Comparison Of Energy Consumption Of Ventilated And Natural Convection Evaporators Of Refrigerators And Freezers

A. Zoughaib
Ecole des Mines de Paris

D. Clodic
Ecole des Mines de Paris

R19-1

COMPARISON OF ENERGY CONSUMPTION OF VENTILATED AND NATURAL CONVECTION EVAPORATORS OF REFRIGERATORS AND FREEZERS

Assaad Zoughaib, Denis Clodic
Ecole des Mines de Paris, Center for Energy Studies
60, boulevard Saint-Michel – F – 75272 Paris Cedex 06
Phone: +33 1 40 51 92 49 – Fax: +33 1 46 34 24 91
zgheib@cenerg.ensmp.fr , clodic@cenerg.ensmp.fr

ABSTRACT

Forced convection on heat exchangers yields to higher heat exchange coefficient and so permits to limit the temperature difference between air and the evaporator. Higher energy performances of the refrigerating cycle is affordable compared to natural convection evaporators. In Europe, many refrigerators and freezers integrate natural convection heat exchangers. Making a review of the actual energy consumption of European appliances, it is obvious that a number of natural convection refrigerators and freezers show higher energy performances compared to ventilated and no frost appliances. The actual inefficiency of usual small electrical motors of fans spoils the energy gains possibly reached by forced convection.

Based on experimental data and dynamic simulation, the paper presents comparisons between heat exchange coefficients, evaporating temperature levels, and overall energy consumption of both ventilated and natural convection refrigerators.

Conclusions are drawn on the required energy efficiency of electrical motors in order to reach better energy performances for ventilated refrigerators and freezers.

NOMENCLATURE

C: daily energy consumption (Wh/day)
Cp: heat capacity (J/kg.K)
COP: coefficient of performance
D: tube diameter (m)
Fp : fin pitch (m)
Gc: mass velocity (kg/m^2.s)
HL: heat losses (W)
h: heat transfer coefficient (W/m^2.K)
LTD: logarithmic temperature difference (K)
P: power consumption (W)
P_t: transversal tube pitch (m)
Q: cooling capacity (W)
S: surface area (m^2)
s: spacing between adjacent fins (m)
T: temperature (K)
UA: overall heat exchange coefficient (W/K)

Greek letters

σ : Stephan-Boltzman constant
ε : total emissivity of the freezer wall

Subscripts

$_{air\_i}$: air temperature at the inlet of the evaporator
$_{air\_o}$: air temperature at the outlet of the evaporator
$_c$: Fin collar outside diameter
$_{cond}$: condensing
$_{def}$: average daily defrosting
$_e$: external
$_{evap}$: evaporating
$_{fan}$: fan
$_i$: internal
$_{rad}$: equivalent radiative

Dimensionless numbers

J: Colburn factor
Nu_L: Nusselt number based on the height L
Pr: Prandtl number
Ra_L: Rayleigh number based on the height L
Re_{Dc}: Reynolds number based on the tube collar diameter

INTRODUCTION

The performance of an ideal refrigerating cycle is calculated by the Carnot Coefficient Of Performance (COP), which is the ratio of the cooling capacity to the mechanical power. The ideal COP is expressed as a function of the cycle operating temperatures as shown in the equation 1.

$$COP_{Carnot} = \frac{T_{evap}}{T_{cond} - T_{evap}} \qquad (1)$$

The real refrigerating cycle presents irreversibilities that lead to lower performance compared to the ideal one, however, the performance variation of both ideal and real cycles is similar.

Equation (1) shows clearly that the performance is non linearly dependant of the evaporating temperature and that the COP decreases very rapidly when the evaporating temperature decreases.

For refrigerators, the evaporating temperature is fixed by the air side heat exchange coefficient, which is very low compared to the refrigerant side one. The no-frost appliances use ventilated fin and tube heat exchanger while the natural convection ones use a static heat exchanger with an increased heat exchange area. Experimental measurements show an average of 5K difference in evaporating temperature between those 2 technologies [ZOU00]. This temperature difference implies that higher energy performances can be affordable with the no-frost technology.

However, no-frost appliances require a fan that blows air over the evaporator and a defrosting system to melt the ice that clogs up the evaporator. These 2 accessories yield to extra energy consumption which can spoil the energy gain, and in many cases the energy consumption of a no-frost appliance is higher than an equivalent static one.

