Project Report 2

REFRIGERATION ENERGY USE IN THE FOOD CHAIN

PROJECT REFERENCE (TBA)

CONTRACT REFERENCE (TBA)

by

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Summary

This report summarises the research so far carried out by the LSBU team into the 'Food Chain Refrigeration Energy Use ' project during months 3 to 6. The report describes the results of a literature investigation into modelling refrigeration systems and outlines some of the LSBU team's work on component and system modelling. The IoR System Efficiency Index is also described and the results of an investigation into seven refrigerator system applications are provided.

20th November 2006

Contents

1. Introduction

A project objective for LSBU is to develop steady state refrigeration system models to test (and optimise) various refrigeration systems used in the industrial production chain for food.

This report describes the results of a literature investigation into modelling refrigeration systems. The report then goes on to outline some of the LSBU team's work on component and system modelling. The IoR System Efficiency Index is also described and the results of an investigation into refrigerator system applications are provided.

2 A Review of Literature on Dynamic Models of vapour compression equipment

The final objective of the work is to produce a complete transient performance simulator of a centrifugal liquid chiller. The process of characterising the operation vapour compression refrigeration cycle can be done through different time regimes. The selection time regime would be non-steady state.

2.1 Transient modelling:

During transient simulation, each component in the system will experience its phenomena according to the non-uniformity of conditions within them. The previous work showed that the transient modelling was done by coupling a set of space-time partial differential equations in mass, energy and momentum balances used to manage ordinary differential & algebraic equations through reasonable assumptions.

Due to that our objective doesn't concentrate on a specific component only (i.e. compressor capacity control), so it's been decided to use for modelling a general energy equation that would capture the transient changes required and also be reasonable with CPU time execution.

2.2 Transients time scale:

The transient time scale will be selected according to the time constants of the responses and their causes. There are two main time scales:

2.2.1 Large time scale transients:

Large scale is selected for responses, which are scaled on the same order of magnitude like the total cycle time. The transients are caused by:
1- Load changes.
2- Start-up
3- Shutdown
4- Feedback control

2.2.2 Small time scale transients:

The responses are scaled on a very small time scale. The transients selected for that scale are caused by random fluctuations in conditions (i.e. compressor valve dynamics).
The literature review showed that there are three different concepts that were used for modelling:

2.3 Phase-dependency

1- Phase-dependent moving boundary:

The heat exchanger is divided into sections of variable volume depending on the refrigerant state (i.e. liquid, two-phase or superheated). The section volumes can't be constant during transient operation because of the constant mass redistribution. Therefore, the boundaries between adjacent volumes are tracked while`moving´ within the heat exchanger.

If Phase-dependent technique is applied for condensers, there will be two boundaries. The first boundary is between superheated vapour and the two-phase regions and the second is between the two-phase regions and the sub-cooled regions. It's important to consider the phase boundary moving in or out of the heat exchanger during large time scale transients (system start-up, shutdown).

2- Phase-independent finite difference:

The heat exchanger is modelled regardless the refrigerant phase. Modelling is expressed by using conservation equations, which are approximated through finite difference scheme by dividing the heat exchanger into a number of possibly constant elements (volumes). Each element is defined with its own state properties. The formulas used for the elements are 'phase-independent', so it's identical in the three phases.

3- Combined moving boundary with finite differencing:

The heat exchanger is modelled by dividing each phase into elements (volumes changing with the time) through finite differencing method. No elements are allowed to span their boundaries.

2.4 Lumped or distributed parameter method

1- Lumped method:

It's a simple computational method because it results in a finite system of equations consisting of algebraic and 1st order, ordinary differential equations. The disadvantage of is that averaging the state parameters over the complete control volume loses spatial detail.

2- Distributed method:

This method enables spatial detail & variations to be monitored. The governing equations for all the elements are identical in case the element properties are framed to be phase-independent.

2.5 Two phase flow as homogenous or slip-flow model

1- Homogenous model:

Through homogenous model, the liquid and vapour phases are considered to be in thermal equilibrium and moving at the same velocity.

2- Slip-flow model:

Through slip-flow model, the liquid and vapour are expected not to be in thermal equilibrium at any section, while the velocities are different and the model is dependent on the flow nature (i.e. bubbly, slug, annular).

2.6 Compressor considerations:

1- Effect of the entrained liquid droplets in suction vapour.

2- Thermal capacity and heat transfer effect to the surrounding

3- Valve losses were optimised by considering refrigerant flow through it.
4- Effect of the oil absorbing at start-up operation.

2.7 Combined intercooler & liquid sub-cooler

The modelling technique used the energy & mass balances to represent four different zones:

a) Vapour bubbles below the surface of the liquid.
b) Liquid Refrigerant inside the vessel
c) Sub-cooling coil.
d) Vapour space above the surface of the liquid

2.8 Pipe connections:

Most papers neglected the pressure drop occurring through the pipe connections, although the pipes of industrial refrigeration systems contain more refrigerant than the components which its influence should be considered. James et al considered pressure drop in pipes by: $\Delta p = Cm^2/ \rho$

2.9 Condenser & Evaporator considerations:

The heat exchangers are considered to be the most important and influential component in the refrigeration system due to its slowness and huge impact on the system's transients performance. The Tables 1 to 5 summarize the heat exchanger considerations.

The abbreviations used for the heat exchanger review are listed below:

PDMB:	Phase-dependent moving boundary
PIFD:	Phase-independent finite difference
CMBFD:	Combined moving boundary and finite difference

2.10 Suction-liquid line heat exchanger:

Main function is to implement heat recovery by:

1) Cooling down the refrigerant liquid that leaves the condenser.
2) Heating up the vapour leaving the evaporator.

If start-up simulation & diagnostic routines are required to be considered, then the probability of that the system would suffer from refrigerant shortage is required, where the vapour could enter with the liquid.

2.11 Expansion valves:

Main functions are:

1) Maintaining the pressure difference created by the compressor.
2) Refrigerant flow control according to the system demand.

The expansion valve types are:

5-a) Constant pressure expansion

Orifice equation was used for modelling.

