Project Report 7

REFRIGERATION ENERGY USE IN THE FOOD CHAIN

by

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Summary

This report summarises the research carried out by the LSBU team into the 'Food Chain Refrigeration Energy Use ' between March and October 2008. The report describes and evaluates the continued development and validation of the VCR software and provides charts comparing the experimental and model data. The results were found to be very encouraging. Some suggestions are made for the improvement of the system used in the case study.

17 October 2008

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1. Introduction

A project objective for LSBU is to develop integrated transient refrigeration food process system models to test (and optimise) various refrigeration systems used in the industrial production chain for food. This report describes the results of LSBU's progress up to 12 October 2008.

2. Model developments

The main part of the research during this period has been to increase the number of refrigeration system control options available to the program user. This has included the following:

Compressor: The user may now select up to 6 compressors. The control options include ‘staged-running’, manual and automatic speed control. Thermostat control of the automatic and staged running options is available.

Evaporator fan: An automatic speed control function has been added. Control of the fan speed is based on the setting of a thermostat within the ‘Refrigerator Design Screen’. Manual control is still available within the ‘Run Screen’.

Condenser fan: Automatic speed control has been added. As with the evaporator fan, this is activated by a choice of thermostat settings. Manual control is still available.

Compressor protection: A compressor start-up timer code has been added to the software. In practice this protects a compressor by preventing it from being started more than a specified number of times per hour. The time delay between consecutive compressor starts can be set by the user within the Refrigerator Design Screen.

Stage start-up and shut-down: In practice it is usual to stage the start-up of electric motors in order to reduce the surge current associated with inductive electrical loads. The needed to simulate the effects of staging the start of condenser fan motor, the evaporator fan motor and compressor motors is particularly important when the operation of a refrigeration system is highly transient, as in the Validation Case Study, described in the next section.

Thermostat options: The range of options for the control thermostat has been extended to include evaporator saturation temperature, evaporator air-on and air-off. At the request of FRPERC the thermostat function can also be switched of within the Refrigerator Design Screen.

Defrost options: This now includes options to choose either a timed defrost or initiated by the temperature difference between evaporator coil air-on temperature and saturation temperature.

3.0 The Validation Case Study

Background: Pieminister is a pie manufacturer based at Brentry, Bristol. The company batch produce pies of various types. This is a labour intensive process. The blast-cooler used to validate the VCR refrigeration system model was manufactured by Fosters (Figure 1). The dimension of the unit, condenser, evaporator and pipe-line information was recorded by the FRPERC team (Table 1).

At the time of our visit it was being used to cool cooked pie filling from about 75^{o}C to about 10^{o}C. After cooking the pie filling in tureens, these are then covered with cling film, stacked on to trolley racks and pushed into the blast- cooler (Figure 1). The blast-cooler air-on coil temperature is controlled by thermostat. Figure 5 shows a schematic plan view of the blast-cooler, showing the layout of the system and the direction of the air-flow through the evaporator, fans and across the food product.

Figure 1: Pie-filling cooling room at Pieminister,
(door dimensions – width = 71 cm, height = 174 cm). Manufactured by Fosters

Table 1: Dimension of the blast freezer and refrigerator components required as input data for the VCR model

Condenser dimensions

Width = 66 cm
Height = 77 cm
Depth = 36 cm
Coil pipe = 9.7 mm O/D
Coil length = 30 rows by 6 deep
Fin spacing = 8 per inch (Aluminium)
Mean face velocity (Air on) = 3.4 m.s[1]

Evaporator dimensions

Width = 66 cm
Height = 162.5 cm
Depth = 30 cm
Coil pipe = 6.4 mm O/D
Coil length = 60 rows by 4 deep
Fin spacing = 8 per inch (Aluminium)
Mean face velocity (Air on) = 2.1 m.s^{-1}

Pipe lengths

Evaporator to compressor = 38.5 m x 28.6 mm O/D (Armaflex insulation = 15 mm thick)
Compressor to condenser = 3 m x 28.6
Condenser to receiver = 3 m x 19.1 mm O/D
Receiver to evaporator = 38.5 m x 19.1 mm O/D

Blast chiller dimensions (Foster)

Internal length = 195.0 cm
Internal width (between inner wall and evaporator face) = 86.5 cm
Internal height = 195.5 cm
External length (excluding doors) = 221.0 cm
External width = 163.0 cm
External height = 226.5 cm

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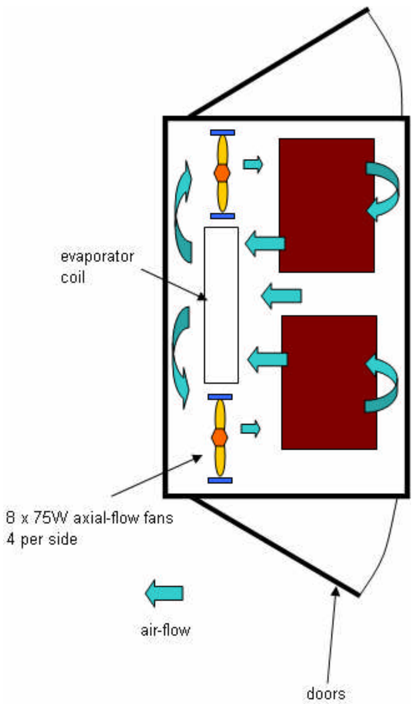

Figure 2: Schematic plan view of blast-cooler

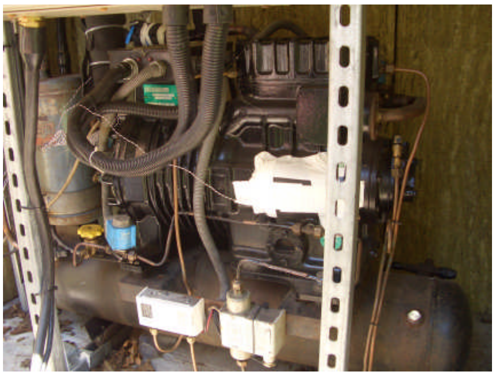

Figure 3: Semi-hermetic (air-cooled?) Presscold, (now owned by Copland) compressor unit, Model R1500/0138 S/D (Presscold model number), Serial No 9483 (380/420VAC, 3-phase, 32 FLA, 40 LRA) Reconditioner's phone number 01491 629671. See Appendix A for technical data

The refrigeration plant: The refrigeration unit is an air-air DX system powered by a single compressor. The compressor was a reconditioned semi-hermetic Presscold compressor unit, Model R1500/0138 S/D, Serial No 9483 (380/420VAC, 3-phase, 32 FLA, 40 LRA). Figure 2 shows a photograph of the compressor installation. The technical data for the compressor is provided in Appendix A. The system included a high-pressure liquid receiver (approximately 100cm x 30cm diameter), filter/drier in the liquid line and a hot-gas line oil separator/muffler. The liquid and suction lines are long as seem in Table 1. The suction line is insulated with a standard type of foam insulation.

