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Jim McGovern

Technological University Dublin, jim.mcgovern@tudublin.ie

S. Oladunjoye

Technological University Dublin

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NON-INTRUSIVE SECOND LAW PERFORMANCE EVALUATION OF A DOMESTIC FREEZER

J. A. MCGOVERN¹ and S. OLADUNJOYE¹

1. School of Mechanical and Design Engineering, Dublin Institute of Technology, Bolton Street, Dublin 1, Ireland; email: jim.mcGovern@dit.ie

Abstract

This paper describes non-intrusive performance testing and second law, exergy-based, performance evaluation of a small, contemporary domestic freezer that used R600a refrigerant. The methodology presented is associated with logging of temperatures and the instantaneous power consumption. It gives a 'second law of thermodynamics' perspective on operating performance by providing rational efficiency values and by localizing and quantifying the instantaneous exergy destruction rates. This approach is compared to a more conventional 'first law' energy analysis. Suggestions are made for the standardization of such an approach in order to enhance the sustainability of domestic freezers.

Keywords: Domestic freezer, Performance testing, Performance evaluation, Exergy, Second law, R600a

1 INTRODUCTION

Tests were carried out on a small, 40 litre, table-top domestic freezer that used R600a refrigerant, Figure 1. In this freezer the evaporator tubes were located between the insulation and the plastic internal liner of the sides, top and back of the unit and the top surface of the internal shelf formed by the compressor housing. The condenser tubes were likewise located between the insulation and the mainly steel external liner of the sides, top and back of the unit. For the tests, the freezer contained only three 500 ml plastic bottles filled with water. Temperatures were measured with thermocouples and logged with a data acquisition system. Where thermocouples were placed in contact with surfaces, they were covered with a pad of thermal insulation. Electrical power consumption was measured with an independent, instantaneous power logger. Infrared photographs were also taken of the surfaces of the freezer, inside and outside. All measurements were thus non-invasive. There was no direct measurement of the mass flow rate of the refrigerant or of pressures within the refrigeration circuit. Figure 2 is a rear view of the freezer and Figure 3 shows the bottles of water within the freezer.

2 TEST READINGS

Figure 4 shows test readings taken at one minute intervals for a twenty minute period that was towards the end of a twenty-four test. The figure includes one full cycle that starts at 23 minutes and ends at 34 minutes. Taking the five cycles,

including this one, that start at 0 minutes and finish at 58 minutes, the average ambient temperature was 19.5 °C and the average temperature of the frozen water in the bottles was -20.4 °C. These temperatures were steady. The average power consumption was 12.8 W.

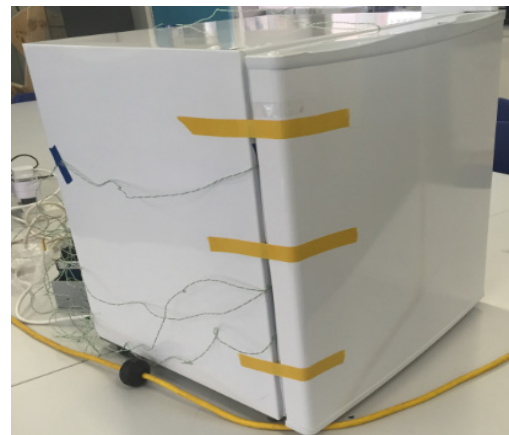


Figure 1. External View



Figure 2. Rear View

An infrared camera was used to investigate the distribution of temperature over the inner and outer surfaces of the freezer. Figure 5 shows one

of the external side panels and Figure 6 shows a view of the inside of the freezer during a test while the door was opened briefly. The infrared images allowed a thermocouple to be placed on the inside surface at the coldest point of the evaporator.