5-b) Hand expansion valve

Refrigerant mass flow rate is expressed by a control equation that allows liquid level in the separator to increase as it deviates from set point. The valve strategy is to maintain constant level.

5-c) Thermostatic expansion valve (TEV):

The refrigerant mass flow rate was expressed through the following concepts:

1- Deriving an equation for particular TEV using the valve capacity & orifice equation.
2- A differential equation relating the suction superheat directly to the mass flow rate.
3- Deriving an equation for forces acting on the valve diaphragm.

The superheat was considered through the following concepts:

1- Modelling the bulb to sense the temperature of the superheated vapour.
2- Considering the superheat signal in terms of pressure & temperature by assuming a linear relationship between the valve opening & superheat pressure.

5-d) Motorised expansion valve:

Valve controller was expressed by PI controller that maintains a constant superheat at the evaporator outlet.

5-e) Liquid sensing expansion valve (LSEV):

The valve control is expressed by sensing the liquid amount in the superheated vapour as an input signal to expansion valve. The disadvantage of using the superheat to control the expansion valve is that at low superheat conditions and changing process, poor control happens. The advantages are that there won't be need for a high superheat and the systems setting isn't required to be reset at conditions change. The simulations showed LSEV to be more stable operation over TEV, which was produced hunting.

Through the last 24 years [1978 – 2002], the interest in dynamic performance of vapour compression systems have increased. Recently, there is growing interest in liquid chiller systems than air-to-air systems. There is a need to develop dynamic

system model of vapour compression centrifugal liquid chiller that includes large and small scale transients for heat exchangers because none of the reviewed models have modelled it.
The success of the proposed transient program on:

1) A standard user friendly interface to input components data.
2) Avoid as possible the concept used in most previous models for "Custom Built" modelling to have a general program.

3. System component models

This section summarize some of the system component models, developed by the LSBU team, which will form part of the dynamic model for refrigeration systems.

3.1 Evaporative condensers

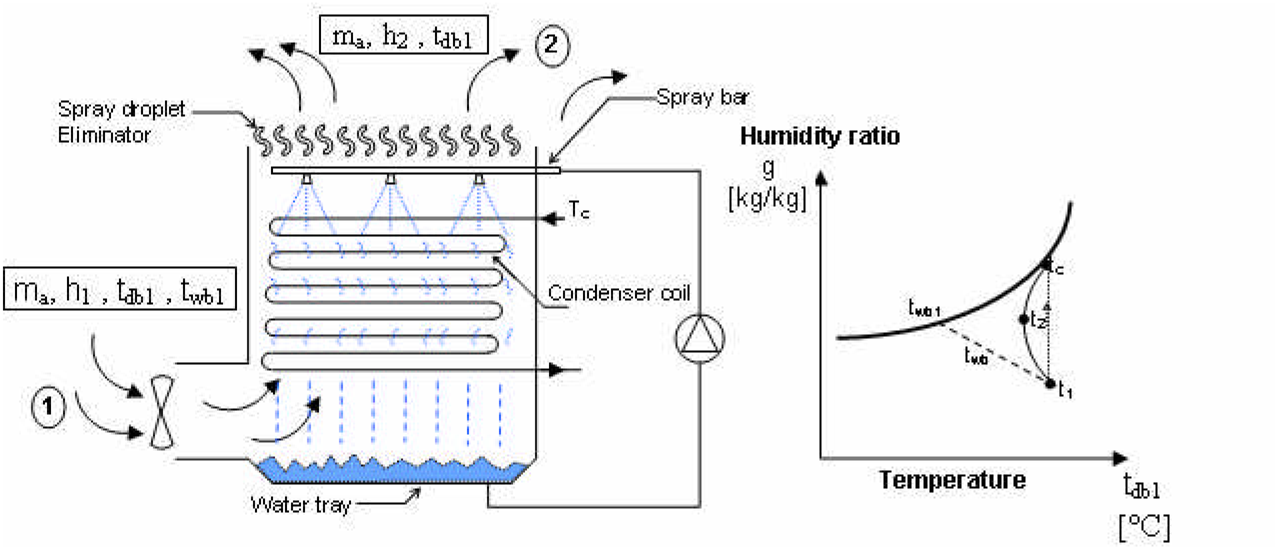


Figure 1 shows the general arrangement of a typical evaporative-air condenser. By definition, (as with all heat exchangers), the effectiveness of the condenser is,

$$\varepsilon = \frac{\text{Actual heat transfer}}{\text{Theoretical maximum heat transfer}} = \frac{Q_{Actual}}{Q_{Max}}$$

Actual heat transfer, $Q_{Actual} = m_a (h_2 - h_1)$

Maximum heat transfer will occur if the air leaving the condenser at (2) was saturated with water vapour at a temperature equal to that of the saturated refrigerant in the condenser coil, T_c.

Thus, $Q_{Max} = m_a (h_2' - h_1)$

Where h_2' = Enthalpy of saturated moist air at $T_{db} = T_c$

Therefore, $\varepsilon = \dfrac{h_2 - h_1}{h_2' - h_1} = \dfrac{Q_{Actual}}{m_a (h_2' - h_1)}$

Therefore, $h_2' = h_1 + \frac{Q_{Actual}}{\varepsilon \, m_a}$

The condensation temperature within the condenser (t_c), can be determined using the following correlations:

$T_c = -6.3\text{x}10^{-5}\, h_2'^3 + 4.64\text{x}10^{-3}\, h_2'^2 + 0.32496\, h_2' - 2.12955$ for h_2' (5.991 to 57.544)
$T_c = 3.9\text{x}10^{-6}\, h_2'^3 - 2.04\text{x}10^{-3}\, h_2'^2 + 0.48397\, h_2' - 1.81968$ for h_2' (57.545 to 166.615)
$T_c = 4\text{x}10^{-7}\, h_2'^3 - 4.9\text{x}10^{-4}\, h_2'^2 + 0.249\, h_2' + 10.267$ for h_2' (166.616 to 460.536)
$T_c = -1.45\text{x}10^{-5}\, h_2'^2 + 0.047563\, h_2' + 41.17353$ for h_2' (460.537 to 1539.414)

The above correlations give condenser temperatures in the range 0 °C to 80 °C with accuracy of +/- 0.2 °C. Note, t_c is given in °C and h must be in kJ/kg.