There were a number of differences between the VCRmodel system and that used for the case study. These are listed in Table 2. Before validation could be carried out the model was further developed to remove these differences.

Table 2: Comparison between the validations 'blast-cooler' system at Pieminister and the VCR model.

Pieminister system	VCR model	Comments
System charged with R22	At present on R404a is available option	Include R22 properties option
No suction line heat exchanger	Suction line heat exchanger option is available	The software can simulate a system without a slhx by setting its effectiveness to zero in the Refrigerator design screen.
An hp liquid receiver is included.	Not included	This option could be added.
A hot-gas line oil separator/muffler(?) is included	Not included	This option could be added
The thermal mass of the freezer cabinet is probably significant	The blast-freezer model neglects the thermal mass and volume of the blast freezer	The cold-store model could be used instead to simulate the system. (This was used as shown by the results given later)
Evaporator fans are switched off for defrost cycle	Evaporator fans are left on	An option to switch the fans off at defrost will need to be included

Operating and control strategies: According to the company, the refrigeration plant operates daily between 8 to 9 am and 11 pm, Monday to Friday. The company believed the blast-cooler takes approximately 45 minutes to pull-down from ambient conditions to it normal operating temperature. However, test results indicate the pull down period to be less than 10 minutes. A thermostat controls the air-on temperature to the evaporator coil. According to the company its upper set-point is -10^{o}C and its lower set-point is -12^{o}C. The defrost is by means of electric resistance heater coil embedded within the evaporator tube rows. The defrost cycle is timed to come on every 5 hours and stays on for 30 minutes. The evaporator fans are switched off during defrost. Food product normally enters the blast-cooler at a temperature between 75^{o}C and 85^{o}C. The target cooling temperature is 10^oC. It normally takes 6 hours to achieve this. However, it can take up to 12 hours, depending on the air gap between layers of tureens.

Other issues: The compressor enclosure was insulted to reduce noise emissions. This would have caused the temperature of the air surrounding the compressor to be high, which over time would have caused early failure were it not for the company fitting a ventilation fan to the enclosure. The power of the ventilation fan is unknown at this time.

Refrigeration system control settings: In order to model the system it was necessary to determine a number of control settings, such as thermostat upper and lower set points, compressor start-up delay timer setting , timer settings for fan starting and defrost power and timer settings. This was achieved by examining the charts shown in Figures 4 and 5 draw using experimental data supplied by the FRPERC team.

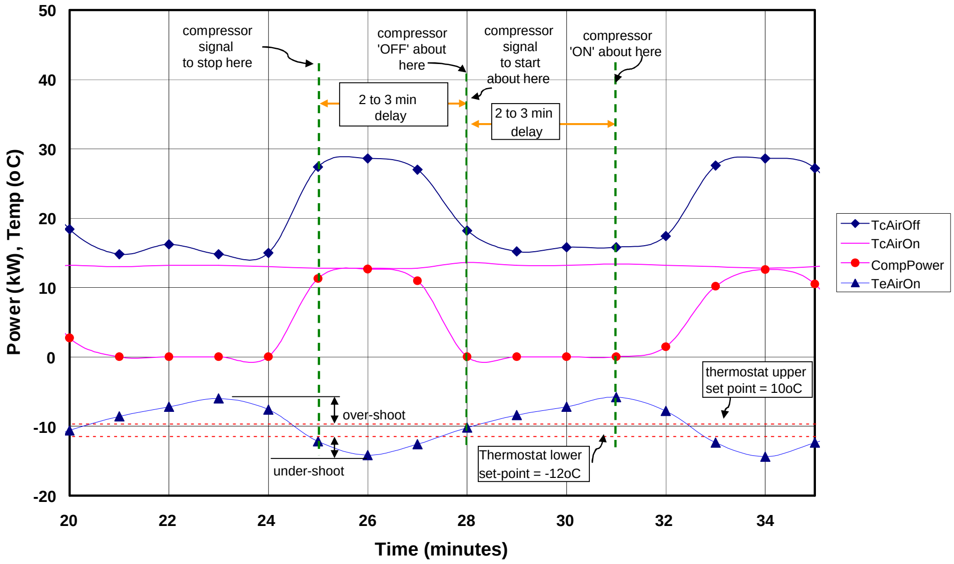

Figure 4: Experimental data – motor control characteristics of the blast-freezer

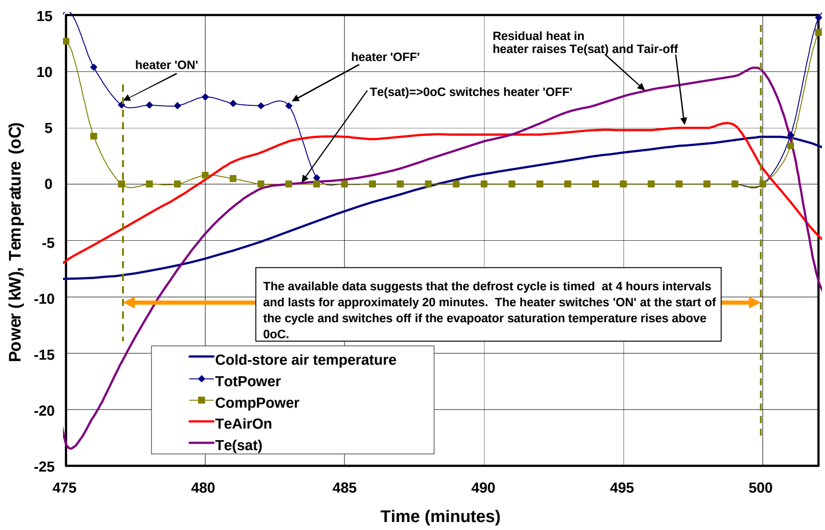

Figure 5: Experimental data - defrost control details for the evaporator

4 Software validation results

Using system data supplied by the FRPERC team and that gathered during our visit to Pieminister Ltd and from data supplied by the compressor manufacturer, the VCR refrigeration system model was set-up and run for the validation experiment. The output results from the VCR model are compared with the experimental data in the following Figures 6 to 12.

Figure 6 compares the experimental data for evaporator saturation temperature and evaporator air-on temperature with that predicted by the VCR model. There appears to be a close correlation between the data. Figure 7 shows a comparison between the experimental and model data during the first 60 minutes of run-time after the system was started. It is seen that the model predicts accurately the rapid switching ON and OFF of the compressor. This makes the system highly transient, which supports the view that a highly developed transient models of refrigeration systems are required by system designers if energy efficiency are to be made. Figure 8 compares the predicted and actual variation in electrical power used by the system over the first 60 minutes of operation.