In addition, because of the higher evaporating temperature, the compressor to be used in a no-frost appliance is smaller than the one used in the equivalent static appliance. For actual hermetic compressors, efficiency decreases when the swept volume decreases, which leads to additional energy consumption.

1. DESCRIPTION OF APPLIANCES

For comparison the chosen appliances present the same geometry and insulation thickness. Figure 1 shows the geometry description in the ENEREF® software [CLO01] and the calculated net volume.

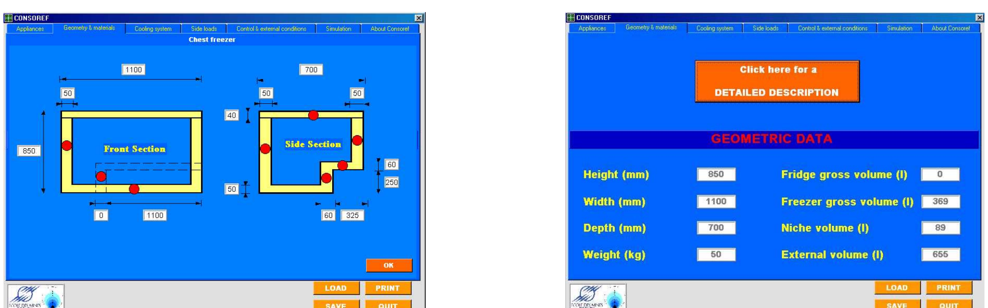

Figure 1 – The geometry description in the ENEREF® software.

The no-frost appliance has a fin and tube heat exchanger and the natural convection appliance uses the vertical walls as exchange area. The evaporator tube length is assumed to be equal for both appliances.

The heat losses, calculated by ENEREF®, are of 60 W. The running time ratio, defined by the ratio of the compressor running time to the overall cycle time, is considered to be 40% for a 25°C test-room temperature. Hence the needed average cooling capacity is 150 W.

2. EVAPORATING TEMPERATURE CALCULATION

The evaporating temperature is calculated by analyzing both evaporators using the logarithmic temperature difference method (LTD). The superheating section of the evaporator is neglected for both evaporators and hence the LTD is defined by equation (2).

$$LTD = \frac{T_{air\_out} - T_{air\_in}}{Ln(\frac{T_{air\_out} - T_{evap}}{T_{air\_in} - T_{evap}})} \tag{2}$$

The overall heat exchange coefficient (Equation (3)) is calculated using the internal and the external convective heat transfer coefficients. The conduction through the tube is neglected.

$$UA = \frac{1}{\frac{1}{h_i \times A_i} + \frac{1}{(h_e + h_{rad}) \times A_e}} \tag{3}$$

The cooling capacity Q is calculated by Equation (4)

$$Q = UA \times LTD \tag{4}$$

The correlation of Gungor-Winterton [GUN97] is used for the calculation of the average internal convective heat transfer coefficient.

$$\frac{h_{i,x}}{h_l} = 1 + 3000 \times Bo^{0.86} + 1.12 \times \left(\frac{x}{1-x}\right)^{0.75} \times \left(\frac{\rho_l}{\rho_v}\right)^{0.41} \tag{5}$$

The liquid heat transfer coefficient is calculated by Dittus-Boelter correlation. The internal heat transfer surface area is assumed to be equal for both appliances.

2.1 External Heat Transfer Coefficient for the No-frost Appliance

The no-frost appliance evaporator is a fin and tube heat exchanger. Its characteristics are presented in Table 1.

Table 1 – Fin and tube evaporator characteristics.

Number of fins /m of tube	166
Tube length (m)	0.2
Number of tube per row	2
Number of rows	5
Transversal pitch (m)	0.03
Longitudinal pitch (m)	0.027

Gray and Webb correlation [GRA86] for plate fins and tube is used to calculate the external heat transfer coefficient. Gray and Webb calculate the Colburn factor using Equation (6).

$$J = 0.14 \times Re_{Dc}^{-0.328} \times \left(\frac{P_t}{P_l}\right)^{-0.502} \times \left(\frac{s}{D_c}\right)^{0.0312} \tag{6}$$

And the external heat transfer coefficient is calculated using Equation (7)

$$h_e = \frac{J \times G_c \times Cp_{air}}{\Pr^{2/3}} \tag{7}$$

This correlation is valid for:

$500 < Re_{Dc} < 24700$; $1.97 < P_t/D_c < 2.55$

$1.7 < P_l/D_c < 2.58$; $0.08 < F_p/D_c < 0.64$

2.2 External Heat Transfer Coefficient for the Natural Convection Appliance

In the natural convection case, the freezer walls are used as external surface area for the evaporator. The geometry of the appliance permits to calculate this surface area.