Enthalpy of the inlet air stream maybe calculated with sufficient accuracy using,

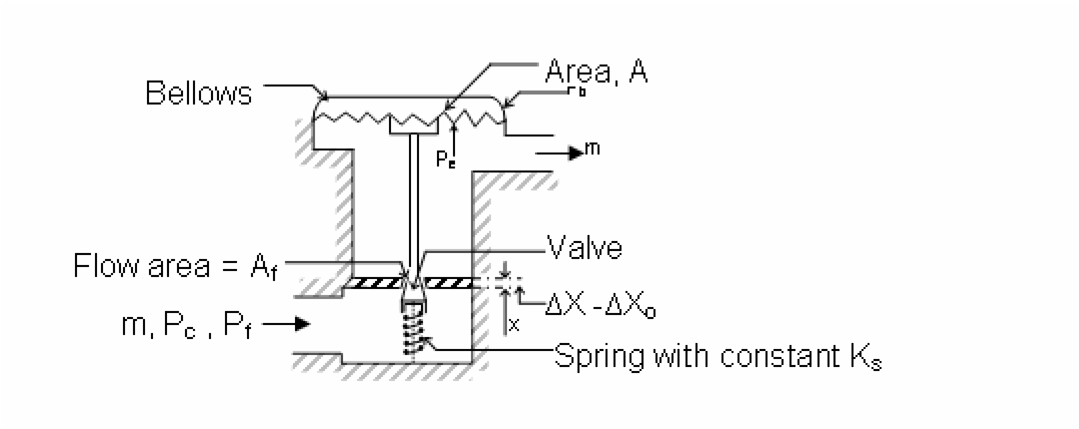
$h_1 = C_{p\,a} @ t_{db,1} + g_1 (C_{p\ w,tdb,1} + 2501)$
Where $C_{p\,a}$ = 1.007 kJ/kg.K and $C_{p\,w}$ = 1.83 kJ/kg.K

$$g_1 = 0.622 \frac{P_{w/s}}{101325} \times \frac{RH\,[\%]}{100}$$

where $P_{w/s}$ = Saturated air partial pressure at $t = t_{db}$
$P_{w/s}$ can be calculated using the following correlation[1]

$$\text{Log}_{10}P_{w/s} = 28.59051 - 8.2\,\text{Log}_{10}(t + 273.16) + 0.0024804\,(t+273.16) - \frac{3142.31}{(t+273.16)}$$

3.2 Thermostatic expansion valves



Based on the above schematic representation, the refrigerant flow through the valve may be calculated at design and part-load conditions as,

$$m = \beta \{(P_b - P_e) - (P_b - P_e)_o\} \sqrt{2\, \rho_f\, (P_c - P_e)}$$

where, $\beta = (P_b - P_e)_o$

The value of β can be determined from manufacturers data sheets or, if a design-point flow, $(P_b - P_e)$, $\rho_f (P_c - P_e)$ are known, the value of β can also be calculated. This will therefore allow the changes in refrigerator performance due to TEV operation to be modelled.

4. IoR System Efficiency Index

This section describes the work done by the LSBU team on the Institute of Refrigeration' System Efficiency Index. Results from this work will feed into the present study.

4.1 Some definitions

System Efficiency Index (SEI):

This is the ratio between the Coefficient of System Performance and the Theoretical Maximum Coefficient of Performance for a system operating between the same Process Fluid Supply Temperature and Ambient Dry-Bulb Temperature, at the Rating Condition..

Coefficient of System Performance (COSP):

This is the ratio between the Total Refrigeration Cooling Duty and the total electrical power supplied to the system, including power to compressors, fans, pumps and controls, cooling towers, but should exclude any input required intermittently, such as for defrost heaters.

Coefficient of Performance (COP):

This is the ratio between the Total Refrigeration Cooling Duty and the total electrical power to the compressors only.

4.2Defining equations

For single-cooling temperature systems

$$SEI = \frac{COSP}{\left(\frac{T_{eo}}{T_{amb,db} - T_{eo}}\right)}$$

Where, T_{eo} = evaporator process fluid supply temperature (K)
$T_{amb,db}$ = outdoor ambient dry-bulb temperature (K)

For muli-cooling temperature systems

$$SEI = \frac{COSP}{Q_{LT}\frac{(T_{amb,db} - T_{eLT})}{T_{eoLT}} + Q_{IT}\frac{(T_{amb,db} - T_{eIT})}{T_{eoIT}} + Q_{HT}\frac{(T_{amb,db} - T_{eHT})}{T_{eoHT}}}$$

Where,
T_{eoLT} = temperature of the fluid that cools the product for LT part of system (K)
T_{eoIT} = temperature of the fluid that cools the product for IT part of system (K)
T_{eoHT} = temperature of the fluid that cools the product for HT part of system (K)

4.3 Rating models

To establish a range of performance classes for refrigeration systems, it was necessary to produce a mathematical model for each classification. The models are based on the Carnot refrigeration cycle COP value, which is scaled according to Equation (1).

$$COSP = \frac{\eta_c \eta_s}{APF}\left[\frac{Te}{Tc - Te}\right] \qquad \textbf{[1]}$$

Where, η_c = compressor isentropic efficiency
η_s = cycle irreversibility factor

$$T_e = T_{ei} - \left(\frac{1}{\varepsilon_e}\right)(T_{ei} - T_{eo})$$

$$T_c = T_{ci} + \left(\frac{1}{\varepsilon_c}\right)(T_{co} - T_{ci})$$

ε_c = compressor isentropic efficiency
ε_s = cycle irreversibility factor

$$APF = \text{ancillary power factor}$$
$$= \frac{\text{compressor electrical power input}}{\text{total electrical power input}}$$

Realistic values for each of the above parameters were determined from manufacturers' catalogues. The following Figures 1 to 4 summarize some of the findings.