Figure 3. Inside View

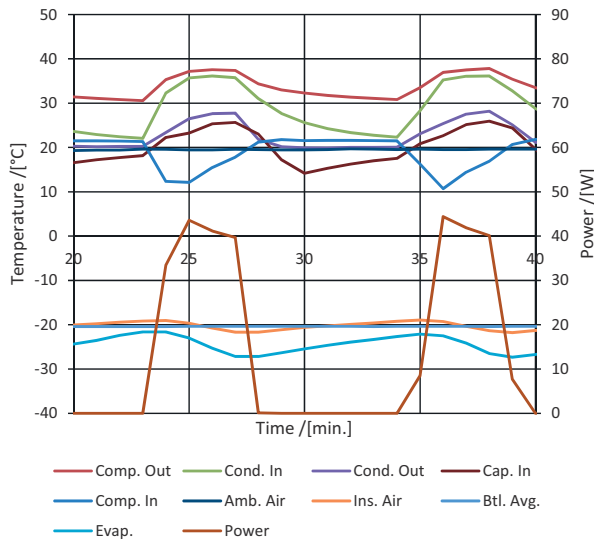


Figure 4. Test Readings

3 PERFORMANCE

The test measurements showed that a very steady temperature of the frozen water in the bottles was maintained, while the air temperature within the freezer fluctuated by about 3 K for each on-off-on cycle of the compressor. Based on the average power consumption at the stated conditions, the annual energy consumption would be 112 kWh.

The coefficient of performance (COP) of the freezer can be defined as the effective rate of heat gain, at the measured ambient temperature and the measured temperature of the items stored within it, divided by the average electric power input required, Equation (1).

$$\text{COP}_{\text{fr}} = \frac{\dot{Q}_{\text{in, eff}}}{\dot{W}_{\text{in}}} \quad (1)$$

It was noted that in this case the average internal temperature was very close to the mean temperature of the frozen water in the bottles.

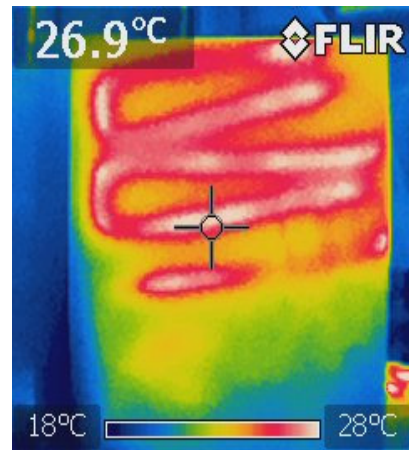


Figure 5. External Side Panel Infrared View

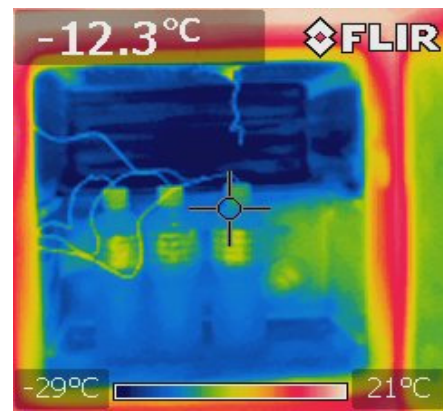


Figure 6. Inside Infrared View

Melo et al. [1] have described how the overall conductance (in W/K) of a freezer cabinet could be measured experimentally. This could also be calculated from the dimensions of the freezer, the thermal properties of the heat conducting materials, empirical correlations for the surface heat transfer coefficients and appropriate expressions based on the Stefan-Boltzmann law to take account of any radiation effects.

A simplified calculation of the overall conductance of this freezer was performed assuming one-dimensional steady state heat transfer from ambient air to the air within the freezer. The thermal conductivity of the insulation was taken as 0.025 W/mK. The surface heat transfer coefficients were taken as 8.29, 9.26 and 6.13 W/m²K for horizontal, upward and downward heat transfer respectively [2]. The estimated overall conductance of the freezer cabinet was 0.346 W/K. Hence, the rate of heat gain at a temperature difference of 39.9 K was 13.8 W. The calculated COP was therefore 13.8/12.8, or 1.08.

4 SECOND LAW PERFORMANCE

The electric power input to a freezer is equivalent to a rate of exergy input. The rate of exergy transfer \dot{X} associated with a rate of heat transfer \dot{Q} at absolute temperature T is given by Equation (2), where T_0 is the absolute temperature of the environment. The rational efficiency of the freezer is given by Equation (3), where T_{fr} is the absolute temperature of the items stored within it.

$$\dot{X} = \dot{Q} \left(1 - \frac{T_0}{T} \right) \quad (2)$$

$$\eta_{fr} = \frac{\dot{X}_{out}}{\dot{X}_{in}} = \frac{-\dot{Q}_{in,eff}}{\dot{W}_{in}} \left(1 - \frac{T_0}{T_{fr}} \right) \quad (3)$$

$$= \text{COP} \left(\frac{T_0}{T_{fr}} - 1 \right)$$

Hence, the rational efficiency was 17.0%. In order to understand the scope for improvement, this is a more useful performance parameter for the particular operating conditions than the COP. The rational efficiency defect (100%–17.0% =83.0%) quantifies the fraction of the exergy input rate that is destroyed or wasted. Such exergy destruction occurs in physical regions such as pipework where there is fluid friction, conducting materials where there is heat transfer through thermal resistance, or assemblies of moving parts where there is mechanical friction, and can be quantified. For example, McGovern and Harte [3,4] described exergy destruction rates within compressors.