According to the recorded data Case Study system used 64.2 kWh of electricity during the 570 minutes of the test run. The model predicted a total energy usage of 43. 7 kWh. However, the VCR model did not account for the additional power used by the compressor enclosure cooling fan. If this was taken into account the results would be much closer.

Figures 9 and 10 show the results for both the measured data and the VCR model during the time interval when the food product was loaded and the first defrost cycle was initiated. There appears to be some differences in predicted and actual evaporator saturation temperature during this period. This is though to have been caused by thermostatic valve hunting when the condenser temperature increased during this period. Thermostatic valve hunting was not present in the model, although the phenomenon can be simulated. This topic will be discussed in the following section.

Figure 11 shows the variation in predicted and measured food-product surface temperature during the chilling process. This predicted results to vary from the measured data as do those for the food-core temperature shown in Figure 12.

Figure 13 shows the predicted and measure results for the ambient air temperature and condenser saturation temperature. The difference between the measured and predicted results may be explained be the error in the predicted ambient air temperature, which controls the temperature of the air-on to the condenser.

5 Observations from the case study data

The measured results shown in Figure 9 for the evaporator saturation temperature indicate that the thermostatic expansion valve (TEV) was hunting when the condenser temperature increased during relatively longer periods of compressor operation. Hunting is caused by either or a combination of oversized valve or a poorly located sensing bulb. This results in the evaporator being alternately flooded and starved of refrigerant fluid which causes the otherwise unexplained rise and fall in saturation temperature. The VCR model has a facility to simulate the hunting phenomenon.

Figure 14 shows the VCR prediction of how evaporator saturation temperature falls as a TEV is increasing undersized whilst Figure 15 shown the effects of over sizing and poor location of the sensing bulb. These results demonstrate how a system designer might use the VCR software to explore options in TEV selection.

Figure 16 shows a close-up of the variation in evaporator saturation temperature during one ON/OFF compressor cycle, which indicates the measured and predicted effects of TEV hunting. This result indicates the ability of the VCR software to simulate the complex nature of refrigerator. However, as shown in Figure 17, the hunting phenomenon can be complex.

A further important application for the VCR software is in the selection of components. Table 2 lists the predicted energy usage for the Pieminister Case Study. Table 2a lists the predicted energy consumption for the Case Study whilst Table 2b lists that predicted if a replacement compressor with a design cooling capacity of 6.5 kWc were to be fitted. The results indicate and overall energy saving of 37% and a similar reduction in CO_2 emissions. Figure 18 compares the predicted variation in both food-product surface temperature and evaporator saturation temperature over the whole chilling process. These results suggest that there is little effect on the mean temperature of the food-product surface resulting from the change in the compressor's design-point cooling capacity. However, Figure 19, which shows a close up of the variation in the evaporator saturation temperature for both the 13 kWc and 6.5 kWc compressors, shows clearly a marked reduction in the switching ON/OFF frequency for the small unit.

Table2 (a) Design-point cooling capacity of the refrigerator = 13 kWc	**Table 2(b) Design-point cooling capacity of a 6.5 kWc refrigerator**
Total energy = 43.7 kWh **Compressor = 17.7 kWh** **Condenser fan = 8.3 kWh** **Evaporator fan = 13 kWh** **Defrost = 4.7 kWh** **Carbon produced = 17.9 kg (1.88 kg/h)**	**Total energy = 27.3 kWh** **Compressor = 9.28 kWh** **Condenser fan = 6.94 kWh** **Evaporator fan = 6.42 kWh** **Defrost = 4.7 kWh** **Carbon produced = 11.5 kg (1.21 kg/h)**

6. Conclusions

At the outset of this project we knew that computer models of refrigeration systems that were available were not capable of modelling complex applications and although new theoretical ideas relating to modelling compressors, expansion valves, heat exchangers, pumps and fans, had been developed by the LSBU team they were untested.

During this project the theoretical ideas concerning modelling of refrigeration systems have been validated. The operation and performance of refrigeration systems used in the food processing industry can be extremely transient as evidenced by the frequent on/off switching of the compressor in the validation case. We now know that the energy consumption of whole refrigeration systems can not be estimated with any reasonable accuracy from steady-state modelling alone and that transient modelling of the type provided by the VCR model provides a clear way forward.

The research carried out as part of this project clearly shows that whole system efficiency is equally a function of component selection, steady-state design and transient operation. Any weakness in any of these factors will strongly impact on the energy consumption and carbon release of a system

Research is now needed to provide the following:

(1) Further validation case studies.
(2) Extend the range of component and system options is needed to reflect those currently being used by industry to provide comparisons between various components types and system types.
(3)Extension of refrigerant property library

If the software is to be made generally available then further work is require to make the software more robust and useful by (1) improving the error trapping function and (2) data output function requires upgrading.

In summary,

- The VCR software has been validated using data from the Pieminister case study
- Results are extremely encouraging
- Using the model it can be shown what energy savings are possible
- Transient modelling is an essential design tool if significant energy savings are to be made