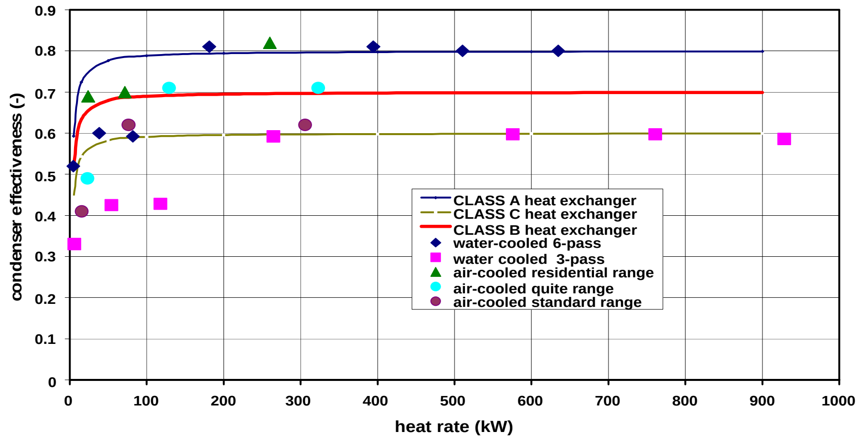


Figure 1: Showing the variation in effectiveness values for a range of proprietary air and water cooled condensers compared with the model of the index effectiveness value for the 3 classifications of heat exchanger; A, B and C.

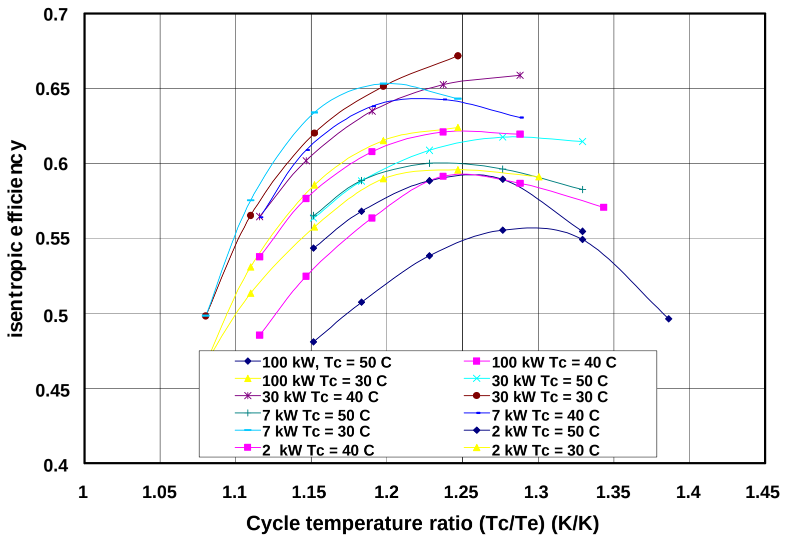


Figure 2: Showing typical isentropic efficiency values for a range of compressor sizes

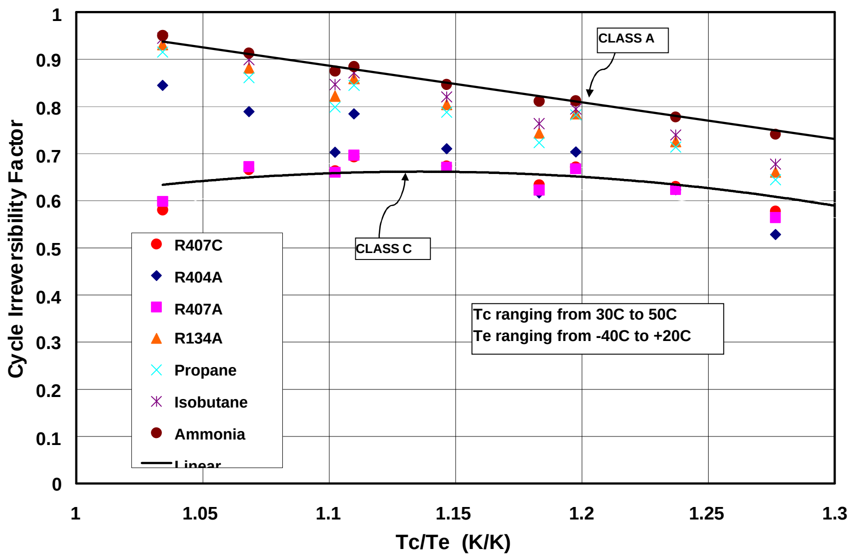


Figure 3: Variation in □c with cycle TLR, (Tc/Te)

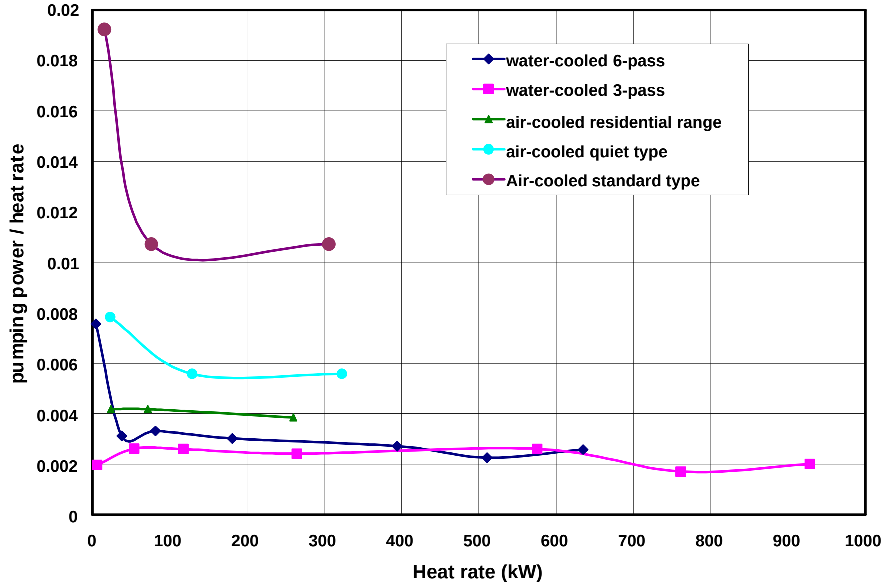


Figure 4: Showing the variation in the pump or fan power as a proportion of condenser heat rate for a number of proprietary water and air cooled condensers

Using best and worst current practice data from that derived from manufacturers' catalogues, the following five classifications were derived.