The average external Nusselt number is calculated using the correlation of natural convection over a vertical wall (8).

$$Nu_L = \left\{ 0.825 + \frac{0.387 \times Ra_L^{1/6}}{\left[1 + (0.492 / \Pr)^{9/16}\right]^{8/27}} \right\}^2 \tag{8}$$

This correlation is valid for: $10^{-1} < Ra_L < 10^{12}$

For natural convection, the radiative heat transfer is an important part in the overall heat transfer. Thus, the equivalent radiative heat transfer coefficient is calculated by Equation (9).

$$h_{rad} = \varepsilon \times \sigma \times \left(T_{wall}^3 + T_{wall} \times T_{ambiant}^2 + T_{wall}^2 \times T_{ambiant} + T_{ambiant}^3 \right) \tag{9}$$

2.3 Evaporators Results

Both evaporators are analyzed by the methodology described previously. The main results for the forced convection and the natural convection evaporators are summarized in table 2.

Table 2 – Results of the forced and natural convection evaporators.

	Forced convection	Natural convection
S_{ext} (m^2)	0.6	2.5
S_{int} (m^2)	0.05	0.05
h_i ($W/m^2.K$)	1280	1460
h_e ($W/m^2.K$)	60.16	3.7
h_{rad} ($W/m^2.K$)	--	2.1
UA (W/K)	22.4	12
Cooling cap (W)	150	150
LTD (K)	6.7	11
T_{evap} (°C)	-25	-29

The calculated evaporating temperatures are in accordance with the experimental observations (4K of evaporating temperature difference between the 2 technologies). The overall external heat transfer coefficient for the forced convection is ~10 times higher than the natural convection one.

3. COMPRESSOR MODEL

The compressor model is based on the experimental data given by the manufacturer. The cooling capacity, the mass flow rate and the input power can be correlated in a polynomial function of T_{evap} and T_{cond} [DAN02].

3.1 Cooling Capacity Variation with the Cycle Temperature

The cooling capacity of the compressor varies strongly with the cycle temperatures.

Figure 2 represents the cooling capacities for different evaporating and condensing temperatures, based on a reference cooling capacity at the ASHRAE 23 testing standard conditions [ASH93] (T_{evap}=-23.3°C and T_{cond}=54.4°C).

This figure indicates that a compressor will provide higher cooling capacity when operating at a higher evaporating temperature, if the condensing temperature is constant. In order to obtain the same cooling capacity for the two compared technologies, at different evaporating temperatures, a smaller cooling capacity compressor shall be selected for the no-frost appliance.

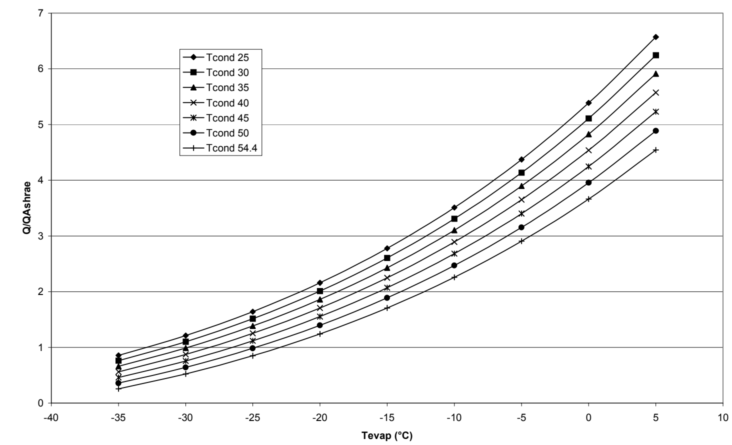

Figure 2 – Cooling capacity at different evaporating and condensing temperatures.

3.2 Variation of the Compressor COP as a Function of the Cooling Capacity

The compressor COP decreases with the cooling capacity for usual compressors, except rated speed ones. This variation is shown in the figure 3 where the COP at the testing conditions is plotted as a function of the cooling capacity at the same conditions.

Hence, the selected compressor for the No-frost appliance is less efficient and this reduces the advantage of having a higher evaporating temperature.