5 DEDUCTIONS FROM TEST READINGS

The logged temperature measurements and instantaneous power measurements, together with software for thermodynamic properties [5], allowed detailed information to be deduced with reasonable accuracy. The saturation temperatures and hence pressures in the evaporator and condenser were implied from thermocouple measurements at positions where the refrigerant was in the saturated mixture state. Energy and exergy balances were applied to systems and subsystems that underwent no net change. For instance, the compressor and condenser undergo no net change for an integral number of complete on/off cycles of the compressor, once an unchanging pattern has been established. Fifty-two successive time points at one-minute intervals, corresponding to five on/off cycles,

were used for the results presented. Average values of eleven measured parameters were determined for the five cycles and also for the fraction of the five cycles during which the compressor was running, as shown in Table 1. As there was no separate thermocouple for the condensing saturation temperature, the average value of the measured condenser inlet temperature over the entire cycle was used, both for the analyses of the entire cycle of the parts of it for which the compressor was running.

Table 1. State Points, Compressor Running

	t /[°C]	p /[kPa]	x
1 comp in	15.8	59.7	
2 comp out	36.2	387.9	
3 cond in	33.8	387.9	
4 cond out	25.7	387.9	
5 cap in	23.8	387.9	
6 evap in	-24.5	59.7	0.127
7 evap out	-24.5	59.7	1.0
8 amb air	19.5		
9 btl avg	-20.4		
10 ins air	-20.2		
Avg. power input /[W]		29.8	

Subcooled liquid, at state point 5, left the condenser. This freezer had capillary/suction-line heat exchange. The dryness fraction entering the evaporator was determined by assuming liquid subcooling due to heat exchange with the suction vapour, followed by adiabatic throttling. It was also assumed that the refrigerant leaving the evaporator was dry saturated as it entered the liquid/suction heat exchange region.

5.1 Rough Estimates for the Refrigerant Cycle

Owing to the absence of any mass flow rate measurements, it was not possible to achieve precision in analysing the refrigeration cycle. However, it was possible to make some useful rough estimates. The logged measurements indicated that the on/off cycles were highly consistent. However, as the data-logging time interval was one minute and the cycle period was about 11 minutes, it was necessary to use multiple cycles to estimate the average values of power input and temperature to reduce errors associated with the sampling interval. It was assumed that the liquid/suction heat exchanger region had negligible heat transfer other than

that between the liquid and vapour streams of refrigerant.

A number of attempts were made to infer mass flow rates from energy balances for parts of the system, making use of the available measurements. For instance, an energy balance on the compressor was considered. This required estimation of the heat loss rate from the compressor by convection and radiation. While infra-red temperature measurements would have assisted, the uncertainties were still excessive due to the intermittent running. A more satisfactory option was found. This was to make use of the rated COP and cooling capacity of the compressor, according to ASHRAE standard 23.1 [6]. These data were available from the compressor manufacturer's brochure [7].

5.2 Overall Isentropic Efficiency

Table 2 compares key operating parameters of the compressor, at the ASHRAE rating point and while the compressor was running.