A - Best possible efficiency using current best design practice

B - A good efficiency

C - An acceptable to good efficiency

D - A low efficiency

E - A very low efficiency

The next section describes some application which have been used to test the SEI and Rating Classification

5. System Applications

During the period to which this report refers the following case studies have been examined.

CASE STUDY 1 - A COLD STORE APPLICATION

General description of refrigeration system.

A cold-store application using a 2-stage compression ammonia system with a dry-air cooled condenser and air cooling evaporators. Compression provided by 2 x LP screw compressors and 1 x HP screw compressor. The system is being designed.

Input data:

	Design Point
Cooling duty	1182 kW
Condenser cooling duty	1689 kW
Compressor power	507 kW
Total input electrical power	591.9
Tei	-26^{o}C
Teo	-30.31^{o}C
Tci	21^{o}C
Condenser air flow	335 m^{3}/s

Results:

Class / Factor	A	B	C	D	E	System results
SEI	0.421	0.34	0.247	0.199	0.171	0.421
CoSP	1.99	1.61	1.17	0.94	0.81	1.99
COP	2.59	2.42	1.99	1.79	1.62	2.33

This system has an 'A' Rating

CASE STUDY 2 - AN AIR CONDITIONING APPLICATION

General description of refrigeration system.

An air conditioning application using a R407C water chiller (single - pass evaporator), with multiple fan coil units, a dry air cooled condenser with an economised screw compressor

Input data:

	Design	Normal operation
Cooling duty	180 kW	120 kW
Condenser cooling duty	260 kW	153 kW
Compressor power	80 kW	33 kW
Total electric power input	112 kW	55 kW
Tei	25	23
Teo	22	21
Tci	35	25
Condenser air flow	51.64 m^{3}/s	30.39 m^{3}/s

Results:

Design

Class / Factor	A	B	C	D	E	System results
SEI	0.416	0.318	0.149	0.113	0.093	0.07
CoSP	9.44	7.24	3.38	2.58	2.12	1.6
COP	12.27	10.86	5.76	4.91	4.25	2.25

This system has an unclassified Rating

Nominal operation

Class / Factor	A	B	C	D	E	System results
SEI	0.462	0.31	0.076	0.055	0.044	0.029
CoSP	33.99	22.83	5.59	4.06	3.24	2.18
COP	44.19	34.25	9.51	7.73	6.48	3.63

This system has also unclassified Rating at nominal conditions

CASE STUDY 3 - A COLD ROOM APPLICATION

General description of refrigeration system.

Cold Room Scroll R404A with remote dry-air condensing unit serving DX fan coil units ,

Input data:

	Design	Normal operation
Cooling duty	3.3 kW	4.37 kW
Condenser cooling duty	5.14 kW	5.75 kW
Compressor power	1.84 kW	1.38 kW
Total electrical power input	2.018	1.558
Tei	5 ^oC	5 ^oC
Teo	0 ^oC	0 ^oC
Tci	32	13
Condenser air flow	0.85 m^3/s	0.95 m^3/s

Results:

Design

Class / Factor	A	B	C	D	E	System results
SEI	0.322	0.256	0.166	0.128	0.107	0.191
CoSP	2.57	2.18	1.42	1.09	0.92	1.63
COP	3.57	3.27	2.41	2.08	1.84	1.79

This system has 'C' Rating at design

Nominal operation

Class / Factor	A	B	C	D	E	System results
SEI	0.213	0.166	0.11	0.084	0.069	0.133
CoSP	4.48	3.49	2.32	1.76	1.46	2.8
COP	5.83	5.24	3.95	3.35	2.92	3.16

This system also has a 'C' Rating at nominal conditions

CASE STUDY 4 - A SUPERMARKET 'CHILL-PACK' APPLICATION

General description of refrigeration system: Supermarket Chill Pack application using 8-10 scroll compressors and dry-air cooled condenser with typically 6 fan stages – i..e. 0-6 fans on. The number of compressors running is controlled by suction pressure with set-point and number of fans running is controlled by set-point. A set-point of 20°C for the condenser and –11°C for evaporating. The condensing set-point achieved for 60% of time. Temperature rises up to c. 45°C for rest of time – high load and high ambient temperature times. The evaporating temperature set-point optimised up to c. –6°C if load conditions allow – i.e. at low load conditions. Load conditions are determined by whether all cabinets are achieving their set-points. Compressors always run close to "fully loaded". If there is too much capacity compressor is switched off.

Input data:

	Design	Normal operation
Cooling duty	90 kW	35 kW
Condenser cooling duty	120 kW	53 kW
Compressor power	30 kW	18 kW
Total electric power input	42 kW	30 kW
Tei	5°C	5°C
Teo	-7°C	-4°C
Tci	30°C	15°C
Tco	35°C	22°C
Condenser air flow	19.86 m^3/s	6.2 m^3/s

Results:

Design

Class / Factor	A	B	C	D	E	System results
SEI	0.407	0.322	0.212	0.162	0.134	0.297
CoSP	2.93	2.31	1.52	1.16	0.92	2.14
COP	3.81	3.47	2.59	2.21	1.94	3

This system has 'C' Rating at design

Nominal operation

Class / Factor	A	B	C	D	E	System results
SEI	0.29	0.227	0.154	0.117	0.097	0.082
CoSP	4.11	3.22	2.18	1.66	1.37	1.16
COP	5.34	4.83	3.72	3.15	2.75	1.94

This system has an unclassified rating at nominal conditions

CASE STUDY 5 - A SUPERMARKET 'FROZEN PACK' APPLICATION

General description of refrigeration system: Supermarket Refrigeration Frozen Pack application with typically 8-10 scroll compressors and air cooled condenser with typically 6 fan stages – i.e 0-6 fans on. Number of compressors running controlled by suction pressure with set-points controlling the number of fan running. Set-point values of 20°C for the condenser and –30°C for evaporator. Condensing set-point is achieved for 60% of the time. Temperature rises up to c. 45 °C for the rest of the time – high load and high ambient temperature times. Evaporating temperature set-point optimised up to c. –24°C if load conditions allow – i.e. at low load conditions. Load conditions are determined by whether all cabinets are achieving their set-point values. Compressors always run close to “fully loaded”. If there is too much capacity compressor is switched off.