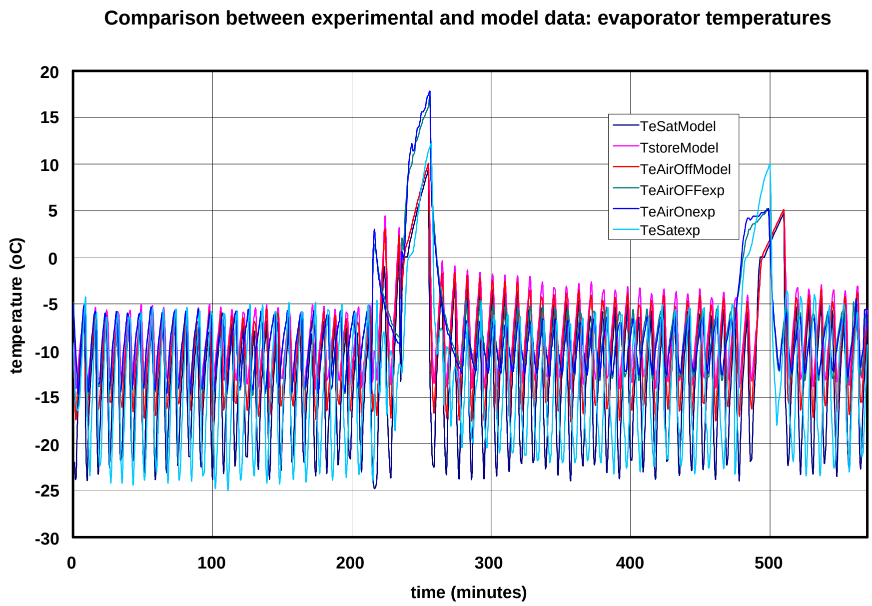

Figure 6

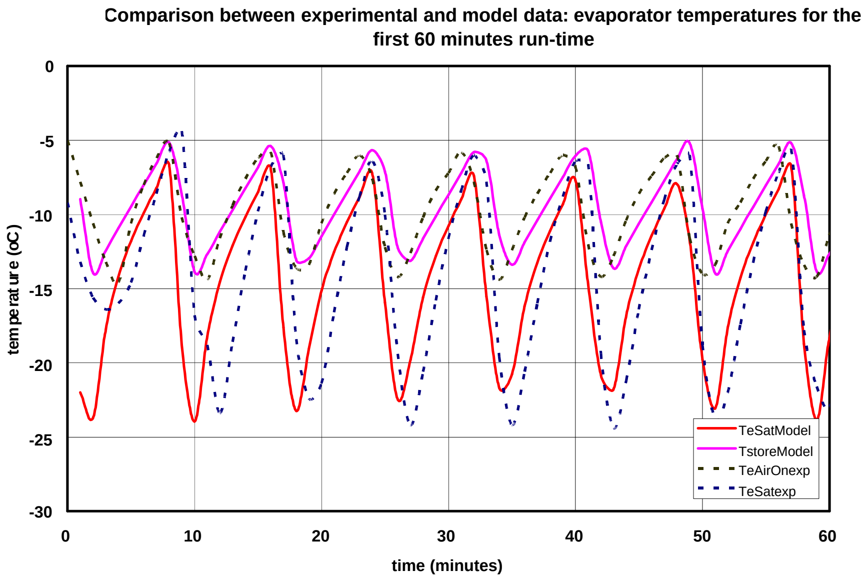

Figure 7

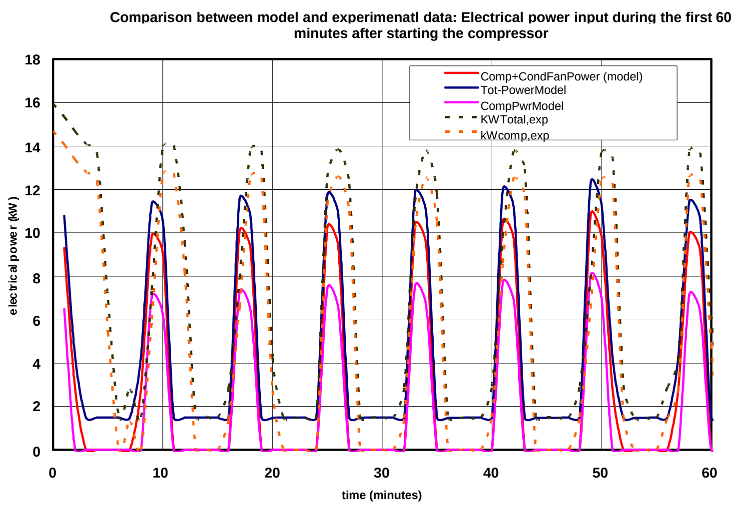

Figure 8

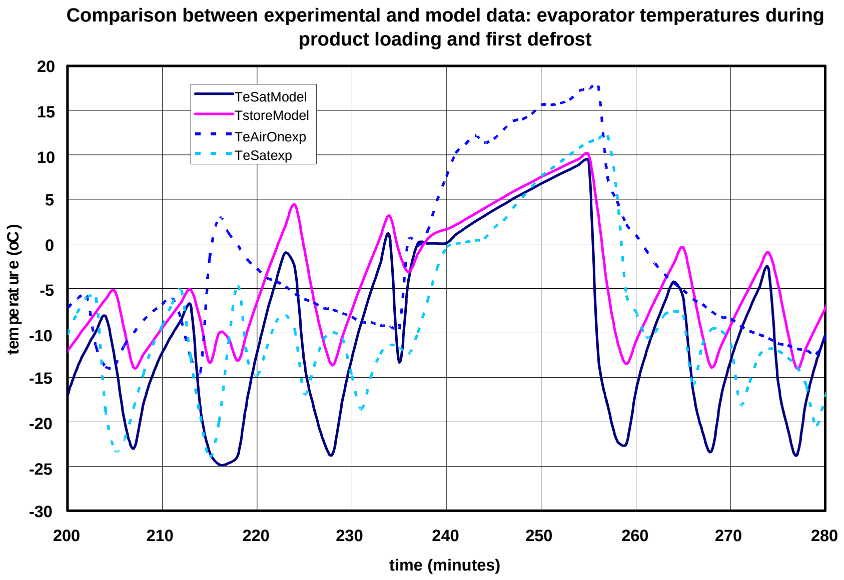

Figure 9

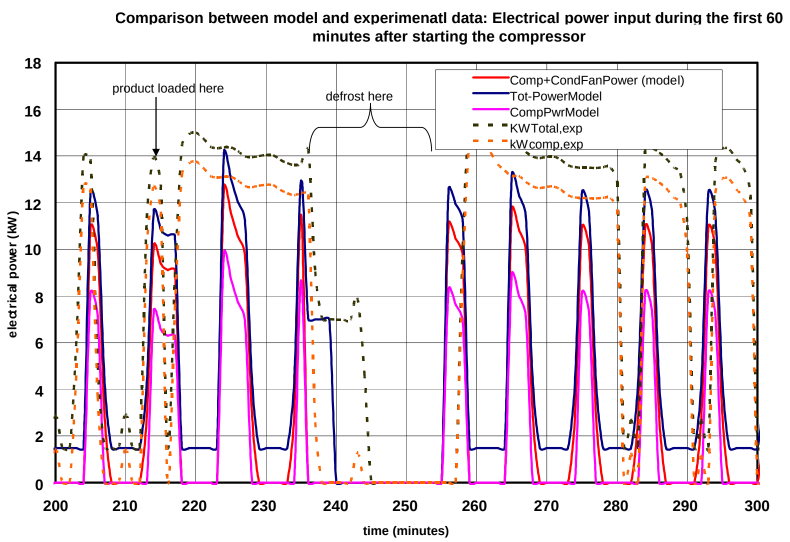

Figure 10

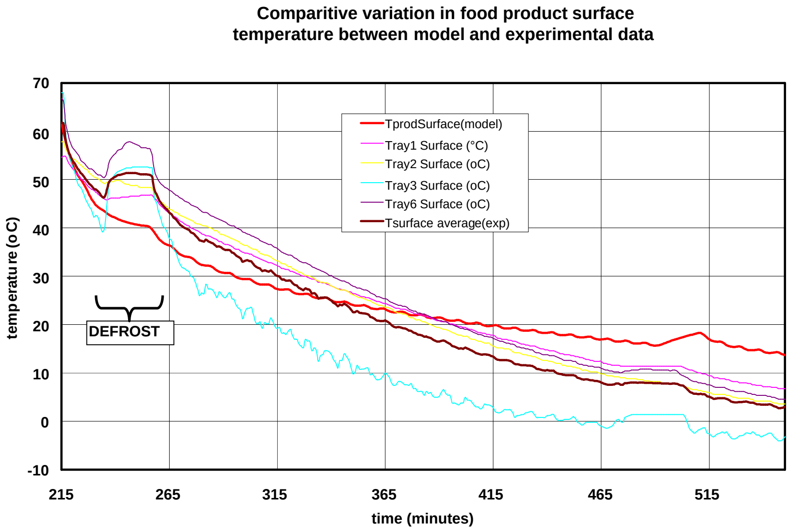

Figure 11

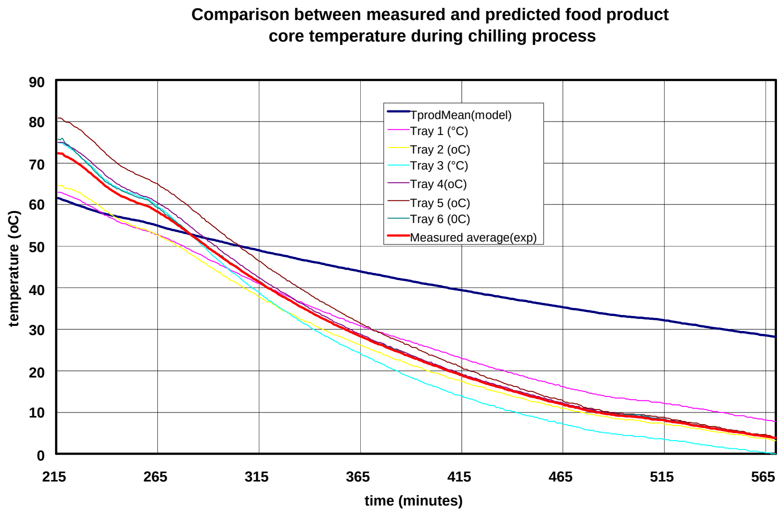

