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Modelling thermostatic expansion valves

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ABSTRACT

This paper describes and evaluates the development of generalised steady state and transient mathematical models for thermostatic expansion valves (TEVs) of the types used in commercial refrigeration systems. The model is of a generalised nature, because it is not necessary to input performance or geometrical data for a particular valve to operate the model. However, if required, the models can provide an accurate correlation of valve manufacturer's data. Derivations are provided and validating data is presented. The mathematical models described in the paper form part of computer software that simulates the thermal operation of whole refrigeration systems. The software is titled Vapour Compression Refrigeration System (VCRS) simulator, and by way of example the paper presents results from the VCRS simulator which are used to aid a discussion of operating faults, such as hunting and under damping and their possible causes, where these can be attributed to the expansion valve.

Modélisation thermostatique des détendeurs

Mots clés : Système frigorifique à compression de vapeur ; Simulation ; Modélisation ; Détendeur thermostatique ; Logiciel

1. Introduction

The majority of commercial refrigeration systems, such as those used for food storage and processing, use a thermostatic expansion valve (TEV) to meter refrigerant flow between the condenser and evaporator in a way that causes the refrigerant to be superheated at the inlet to the compressor. In this way the compressor is protected from damage by liquid ingestion and the heat transfer performance of the evaporator is optimised. Recent research lead the authors of this paper to believe that computer models of refrigerator components, including TEVs, are more useful when they are non-specific; in other words where ever possible the need for empiric performance data or geometric information about a specific valve as input data to a model should be avoided.

This paper is concerned with modelling the steady state and transient performance of thermostatic expansion valves, (TEVs). Published literature on thermostatic expansion valve

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Nomenclature

A	area (m^2)
C_n	constants of proportionality
h	enthalpy (J kg^{-1})
MOP	maximum operating pressure (Pa)
$\dot{m}$	refrigerant mass flow (kg s^{-1})
P	pressure (Pa)
$\dot{Q}_e$	evaporator heat rate (J)
R	individual gas constant (J kg^{-1} K^{-1})
T	temperature (°C)
$\overline{MC}$	the thermal mass of the sensing-bulb (J K^{-1})
k_s	spring constant (N m^{-1})
U	overall heat transfer coefficient (W m^{-2} K^{-1})
z	compressibility factor
Greek	
α	pressure equivalent of static superheat setting (Pa)
β	valve flow area constant (m^2 Pa^{-1})
δ	value of $(P_b - P)_e$ when a valve is fully opened
τ	time constant (s)
ρ	density of saturated liquid (kg m^{-3})
Subscript	
b	bulb
c	condenser
d	valve diaphragm
e	evaporator
f	saturated liquid
g	saturated vapour
n	nominal
mop	maximum operating pressure
r	reserve
sat	saturated
sc	subcooling
sh	superheat
v	valve

modelling goes back several decades and includes work covering theoretical and experimental research of both steady state and transient operation. Often, studies focussed on modelling vapour compression refrigerators as a whole; with the TEV forming only a part of the formulation. An early example of transient refrigeration system modelling is provided by Chi and Didion (1982). Their model incorporated a steady state linear model for a TEV. It is not clear from the paper what input data was needed to run the Chi and Didion's (1982) TEV model. Experimental measurements and theoretical results are compared graphically for 6 min periods after start-up. Up to the 2 min point there is significant difference between the experimental data and model predictions for evaporator inlet temperature. This may have been due to the linear TEV assumption used in the formulation of the model. Also, measured data were taken at 30 s intervals making impossible to know what was happening to the system's temperatures and pressures during the first 30 s after start-up. However, after about 2 min of simulated time following start-up the predicted and measured values were shown to be close.

James and James (1987) described their transient model of a TEV. This was a particularly interesting study. However, in order to run the model geometric data of the valve and bulb was needed, which made it unsuitable for the purposes of a generalised refrigerator system model.

Conde and Sutera (1992) described and validated their mathematical model to simulate the steady-state operation of thermostatic expansion valves. The disadvantage of the Conde and Sutera's (1992) model was that it required an accurate description of the throttling section geometry. While the model described in this paper has similarities to that of Conde and Sutera's model it does not require geometric data as inputs, which makes it more flexible.