Input data:

	Design	Normal operation
Cooling duty	30 kW	22 kW
Condenser cooling duty	60 kW	44 kW
Compressor power	30 kW	22 kW
Total electric power input	50.5 kW	42.5 kW
Tei	-20°C	-20°C
Teo	-26°C	-24°C
Tci	30°C	15°C
Tco	35°C	22°C
Condenser air flow	9.93 m3/s	5.2 m3/s

Results:
Design

Class / Factor	A	B	C	D	E	System results
SEI	0.415	0.332	0.224	0.175	0.148	0.134
CoSP	1.83	1.46	0.99	0.77	0.65	0.59
COP	2.38	2.2	1.68	1.47	1.3	1

This system has an unclassified rating at design

Nominal operation

Class / Factor	A	B	C	D	E	System results
SEI	0.371	0.299	0.219	0.174	0.149	0.081
CoSP	2.37	1.91	1.4	1.11	0.95	0.51
COP	3.08	2.86	2.38	2.12	1.91	1

This system also has an unclassified rating at nominal conditions

CASE STUDY 6 - A REFRIGERATED TRANSPORT CONTAINER APPLICATION

General description of refrigeration system: A refrigerated transport application using a single-stage reciprocating compressor. Refrigerant used was R134a. The condenser is air-cooled condenser and the evaporator cools air. The performance data is believed to be typical for this application. The two data sets are taken from capacity test at normal-rating conditions and extreme conditions.

Input data:

	Design	Normal operation
Cooling duty	2.894 kW	4.762 kW
Compressor power	5.55 kW	5.04 kW
Condenser cooling duty	8.444 kW	9.802 kW
Total electric power input	6.395 kW	5.59 kW
Te (sat)	-32°C	-28°C
Tei	-24.3°C	-17.4°C
Teo	-25.4°C	-20.7°C
Tc	57.5°C	48°C
Tci	49.5°C	37.7°C
Tco	56.8°C	46.4°C
Condenser air flow	0.957	0.932

Results:
Design

Class / Factor	A	B	C	D	E	System results
SEI	0.318	0.249	0.13	0.095	0.076	0.136
CoSP	1.05	0.82	0.43	0.31	0.25	0.45
COP	1.37	1.23	0.73	0.6	0.5	0.52

This system has 'C' Rating at design

Nominal operation

Class / Factor	A	B	C	D	E	System results
SEI	0.343	0.271	0.163	0.121	0.098	0.197
CoSP	1.48	1.17	0.7	0.52	0.42	0.85
COP	1.93	1.75	1.2	0.99	0.85	0.94

This system also has a 'C' Rating at nominal conditions

CASE STUDY 7 - A COLD STORE REFRIGERATION SYSTEM

General description of refrigeration system: A pump circulated ammonia system with 2 compressors, 1 condenser and 4 coolers. The condenser is cooled by an evaporative-air cooled condenser.

Input data:

	Normal operation
Cooling duty	309 kW
Compressor power	148.8
Tei	-20
Teo	-22
Condenser coolant flow	9.9 m^3/s
Tci	22
TambDB	32
TamWB	22
Total power	182

Results:

Class / Factor	A	B	C	D	E	System results
SEI	0.386	0.307	0.189	0.144	0.119	0.365
CoSP	1.79	1.42	0.88	0.67	0.55	1.69
COP	2.33	2.14	1.49	1.27	1.11	2.07

The systems has a 'B' Rating

5.1 Comparison of SEI with EUROVENT certification

The Eurovent/Cecomaf organisation represents 150 manufactures of air handling, air conditioning and refrigeration equipment. One of its tasks is to develop certification programmes and since 1994 it the organisation has issued directories of certified equipment. At this time there are 13 ongoing programmes. The object of certification is to independently test equipment performance measured in terms of cooling capacity and electrical energy input at full load against manufacturers' catalogue data at predetermined flow conditions. If the performance of the equipment equals or is better than quoted by the manufacturer then the equipment receives certification. If the equipment does not achieve the quoted performance then the equipment catalogue data must be amended to reflect before certification can be provided. Up to recently Eurovent was concerned only that manufactures catalogue data reflected the true performance of a piece of equipment. Since 2002 Eurovent has taken an interest in the energy efficiency of equipment.

The Eurovent scheme for the energy classification of equipment follows the 'A', 'B', 'C' … rating system used for domestic electrical goods, such as small room air conditioners (<12kW), refrigerators and other 'white' goods. For air conditioning chillers the rating system given in the following table has been adopted:

Chiller Energy Classification[1]

EER Class	**Air Cooled**	**Water Cooled**	**Remote Condenser**
A	EER => 3.1	EER => 5.05	EER => 3.55

[1] Yasmina Saheb, Sulejman Becirspahic and Jerome Simon, Effects of the Certification of Chillers Energy Efficiency, IEECB Conference, Frankfurt, April 2006

B	2.9<=EER<3.1	4.65<=EER<5.05	3.4<=EER<3.55
C	2.7<=EER<2.9	4.25<=EER<4.65	3.25<=EER<3.4
D	2.5<=EER<2.7	3.85<=EER<4.25	3.10<=EER<3.25
E	2.3<=EER<2.5	3.45<=EER<3.85	2.95<=EER<3.10
F	2.1<=EER<2.3	3.05<=EER<3.45	2.8<=EER<2.95
G	EER <2.1	EER <3.05	EER <2.8

The data in the above table relates to the performance of chillers at their nominal (standard) rating or test conditions listed in the Eurovent directory[2], with reference to test standard pr EN 12055. The above table of classifications does not appear to take account of the different certification temperature conditions, which appears to be a weakness of this system.