Figure 12

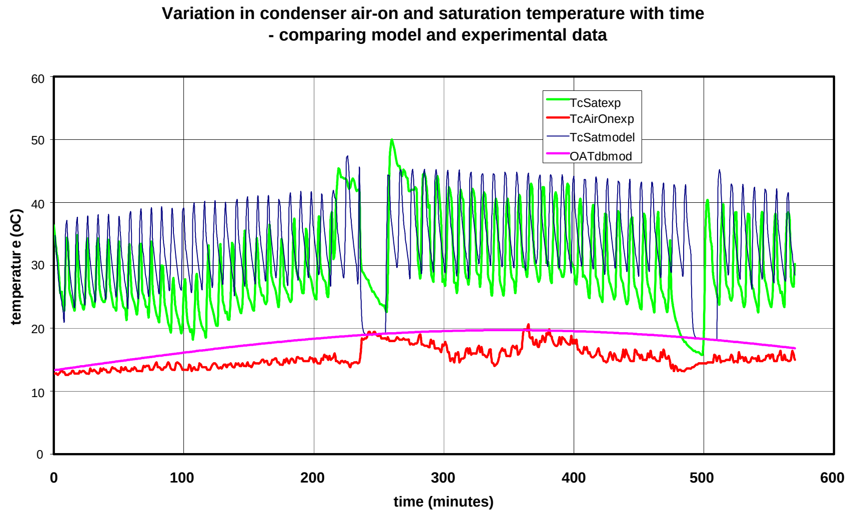

Figure 13

Predicted variation in evaporator temperature with time:
Effects of an undersized TEV starving an evaporator

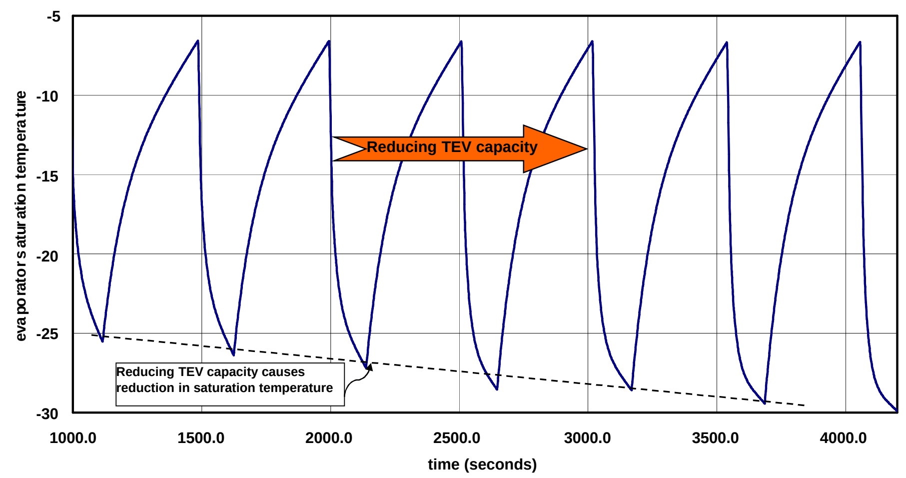

Figure 14

Predicted variation in evaporator temperature with time:
Effects of oversized TEV and reduced response time