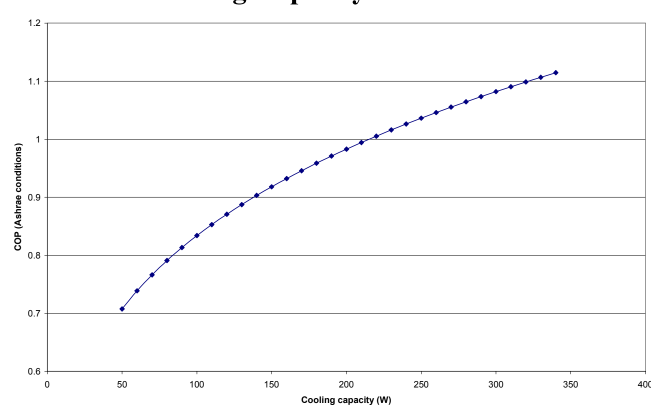

Figure 3 – COP ASHRAE variation with the cooling capacity.

3.3 Compressor Selection

Both compressors are selected for the operating conditions using the previous results. The table 4 presents the characteristics of both compressors.

The No-frost appliance compressor is 8% less efficient, however, the system COP is still 27% higher. Hence, the penalty of having a smaller compressor does not spoil the advantage of operating at higher evaporating temperature.

Table 4 – Compressors characteristics.

	No-frost	Natural convection
Tevap (°C)	-25	-29
Tcond (°C)	45	45
Q_{Ashrae} (W)	151	207
COP_{ashrae}	0.88	0.95
Q (W)	150	150
COP	0.93	0.73

4. DEFROSTING HEATER MODEL

The frost appears on the evaporator because of the cabinet air dehumidification. For the non ventilated appliances, the freezer defrosting is manual by stopping the appliance once every two months. For the ventilated appliances, a defrosting system is installed because the evaporator is accumulating all the frost in the cabinet, and this frost layer lowers the average performances. Defrosting is commonly realized by an electrical heater. Electronic control permits to perform an adaptive defrost that yields to reduce energy consumption and to improve food preservation [ASH94].

The calculation of the heater power to be installed considers the number of the door openings, the cabinet internal volume, the external temperature, and relative humidity. The standard used by the Korean refrigerator industry [BEJ94] gives an idea of the real use of the freezer. The conditions used by this standard considers:

- ambient temperature of 30°C and a RH of 75%
- the refrigerators door is opened 10s every 12min
- the frozen food compartment is opened 10s every 40min.

A complete volume renewal at each door opening is assumed. The calculations are based on a single 20min defrost each 24hrs. A defrosting efficiency is introduced, it is the ratio of the energy used to melt the frost and heat the evaporator mass and the total input energy of the heater. The defrosting efficiency depends on the heater technology and on the defrosting system control. The calculations assume three levels of defrosting efficiency: 40%, 60% and 80%.

5. FAN CONSUMPTION AND EFFICIENCY

The fan consumption is determined by the required air mass flow rate on the evaporator and the efficiency of the fan. Three technologies of fans are available:

- regular AC fans,
- improved AC fans,
- brushless DC fans.

The mass flow rate needed for the studied freezer is 150m^3/h. For this mass flow rate the consumption of the three types of fans are represented in figure 4.

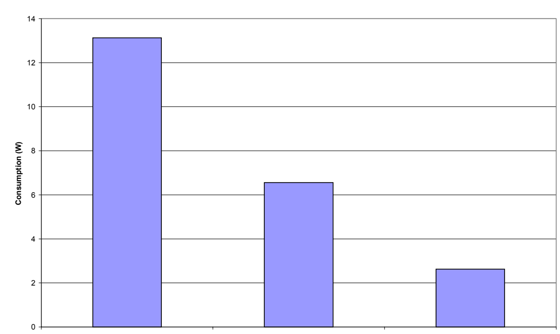

Figure 4 – Energy consumption of fan technologies.

6. GENERAL EQUATION FOR THE DAILY CONSUMPTION

The daily consumption of a freezer can be calculated by the general equation (10) [EUR00].

$$C_{Wh/day} = \frac{HL}{COP} \times 24 + \left(P_{fan} + P_{def}\right) \times \left(1 + \frac{1}{COP}\right) \times 24 \qquad (10)$$

The fan is running only when the compressor is running, so the average power of the fan is calculated using the running time ratio. The energy of the fan is released into heat that is removed as a thermal load by the compressor.