The following figure shows EER data a number of Eurovent Class A to C chillers. It is interesting to note that the performance data in every case used in this figure were measured at the same inlet and outlet temperatures from the evaporator and condenser.

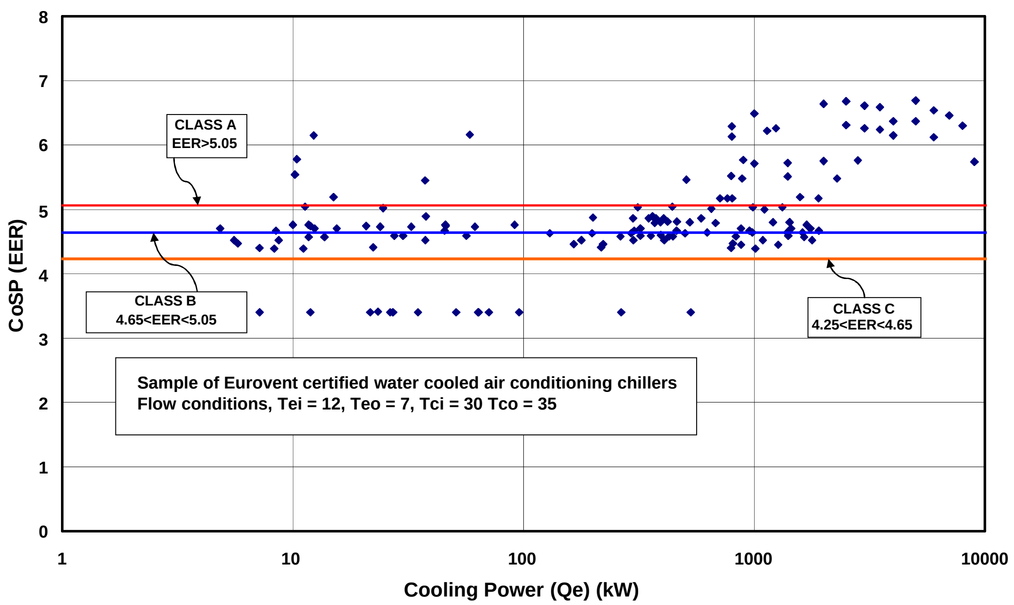


Figure 7: Showing a range of Eurovent certified water-cooled air conditioning chiller data at the flow temperatures given in the diagram. EER is defined by Eurovent as the ratio of total cooling capacity and total electrical energy input to compressor and ancillary equipment. This is equivalent to the IoR's CoSP figure of merit.

The following Table 6 lists the standard certification temperatures for the range of liquid chiller (cooling) applications taken from the Eurovent directory[1]. Included in this table is the IoR's SEI CoSP classification.

[2] Eurovent Directory of Certified Products 1st Feb 02 - 31st Jan 03, chap 6, page 356

Table 6: Listing some comparative results between SEI and Eurovent Classifications

APPLICATION	TYPE	EVAP Tei/Teo (°C)	CON Tci/Tco (°C)	A-Class SEI CoSP	B-Class SEI CoSP	C-Class SEI CoSP
Air conditioning	LCP/W Water/Water	12/7	30/35	4.78	3.83	2.97
	LCP/T without flash economiser	12/7	45-40	3.76	3.08	2.45
Medium Brine	LCP/W Water/Water	0/-5	30/35	3.46	2.85	2.28
	LCP/T without flash economiser	0/-5	45-40	2.81	2.33	1.89
Low Brine	LCP/W Water/Water	-10/-15	30/35	2.7	2.24	1.81
	LCP/T without flash economiser	-10/-15	45-40	2.19	1.82	1.48
Cooling Floor	LCP/W Water/Water	23/18	30/35	7.02	5.04	3.98
	LCP/W Brine/Brine or Brine/Water	23/18	30/35	7.02	5.04	3.98

6 Work programme

The LSBU team are currently working on a new reciprocating compressor model and cooling tower model for inclusion in the dynamic system model. Some of this work will be reported at the meeting. In addition sub-routines for component models are being written and tested.

Further applications data is being sought and component data is being collated from manufacturers' catalogues.

Once a specification for the software has been agreed, a sample front-end (user interface) will be produced, which will be passed to partners for their comments. It is hoped that this will be available for the next management group meeting.

7. Conclusions

This has report described the results of a literature investigation into modelling refrigeration systems and outlined some of the LSBU team's work on component and system modelling. The IoR System Efficiency Index was also described and the results of an investigation into refrigerator system applications were provided.

Author	Table 1: Heat exchanger modelling considerations		
	Phase dependency classification	**Lumped or distributed method**	**Two phase model**
Wedekind et al [1978]	**PDMB**	**Lumped**	**N/A**
	1- Adequate detail for two-phase flow by using lumped method without the need of handling the transient form by momentum equation		
Dhar & Soedel [1979]	**PDMB**	**Lumped**	**N/A**
	1- Coupled pair of lumps represent in the two-phase flows, the exchange of mass internally & heat externally. 2- All major transients were well captured for liquid chillers & air cooled condenser, using fairly simple representations		
Goldschmidt & Hart [1982]	**N/A**	**N/A**	**N/A**
	1- Exponential fit is used for modelling to steady state performance extracted from the manufacturer's data		
Chi & Didion's [1982]	**PDMB**	**Lumped**	**N/A**
	1- Momentum equation is used for handling transients. 2- All components dynamics are captured including air momentum across heat exchangers & start-up.		
Yasuda et al [1983]	**PDMB**	**Lumped**	**N/A**
	1- Small-scale transients were the objective to capture including feedback control and valve setting instability. 2- Broad assumption worked without serious consequence as constant sub-cooling & uniform two-phase condition in the condenser.		

