refrigerator compartment and the compressor COP is higher as the pressure ratio decreases due to a lower condensing temperature (maintaining the same evaporating temperature).

An analysis of the compressor performance of different compressor displacements was carried out to determine when there is a significant improvement in isentropic efficiency and COP under refrigerator typical operation conditions.

The RS+3 compressor and condensing unit selection tool program developed by Danfoss was used to obtain performance data for each compressor displacement for the desired usage conditions, i.e. condensing temperature of 35°C, ambient temperature of 25°C, evaporator superheat of 1K and compressor suction temperature of 15°C, assuming a suction line heat exchanger (SLHE) efficiency of 65%. ASHRAE conditions were used as the starting point for the test conditions analysis. The condenser outlet temperature was obtained by plotting the ASHRAE conditions onto a p-h diagram for R600a refrigerant (see Figure 1) together with the test conditions used in the calculations.

 

**Figure 1:** R600a p-h diagram

Equation 1 was used to determine the enthalpy h2 presented in Figure 1.

 $h\_{2}=\frac{\left(h\_{1}-h\_{4}\right)+ (COP×h\_{1})}{COP}$ (1)

The enthalpy difference in the SLHE was obtained from Equation 2.

ΔhSLHE = h1’- h5’ (2)

Where, h5’ corresponds to the enthalpy at the evaporator exit with 1K superheat.

As, h4’ = h3’ = h3 – 0.65×ΔhSLHE (3)

The corresponding condenser outlet temperature for h3’ is determined from Equation 3.

CoolPack software (version 1.46) was then used to calculate the compressor isentropic efficiency for each compressor model at various evaporating temperatures. The input data used in the program (cooling capacity, power consumption and COP) were obtained from the RS+3 program. The isentropic efficiencies for several compressor displacements are shown in Figure 2.



**Figure 2:** Isentropic efficiency for different compressor displacements at test usage conditions

As can be seen in Figure 2, two distinct groups of compressors are apparent from the graph, from 4cc to 7cc and from 8cc to 11cc. These two groups correspond to a different compressor shell size. The results have shown that in general larger compressors are more efficient, since the isentropic efficiency increased from 0.4 to 0.6 as the displacement increased from 4 to 8cc. In the second group of compressors (8cc to 11cc) the 8cc compressor seems the most efficient at fridge storage conditions, which may be the result of using a smaller cylinder (less heat transfer losses) or may be due to improved electrical characteristics of the motor.

The coefficient of performance (COP) is also higher for the second group of compressors (8cc to 11cc) and increases linearly with evaporating temperature as can be seen in Figure 3.



**Figure 3:** COP for different compressor displacements at normal usage conditions

1. **REFRIGERATOR THEORETICAL MODEL**

The refrigerator performance, evaluated in terms of energy consumption and running time of the appliance, was estimated by calculating the heat load into the refrigerator compartment and the power consumed by the fridge components, i.e. compressor; defrost heater, evaporator fans, power supplies, etc.

The system performance was then evaluated for various conditions of evaporating temperature, ambient temperature and for several compressor displacements.

* 1. **3.1 Heat load into the refrigerated compartment**

The heat load into the refrigerator of external dimensions 1.22×0.56×0.55m was calculated by considering a steady state conduction gain and then by adding an additional gain of 7.5% to accounts for periods of door openings. The refrigerant migration during the off-cycle was not considered in the transient margin, since it is possible to avoid this efficiency loss by fitting a liquid line solenoid valve.

The ambient temperature was assumed to be 25°C, which corresponds to the test temperature of the BS EN Standard. It was considered that the back wall of the refrigerator would be at 35°C due to the condenser being mounted on this wall. Additional heat from the compressor compartment was assumed to provide an extra 7.5°C to the bottom surface of the refrigerator. The compartment temperature was maintained at 3°C and as a baseline, an evaporating temperature of -10°C was considered.

The three mechanisms of heat transfer, conduction, convection and radiation were considered in the heat load to the refrigerated compartment. Therefore, a global heat transfer coefficient was assumed in the calculations. The total heat transfer rate was determined by equation 4.

Q = UAΔT (4)

The global heat transfer coefficient was calculated using equation 5.