The defrosting heater runs once every 24hrs. The average power consumption of the heater is the energy consumed for a defrost divided by 24hrs. The heat generated by the heater is removed by the compressor as well.

For the natural convection freezer, the terms of the fan and the heater are nil.

7. RESULTS AND DISCUSSION

Using equation (10), the daily consumption of the natural convection appliance is calculated. This consumption is compared to the one of the no-frost appliance calculated using the 3 technologies of fans and 3 defrost efficiencies. The results are shown in figure 5.

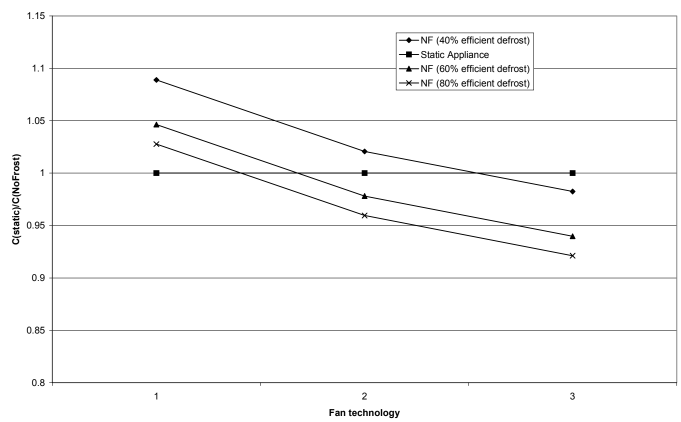

Figure 5 – Consumption of the no-frost appliance relative to the static appliance.

Figure 5 permits to analyze the impact of the technology of accessories on the consumption of the no-frost freezer.

The use of conventional technologies will lead to ~10% energy consumption increase. In order to achieve better energy performance than the natural convection appliance, two choices are available: either using the best available technology for one of the accessories and continue to use the conventional technology for the other; either using an "intermediate" technology for both accessories.

With the best available technologies for both accessories, the no-forst appliance can achieve an energy saving of 8%.

8. CONCLUSIONS

A no-frost chest freezer has been compared to an equivalent static freezer and a general energy consumption model is developed. The energy consumption of accessories is discussed and conclusions are drawn on the required energy efficiency in order to reach better energy performances for ventilated refrigerators and freezers.

The calculations show that with an "intermediate" technology of accessories, a no-frost appliance can offer a better energy performance than a static one. These technologies are widely available and their costs are affordable.

However, with the best available technologies no-frost appliances can achieve 8% energy saving with a higher cost. A life cycle cost analysis including appliance cost break down is necessary in order to conclude if these technologies are the best cost and consumer service compromise.

REFERENCES

[ASH94] Household refrigerators and freezers, Refrigeration handbook, ASHRAE 1994.

[ASH93] Standard 23-1993 - Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units (ANSI approved)

[BEJ94] Bejan, Vargas and Lim.When to defrost a refrigerator, and when to remove the scale heat from the heat exchanger of a power plant, 1994

[CLO01] Clodic D., Ciucasu C., EL Khoury K., Mercier F., Pan X., Pin O, Zoughaib A. ENEREF and CYCLEREF: Dynamic Simulation Tools for the Energy Consumption Calculation of Domestic Refrigerating Appliances. International Appliance Manufacturing 2001. p.131-135.

[DAN02] Danfoss, compressors and condensing units selection catalogue, 2002.

[EUR00] Revision of Energy Labeling and the 2nd Stage of Minimum Energy Efficiency Standards for Domestic Refrigeration Appliances. SAVEII-4.1031/Z/98-269. European Community Final Report. 2000.

[GRA86] Gray D.L., Webb R.L. Heat transfer and friction correlation for plate finned tube heat exchangers having plain fins, Proc. 8th Heat Transfer Conference, 2745-50, 1986.

[GUN97] Gungor K.E. and Winterton R.H.S. Simplified general correlation for saturated flow boiling and comparisons of correlation with dat', *Chem. Eng. Res. & des. ,* Vol.65, March 1997.

[ZOU00] Zoughaib A., Ciucasu C., Beaufils M., Clodic D. Analyse des consommations energétiques en fonction des caractéristiques des différentes classes d'appareils et des nouveaux objectifs de l'étiquetage energétique des réfrigérateurs-congélateurs domestiques. ADEME final report, 2000.