<table>
<tr><th rowspan="2">Author</th><th colspan="3">Table 2 Heat exchanger modelling considerations</th></tr>
<tr><th>Phase dependency classification</th><th>Lumped or distributed method</th><th>Two phase model</th></tr>
<tr><td rowspan="2">MacArthur [1984a], MacArthur & Grald [1987], Rasmussen et al [1987]</td><td>PIFD</td><td>Distributed</td><td>Homogenous in condenser
Slip in Evaporator</td></tr>
<tr><td colspan="3">1- One-dimensional flow is assumed to simplify space-time dependent conservation equations, where discretisations were fully implicit allowing stable solutions up to 10 s time step.
2- [1984a] used simple version for heat exchanger formulation that resulted in inaccurate mass distribution predictions because the uniform flow velocities have de-coupled pressure response from thermal response.
3- [1987] improved modelling by coupling the mass balance to the energy balance to dictate the pressure response.</td></tr>
<tr><td rowspan="2">Murphy & Goldschmidt [1984, 1985]</td><td>N/A</td><td>N/A</td><td>N/A</td></tr>
<tr><td colspan="3">1- Start-up transients for condenser were captured, like the capillary tube dynamics & liquid backing into the condenser including refrigerant pressure response & tube material dynamics.
2- Shutdown process is predominated by refrigerant quality at the entry of the capillary tube, so liquid line is modelled in detail to capture refrigerant quality. Heat exchangers are modelled as tanks containing two-phase refrigerant at different pressure to begin.</td></tr>
</table>

<table>
<tr><th rowspan="2">Author</th><th colspan="3">Table 3 Heat exchanger modelling considerations</th></tr>
<tr><th>Phase dependency classification</th><th>Lumped or distributed method</th><th>Two phase model</th></tr>
<tr><td rowspan="2">Sami et al [1987]</td><td>PIFD</td><td>Distributed</td><td>N/A</td></tr>
<tr><td colspan="3">1- Multiple configurations were modelled including Shell & tube condenser & evaporator, air-cooled condenser, capillary tube & direct expansion evaporator.
2- Drift-flux model based on vapour & liquid phases is used for modelling while coupling mass and energy balances of each phase through evaporation & condensation.</td></tr>
<tr><td rowspan="2">Nyers & Stoyan's [1994]</td><td>CMBFD</td><td>N/A</td><td>N/A</td></tr>
<tr><td colspan="3">1- The model predicted the following: a) Evaporator's behaviour under step jump. B) Exponential saturation c) Periodic oscillation of the temperature d) Flow rate of the secondary fluid e) Condenser pressure</td></tr>
<tr><td rowspan="2">sami & Comeau [1992], Sami & Dahmani [1996]</td><td>PIFD</td><td>Distributed</td><td>N/A</td></tr>
<tr><td colspan="3">1- sami et al [1987] was expanded to predict system performance by including finite differencing within the drift-flux model.
2- [1996] work dealt with HFC alternatives to R22 as R407a, R507 & NARM502.</td></tr>
<tr><td rowspan="2">Xiandong He et al [1997]</td><td>PDMB</td><td>Lumped</td><td>N/A</td></tr>
<tr><td colspan="3">1- The objective was to study the effect of multivariable feedback control through introducing of MIMO (Multi-Input-Multi-Output) control method.</td></tr>
</table>

<table>
<tr><th rowspan="2">Author</th><th colspan="3">Table 4 Heat exchanger modelling considerations</th></tr>
<tr><th>Phase dependency classification</th><th>Lumped or distributed method</th><th>Two phase model</th></tr>
<tr><td rowspan="2">Williatzen [1998]</td><td>PDMB</td><td>Distributed</td><td>N/A</td></tr>
<tr><td colspan="3">1- The main objective was to study the phenomena of the appearance & disappearance of phases-regions.
2- The model structure allows any physical combination of phases (4 different phase's combinations) to be handled by an algorithm that switches between sets of equations.</td></tr>
<tr><td rowspan="2">Rossi & Braun [1999]</td><td>N/A</td><td>N/A</td><td>N/A</td></tr>
<tr><td colspan="3">1- First objective was studying start-up and on-off cycling by a smart automatic integration step sizing algorithm.
2- Second objective was emphasizing real time simulation using fully finite volume formulation of mass & heat balances.</td></tr>
<tr><td rowspan="2">Jing Xia et al [1999]</td><td>PDMB</td><td>N/A</td><td>N/A</td></tr>
<tr><td colspan="3">1- The objectives were studying the dynamics of condensing temperature and secondary fluid conditions.</td></tr>
<tr><td rowspan="2">Jakobsen et al [1999]</td><td>PDMB</td><td>N/A</td><td>Both considered</td></tr>
<tr><td colspan="3">1- The objective was comparing slip and homogenous flows where slip is proved to be more accurate.</td></tr>
<tr><td rowspan="2">Svensson [1999]</td><td>PDMB</td><td>lumped</td><td>N/A</td></tr>
<tr><td colspan="3">1- The objectives were studying the dynamics of load disturbances which is introduced by step changes in the condenser side water flow rates.</td></tr>
</table>

Author	Table 5 Heat exchanger modelling considerations		
	Phase dependency classification	**Lumped or distributed method**	**Two phase model**
Wang & Wang [2000]	**PDMB**	**Distributed**	**N/A**
	1- Highly simplified manner was used in modelling where the effectiveness is correlated to water flow rate & constructional parameters. 2- The dynamics is studied by introducing lumped thermal capacitances to refrigerant, heat exchanger body & secondary fluid.		
Browne & Bansal [2000]	**PDMB**	**lumped**	**N/A**
	1- First objective was comparing the simple dynamic model against dynamic neural network model of screw chillers. 2- The time scale selected was Quasi-Steady state.		
Grace & Tassou [2000]	**PIFD**	**Distributed**	**Homogenous in condenser** **Slip in Evaporator**
	1- Identical solution to MacArthur [1987] was used. 2- Expansion deice was modelled isenthalpic, where the sensing bulb is modelled in detail by considering all heat transfer resistances & capacitances.		