 $U= \frac{1}{^{x}/\_{k} + R}$ (5)

The insulation material considered was PU foam, 57mm thick with a thermal conductivity of 0.022 W/mK and 0.1 m2K/W surface resistances (ASHRAE, 2009). The evaporator was positioned in the refrigerator ceiling, and the reduced insulation over the evaporator was accounted for in the heat gain calculations.

The defrost heat load over the cycle period was calculated by assuming a defrost heater duty of 200W for a defrost period of 3.5 minutes and assuming that one defrost was required every 40 hours.

The calculated heat gains into the refrigerator are summarized in Table 1.

**Table 1:** Refrigerator heat gains



As can be seen in Table 1 the side and the back surfaces are the main contributors to the total heat gain.

The evaporator fans were selected to correspond to the compressor cooling capacity (previously determined with the RS+3 program). An evaporating temperature of -10°C and a minimum ΔT of 2°C between the refrigerant and the compartment air temperature were assumed. The fans running time was calculated by dividing total heat gain to the compartment (shown in Table 1) by the compressor cooling capacity. The heat gain from the fans was determined by multiplying the fans input power by their running time.

The total heat gains to the refrigerated compartment are presented in Table 2 and include the heat gain from the evaporator fans and a transient margin.

**Table 2:** Total heat gains including fans and transients



The running time of the unit was estimated from Equation 6.

 $Run time for appliance= \frac{Total heat gain}{Cooling capacity}$ (6)

The total power input to the compressor was obtained from Equation 7.

 $Total power input to compressor= \frac{Cooling capacity}{COP} ×Run time$ (7)

Table 3 illustrates the energy consumption of each component; the compressor employed was the 8cc, identified as the most efficient in the compressor performance section.

**Table 3:** Components energy consumption



The overall energy consumption of the refrigerator is 237 Wh/day, which is equivalent to 86.5 kWh/year.

* 1. **3.2** **Impact of evaporating temperature**

The impact of evaporating temperature on the refrigerator energy consumption for the 8cc compressor is illustrated in Table 4.

 **Table 4:** Refrigerator performance at different evaporating temperatures



The refrigerator energy consumption increased by 5% with a 5°C decrease in evaporating temperature.

**3.3 Impact of ambient temperature**

The impact of ambient temperature on the refrigerator heat load and power consumption are shown in Table 5 for the selected 8cc compressor, assuming an evaporating temperature of -10°C and a compartment temperature of 3°C.

**Table 5:** Refrigerator performance at different ambient temperatures



Table 5 induces that a 5°C increase in ambient temperature results in a 22% increase in the heat load to the refrigerated compartment and a 33% increase in the refrigerator energy consumption. In practice the relationship between condensing temperature and ambient is dynamic and the condenser and evaporator would be sized to maintain the required compartment temperature for the refrigerator climate class.

**3.4 Impact of compressor selection**

The common temperature parameters used in the calculations were an ambient value of 25°C, condensing at 35°C, evaporating at -10°C and a compartment temperature set to 3°C. Table 6 presents the system’s performance for the different compressor models listed previously.

**Table 6:** Refrigerator performance with different compressor models

1

4.0

0.215

287

35.5

A+

2.52

2

5.7

0.144

277

34.2

A+

2.63

3

7.48

0.106

277

34.2

A+

2.63

4

8.05

0.097

237

29.3

A++

3.18

5

8.76

0.088

241

29.8

A++

3.11

6

11.2

0.070

249

30.9

A+

2.98

**COP**

**Energy**

**Index**

**Displacement**

**(cc)**

 **Class**

**Compressor**

**model**

**Run Time**

**Energy Input**

**(Wh/24h)**

As can be seen in Table 6, the 8cc compressor was optimal for the refrigerator studied under normal operating conditions and the most efficient energy class can be achieved. An energy reduction of 17% was obtained by replacing the smaller 4cc compressor by an 8cc model and the run time decreased from 21.5% to 9.7%. Since the run time of the appliance decreases as the compressor displacement increases the time required to cool the compartment is short but the compressor needs to be used quite frequently.