Table 2. Compressor Operating Points

	ASHRAE	While running
$t_{\text{evap}} / [^{\circ}\text{C}]$	-23.3	-24.5
$t_{\text{suct}} / [^{\circ}\text{C}]$	32.2	15.8
$t_{\text{cond}} / [^{\circ}\text{C}]$	54.4	28.5
$T_{\text{liq}} / [^{\circ}\text{C}]$	32.2	23.8
$t_{\text{disch}} / [^{\circ}\text{C}]$	-	36.2
Press. ratio	12.1	6.49
$t_{\text{amb}} / [^{\circ}\text{C}]$	32.2	19.5
COP	1.4	-
Cooling cap. / [W]	68	-
Compr. $E_{s,\text{overall}}$	48.3%	-

The average parameters of the compressor, while it was running during the test, were clearly not the same as those at the rating point, yet it was to be expected that the rating data could provide supplemental information that could be used in the estimation of the refrigerant mass flow rate. In order to achieve this, it was decided to make use of the overall isentropic efficiency of the hermetic compressor, Equation (4).

$$E_{s,\text{overall}} = \frac{\dot{m}_{\text{refr}}(h_{\text{disch},s} - h_{\text{suct}})}{\dot{W}_{\text{in,elec}}} \quad (4)$$

The denominator in Equation (4) is the actual electrical power input, \dot{m}_{refr} is the actual refrigerant mass flow rate, h_{suct} is the suction specific enthalpy and $h_{\text{disch},s}$ is the specific

enthalpy at discharge, assuming isentropic compression. This overall isentropic efficiency can be evaluated when the suction state (e.g. temperature and pressure), the discharge pressure, the mass flow rate and the electric power input are known.

Pérez-Segarra et al. [8] have provided detailed thermodynamic characterization of common performance parameters of hermetic reciprocating compressors, including a number of variants of isentropic efficiency and the overall mechanical/electrical efficiency. The overall isentropic efficiency of Equation (4) is equivalent to the product of an isentropic efficiency and the overall mechanical/electrical efficiency of a compressor. From the information presented in reference [8] it is apparent that $E_{s,\text{overall}}$ is unlikely to vary greatly between the ASHRAE rating point and the actual 'compressor running' point of Table 2. From the rating point data, the overall isentropic efficiency of the compressor was determined to be 48.3%.

5.3 The Cycle while Running

The cycle while the compressor was running (Table 2) was used to estimate the average refrigerant mass flow rate while the compressor was running as 178×10^{-6} kg/s. This was done by solving for the unknown mass flow rate in Equation (4), on the assumption that the overall isentropic efficiency of the compressor E_s was the same as at the rating point. It is important to appreciate that the energy of a compressor changes while it runs intermittently, as does the amount of refrigerant it contains. There was therefore insufficient information to carry out energy and exergy balances.

6 THE EQUIVALENT AVERAGE CYCLE

The equivalent mass flow rate of refrigerant for an integral number of on/off cycles was calculated by multiplying the average mass flow rate while running by the run-time fraction. This equivalent mass flow rate was 73.6×10^{-6} kg/s. For the purposes of (rough) analysis an equivalent cycle based on the time-averaged parameters, Table 3, was used. Some energy and exergy analysis results are also shown in this table.

There is scope to localize rates of exergy destruction in very fine detail. For instance, the rate of exergy destruction associated with heat

Table 3. Equivalent Cycle Performance

$t_{\text{evap}} / [^{\circ}\text{C}], p / [\text{kPa}]$	-24.4	60.0
$t_{\text{suct}} / [^{\circ}\text{C}], p / [\text{kPa}]$	19.1	60.0
$t_{\text{cond}} / [^{\circ}\text{C}], p / [\text{kPa}]$	28.5	388
$t_{\text{disch}} / [^{\circ}\text{C}], p / [\text{kPa}]$	33.7	388
Press. ratio	6.47	
$t_{\text{amb}} / [^{\circ}\text{C}]$	19.5	
Energy Analysis (1st law)		
Electric power / [W]	12.8	
Eff. cooling load / [W]	13.8	
Compr. $E_{s, \text{overall}}$	46.9%	
Evap. cooling rate / [W]	25.3	
$\text{COP}_{\text{cycle}} = \frac{\dot{Q}_{\text{in, evap}}}{\dot{W}_{\text{in, elec}}}$	1.97	
$\text{COP}_{\text{fr}} = \frac{\dot{Q}_{\text{in, eff}}}{\dot{W}_{\text{in, elec}}}$	1.08	
Exergy Analysis (2nd law)		
$\dot{X}_{\text{out, eff}} / [\text{W}]$	2.3	
$\dot{X}_{\text{out, evap}} / [\text{W}]$	4.46	
$\eta_{\text{compr}} = \frac{\dot{X}_{\text{out, comp}}}{\dot{W}_{\text{in, elec}}}$	42.9%	
$\eta_{\text{cycle}} = \frac{\dot{X}_{\text{out, evap}}}{\dot{W}_{\text{in, elec}}}$	34.8%	
$\eta_{\text{fr}} = \frac{\dot{X}_{\text{out, eff}}}{\dot{W}_{\text{in, elec}}}$	17.0%	