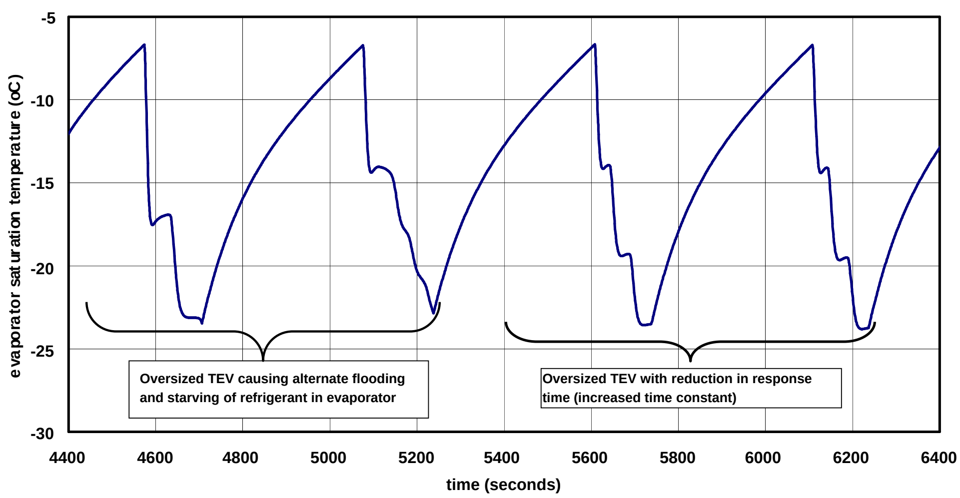

Figure 15

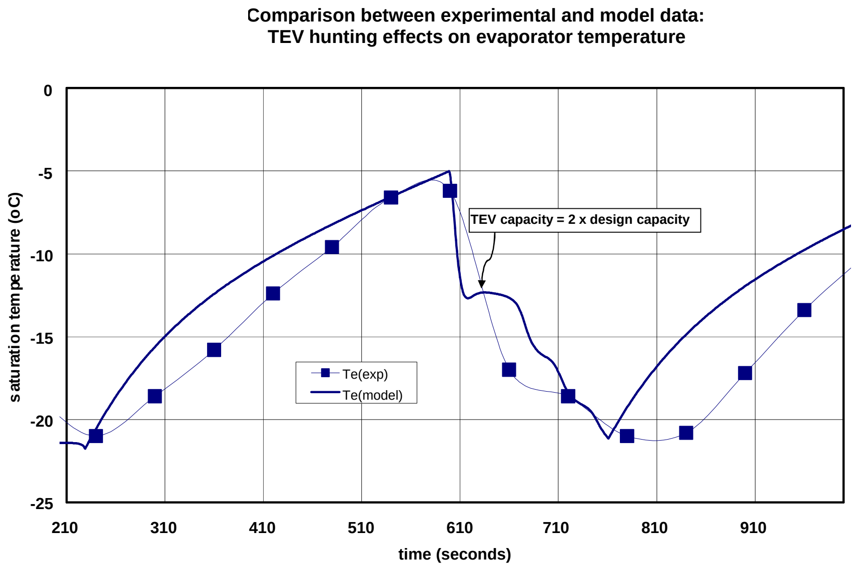

Figure 16

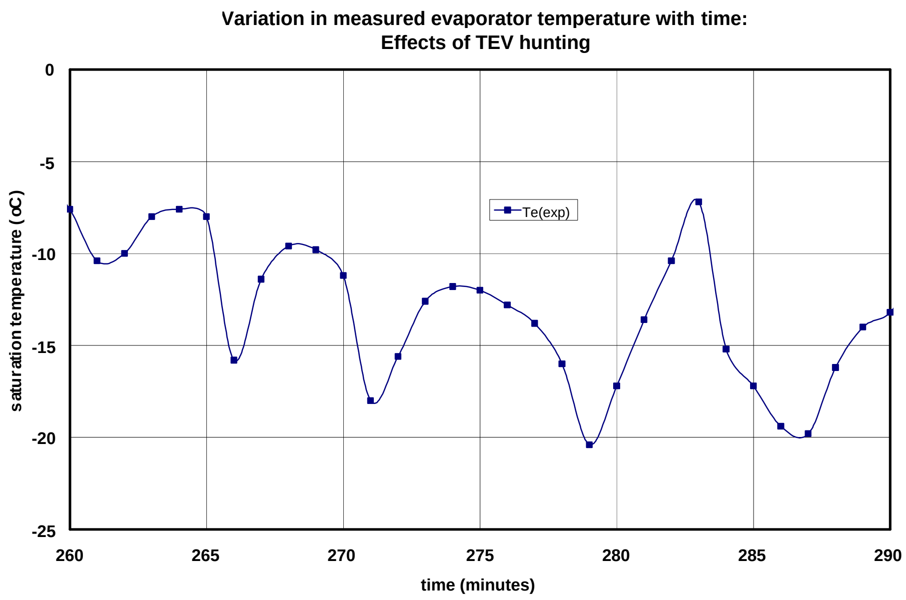

Figure 17

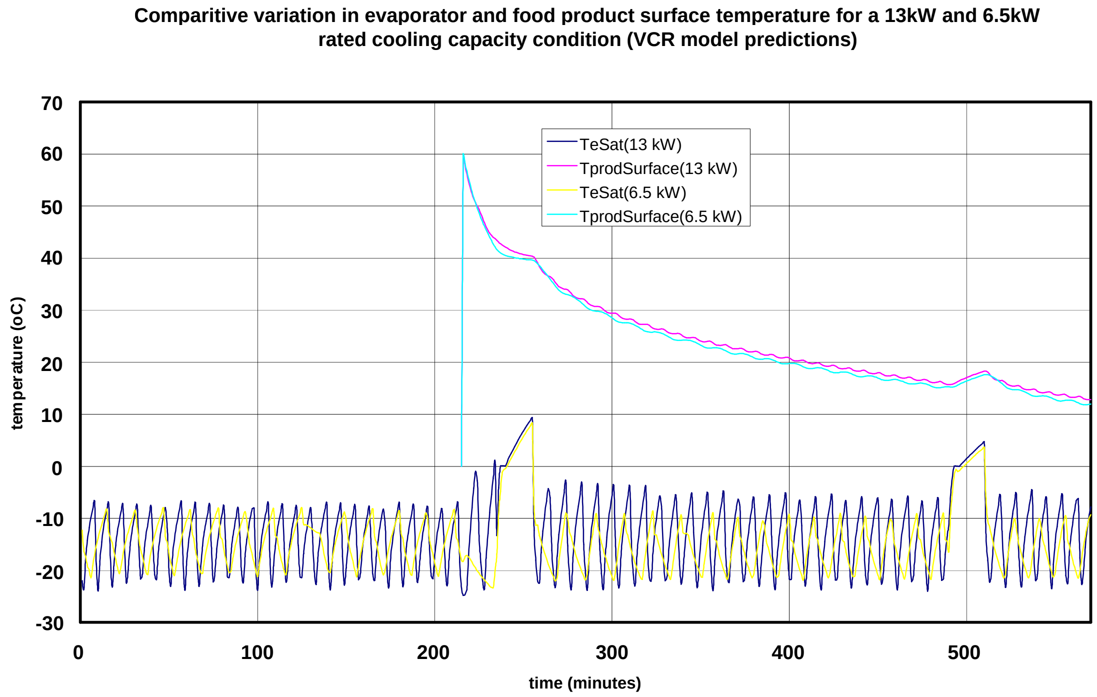

Figure 18

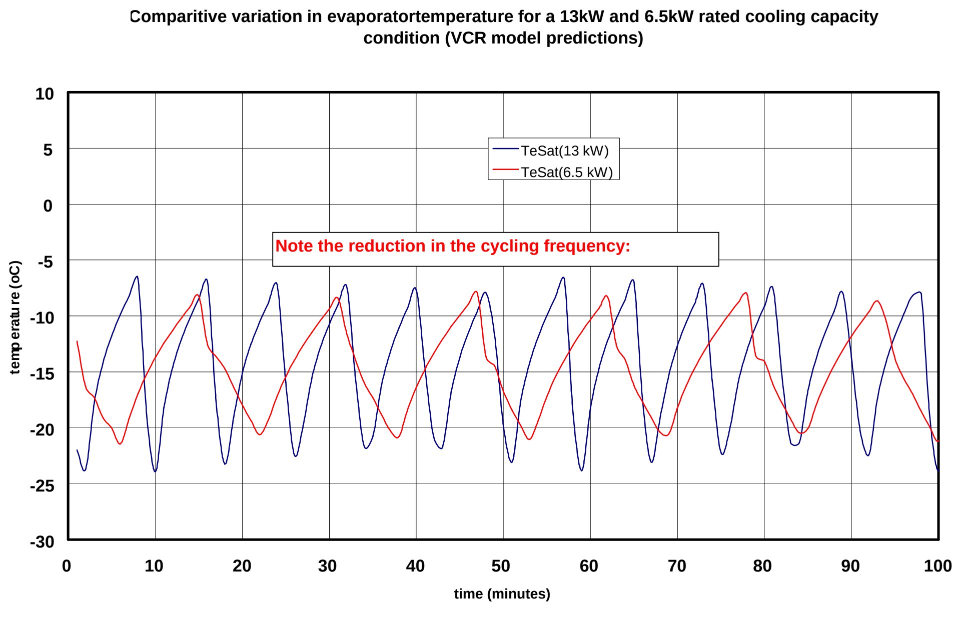

Figure 19

Appendix A
Compressor installation technical data and analysis

The following provides some technical details and analysis for the PR1500/0138 compressor. Original data provided by Dr Guy Hundy and Copeland Compressors Ltd

Compressor manufacturers data, courtesy of Copeland Ltd

Compressor model:	PR1500/0138
Copeland compressor model input data:	
Motor speed:	50Hz
Required refrigeration capacity:	12kW
Refrigerant:	R22
Suction return temperature:	20.0^{o}C
Liquid sub-cooling:	0.0K
Condenser temperature:	30.0^{o}Csat
Evaporator temperature:	-25^{o}Csat

Predicted performance at the above conditions:

Refrigeration capacity:	13.19 kW
Electrical power consumption:	5.83 kW
Current at 400VAC:	13.32A

Mechanical data:

Displacement at 50Hz:	49.9 m^3/h (103.19m3/h based on 3 cylinders with bore and stroke given below
Optional capacity steps:	33.3%
Number of cylinders:	3
Bore:	61.9(125)mm
Stroke:	63.5mm
Gross weight:	176kg
Max high pressure:	25bar
Suction inlet diameter:	1 5/8 inch
Discharge diameter:	1 1/8 inch
Max standstill pressure:	20.5bar

Optional accessories:

- Crankcase heater
- Additional fans
- Oil pressure switch
- Discharge temperature protection
- Start un-loader

The following performance curves, shown in Figures A1 and A2, were drawn using tabulated data provided by Copeland Compressors Ltd.