1. **CONCLUSION**

The model demonstrated that the energy consumption of refrigerators is strongly influenced by the room temperature and that although larger compressors consume more power they are significantly more efficient as proven by 17% the energy reduction obtained by increasing the displacement from 4cc to 8cc. In order to fully exploit the higher performance of large compressors they need to be applied in a novel way so that their excess cooling capacity is used and reasonable run times are achieved.

One way to use the larger compressor for longer is to store excess cooling capacity. This can be achieved by storing thermal cooling energy in a phase change material connected to the evaporator. This accumulated thermal cooling energy can be used to increase the autonomy of the refrigerator without power supply by reducing the start/stop operations of the compressor and to overcome intense periods of door openings and product loading. The performance improvement provided by phase change materials will be the subject of a future study and the theoretical model proposed will be validated experimentally.

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**NOMENCLATURE**

A Area (m2) R Surface resistance (m2K/W)

COP Coefficient of Performance R600a Isobutane

ΔT Temperature difference (K) SLHE Suction Line Heat Exchanger

h Enthalpy (kJ/kg) U Global heat transfer coefficient (W/m2K)

k Thermal conductivity (W/mK) VIPs Vacuum Insulation Panels

PU Polyurethane foam VSC Variable Speed Compressors

Q Heat transfer rate (W) x Wall thickness (m)

# REFERENCES

# ASHRAE Fundamentals handbook, 2009, *Heat, air and moisture control in building assemblies – Material properties.*

# ASHRAE Refrigeration handbook, 2006, *Household refrigerators and freezers*.

# Bansal, P. K., 2003, Developing new test procedures for domestic refrigerators: harmonisation issues and future R&D needs - a review, *International Journal of Refrigeration* 26(7): 735-748.

1. Björk, E., Palm, B., 2006, Refrigerant mass charge distribution in a domestic refrigerator, Part I: Transient conditions, *Applied Thermal Engineering* 26(8-9): 829-837.
2. Coolpack Software, 2001, Denmark Technical University, Department of Mechanical Engineering, Available: <http://www.et.web.mek.dtu.dk/Coolpack/UK/download.html> [Last accessed 03 November 2008].
3. Danfoss, 2009, RS+3 Compressor and Condensing unit selection tool, Available: [http://www.danfoss.com/BusinessAreas/RefrigerationAndAirConditioning/Product+Selection+Tools+Details/RSplus3.htm](http://www.danfoss.com/BusinessAreas/RefrigerationAndAirConditioning/Product%2BSelection%2BTools%2BDetails/RSplus3.htm) [Last accessed 15 December 2008].
4. International Institute of Refrigeration, 2003, *How to improve energy efficiency in refrigerating equipment*, IIF/IIR, Paris, Available: <http://www.iifiir.org/en/doc/1015.pdf> [Last accessed 28 January 2009].
5. Liu, D.-Y., Chang, W.-R., Lin, J.-Y., 2004, Performance comparison with effect of door opening on variable and fixed frequency refrigerators/freezers, *Applied Thermal Engineering* 24(14-15): 2281-2292.
6. Market Transformation Programme, 2007, *IBNC16: Innovation Briefing Note: Vacuum insulated panels for refrigerated appliances*. Available: http://www.mtprog.com/spm/download/document/id/667*.* [Last accessed 20 June 2008].
7. Rovaris, J. B., Deschamps, C. J., 2006, Large eddy simulation applied to reciprocating compressors, *Journal of the Brazilian Society of Mechanical Sciences and Engineering* 28(2).
8. Saidur, R., Masjuki, H. H., Choudhury, I. A., 2002, Role of ambient temperature, door opening, thermostat setting position and their combined effect on refrigerator-freezer energy consumption, *Energy Conversion and Management* 43(6): 845-854.
9. The WorldWatch Institute, 2009, *State of the World 2009: Into a Warming World*. Available: <http://www.worldwatch.org/files/pdf/SOW09_chap2.pdf> [Last accessed 2 February 2009].

# United Nations Environment Programme, Montreal protocol on substances that deplete the ozone layer, *2006 Report of the refrigeration, air conditioning and heat pumps technical options committee*. Available: [http://ozone.unep.org/teap/reports/rtoc/rtoc\_assessment\_report06.pdf](http://ozone.unep.org/teap/Reports/RTOC/rtoc_assessment_report06.pdf) [Last accessed 27 January 2009].