transfer between the liquid line and the suction line was evaluated through the use of the specific flow exergy function $b = h - T_0 s$ for each of the streams. The rate of energy transfer from the liquid line to the suction line was evaluated as 5.0 W from the mass flow rate and the increase in specific enthalpy of the suction vapour. The corresponding exergy transfer rates had the opposite direction: the exergy output rate from the vapour stream was 0.41 W, while the exergy input rate to the liquid stream was 0.26 W, giving an exergy destruction rate of 0.15 W (a small price to pay for the benefits this heat exchange brings to the refrigeration cycle).

7 COMPARISON OF FIRST AND SECOND LAW APPROACHES

It needs to be stated that the first and second law approaches are not alternatives, but rather support one another as a means of assessing and

quantifying the performance of a domestic freezer. Whereas the compressor had a nominal COP of 1.4 at its rating point, in practice the real refrigeration cycle had a COP of 1.97, or the freezer had COP for meeting the effective heat gain through its insulation of 1.08. The difference between the latter two COPs is because the evaporator on the inside of the cabinet and the condenser on its outside cause additional heat gain, which also must be countered by the refrigeration system.

COP values are not intuitive, as the COP of a perfect thermodynamic refrigeration plant depends on the temperature levels involved. Rational efficiency values would all be unity for perfect thermodynamic devices and are therefore more intuitive.

The domestic freezer that was studied had a complicated tangle of energy flows. Probing deeply enough to put numbers on these often fails to explain where problems lie. In exergy analysis, every rate of exergy destruction takes place within a physical region—it can be localized. The thermodynamic problem is invariably one of exergy destruction. Then economics and practicality come into play. A thermodynamically perfect freezer would have such excellent insulation that its effective heat gain rate would be negligible, and it would be preferable never to open it!

8 THE METHODOLOGY

The elaboration of the methodology described herein was taken on by the first author as a kind of personal challenge. It was met only by pushing the terms of reference further than originally intended. The second author undertook the experimentation within the scope of a thesis project in partial fulfilment of the requirements for a master's degree.

8.1 Thoughts and Suggestions

- In this work, infrared photographs proved very useful. Thermal maps obtained in this way are ideal for heat transfer analysis and for evaluation of the associated rates of exergy transfer or destruction.
- It would be nice if all mass-produced fridges and freezers came with some form of low-cost mass flow rate sensor built-in.
- It would be useful for the documentation or nameplate of each domestic freezer to include the overall thermal conductance of the cabinet. For each hermetic compressor, it

would be useful if manufacturers automatically provided its surface area, emissivity and perhaps a regression curve expressing the rate of heat loss in terms of its surface temperature and the temperature of the surroundings.

- For the hermetic compressor, the standard rating point performance parameters, consisting of the COP and the cooling capacity, proved useful for this work. Had the discharge temperature also been provided, that would have been valuable. It would allow disaggregation of heat rejection that occurs from the compressor and that which occurs from the condenser.

8.2 Intermittent Running

Intermittent running certainly poses a challenge for the second law assessment of the performance of a domestic freezer. However, when the contents have reached their set temperature and the surroundings are steady, a regular pattern is set-up and the system returns repeatedly to the same state. This is ideal for first and second law analysis. Energy balances and the Clausius inequality, which gives rise to the Guoy-Stodola theorem, can be applied over surfaces and over time and convenient equations for the first and second law analyses are available [9]. For the coarse analysis described here, an estimate of the average mass flow rate was made for the ‘compressor running’ period and the balance of plant was modelled as being in steady flow. Detailed computer simulations are capable of achieving highly precise first and second law analyses of intermittent or transient performance.

9 CONCLUSIONS

The methods described here for second law performance evaluation can be applied to fridges or freezers to construct progressively more detailed analyses of where exergy is destroyed and to localize exergy destruction rates along the paths of energy flow within them. The same techniques can be incorporated in detailed simulation models for design and development and this can enhance the sustainability of these devices. In the absence of better information, the overall isentropic efficiency of a hermetic compressor is a useful parameter that may not vary too greatly over a range of operating conditions.

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