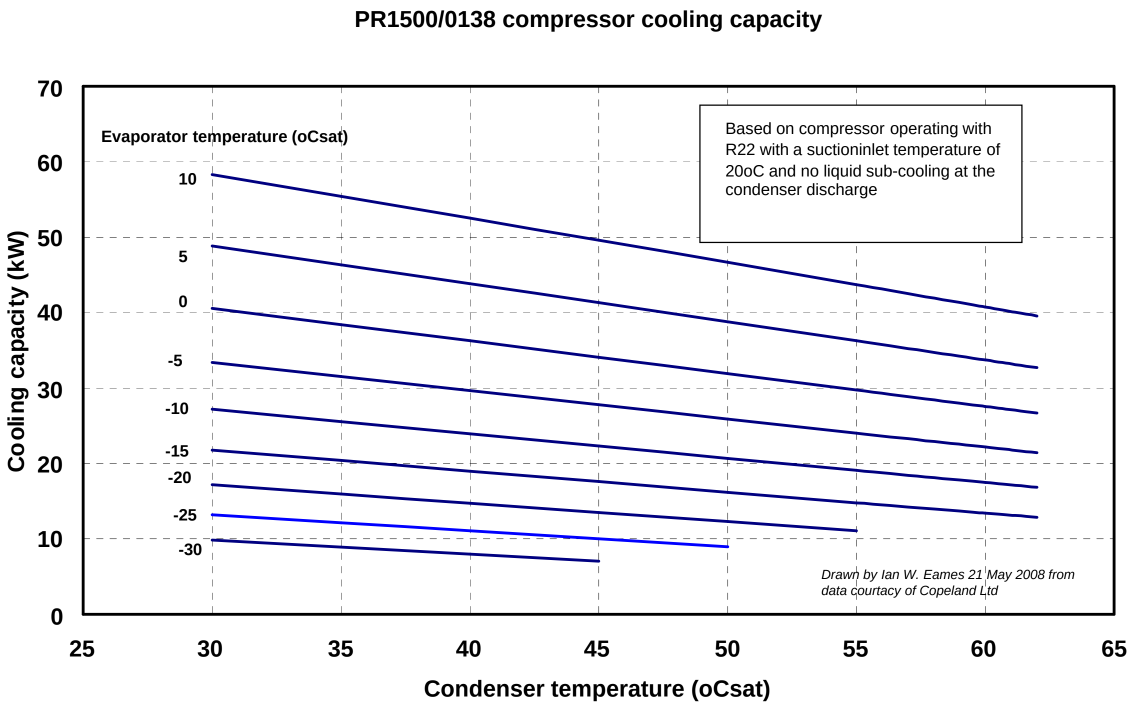

Figure A1

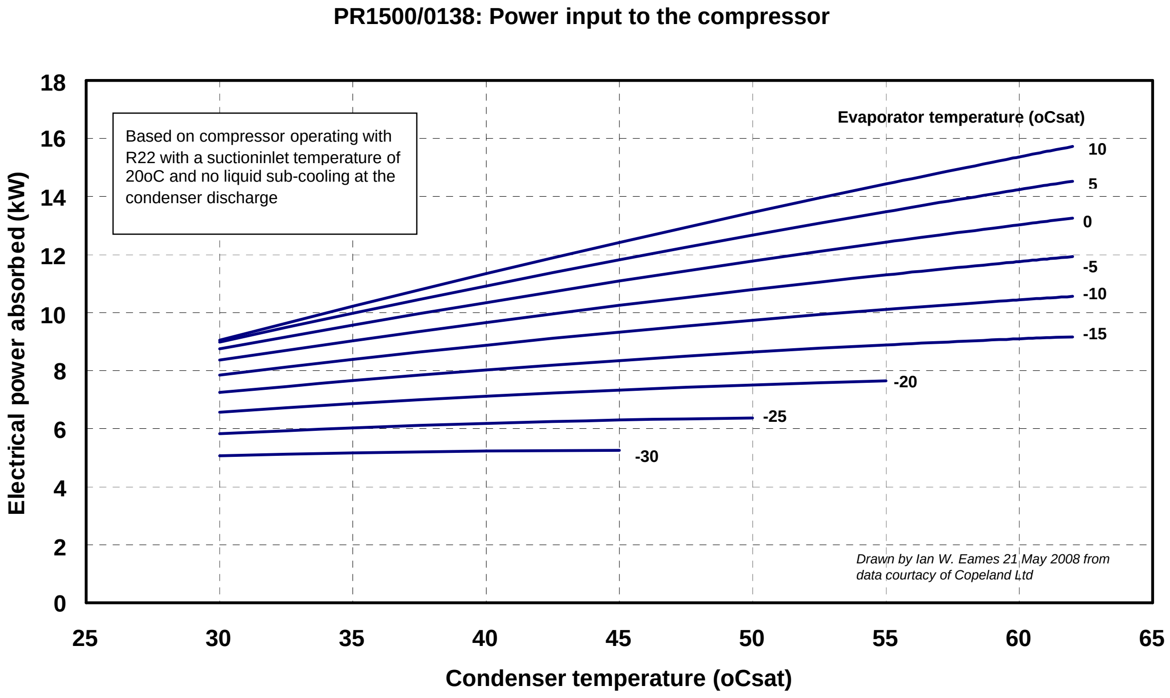

Figure A2

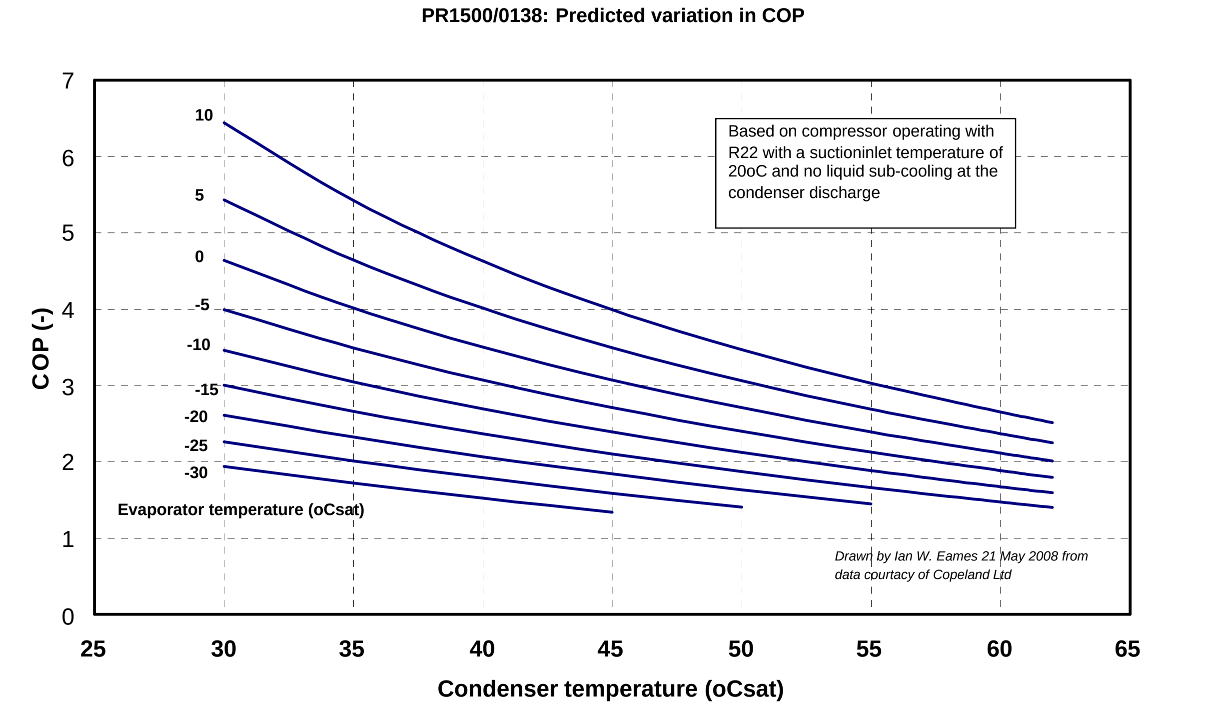

Figure A3

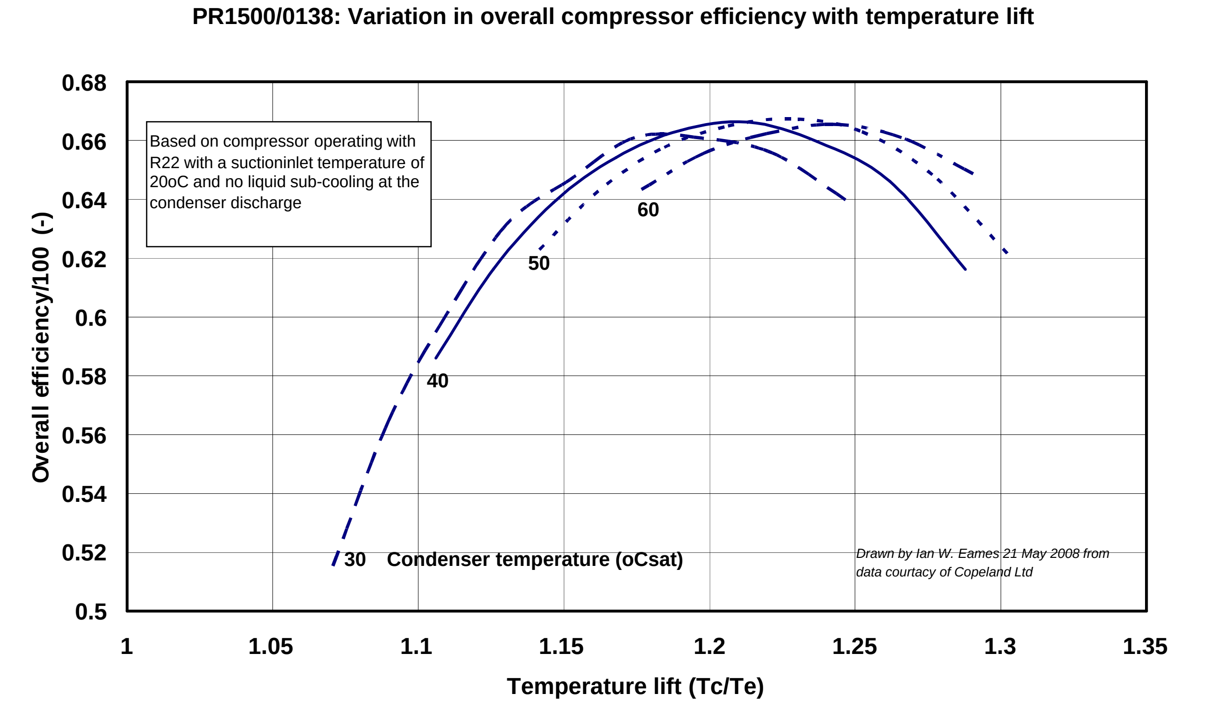

Figure A4