Mithrarartne and Wijeysundera (2002) carried out a theoretical and experimental study of a refrigerator system. They compared the results from both linear and a non-linear TEV models. Both models were steady state and empirically based, being curve fits of volumetric flow as a function of pressure differential, (Pb−Pe), and specific to the TEV valve used in the experiments. They found that their linear model was able to capture stable dynamic changes in evaporator conditions but not hunting, which was predicted only when their non-linear model was used. The Mithrarartne and Wijevsundera's model failed to predict the TEV flow characteristic when the suction-line vapour temperature, measured close to the sensing bulb, was above about 16 °C. On checking the specification for the type of TEV used in the experiments the authors of this paper found that it was a MOP (maximum operating pressure), or gas-charged type, which may explain their unexpected result. The operation of a MOP valve is described later in this paper.

Aprea and Mastrullo (2002) reported a comparative study between the performance of a TEV and electronic expansion valve (EEV) when replacing a system's refrigerant from R22 to R407C. Under steady state operation the results show the two types of valve produced similar results regardless of which refrigerant was used in the system tested. During compressor start-up though, when the pressures in the evaporator and condenser were equalised, the EEV's response was found to be faster than that of the TEV, but when the start-up was initiated with the condenser and evaporator pressures out of balance the opposite was found. It is not entirely certain why this reversal occurred, however, the authors of this paper believe that it may have been due to the type of EEV used in the tests, which relied entirely on temperature signals to initiate flow changes whereas a TEV uses both pressure and a temperature signals to meter the flow.

Yu et al. (2006) investigated how TEVs in air-cooled condenser systems can prevent COP values from rising as condenser air-on temperature falls. Their analysis showed that in order to maintain a desired refrigerant flow through a TEV it was necessary to maintain a high differential pressure across the valve. This was achieved by shutting-down or slowing variable speed condenser cooling fans as the ambient air temperature fell. In their analysis Yu et al. (2006) used a linear TEV model similar to that proposed by ASHRAE (1994). The degree of scatter in their results may have been due to the simplifying assumptions implicit in the linear model used. The authors recommended the use of an electronic expansion valve (EEV) to over come this problem. Another solution suggested by the authors of this paper may have been to use a pumped return from the condenser to the evaporator.

Ndiaye and Bernier (2009) modelled refrigerant flow through a TEV bleed port. A bleed port is required for systems that need to quickly equalise evaporator and condenser pressures before compressor start-up. They used four types of inlet/outlet steady-state flow condition and showed that their model's predictions generally compared well with experimental results.

The research described in this paper forms part of a wider study aimed at the development of high fidelity models that simulate the transient and steady-state operation of whole refrigeration systems of the type used for the storage and processing of food. At the time of writing, the models are integrated within a software package titled Vapour Compression Refrigerator System (VCRS) simulator. This has been described by Eames et al. (2010, 2012) and was developed as part of a larger study into energy use by refrigeration systems in the food chain carried out for the UK Government's Department for the Environment, Food and Rural Affairs as described by James et al. (2009). This simulator provides a fully integrated model that simulates food cooling processes, cold store or room and refrigeration system, together, in a single package. This allows practitioners to investigate the implications of refrigeration system design choices in terms of energy usage and carbon generation. At the time of writing the VCRS package can be downloaded free of charge from the project website at; http://www.grimsby.ac.uk/industry/defra-refrigerationmodels.php or from the corresponding author.

In order to enable a wide range of refrigerator systems to be easily modelled it was necessary that the TEV model was not complicated by the need for geometric descriptions of the valve or empiric performance data. This paper shows that the aim was achieved and in addition the steady-state model provides described here provides a useful correlating equation for manufacturer's data, if that is required. Together with a transient model, a practical way is provided to simulate and investigate operational problems sometimes encountered in refrigeration system. These included such problems resulting from over-sizing or under-sizing a TEV, or by incorrect installation.

Section 2 provides descriptions of various types of TEV covered by the research described in this paper. Sections 3 and 4 describe the derivation of a TEV steady-state model. Part-load performance of a TEV is discussed in Section 5 and a simplified transient model is derived in Section 6. Some validation results are presented in Section 7 and in Section 8 results for the VCRS software, which incorporates the models described in this paper, are used to aid the discussion of TEV operating faults.

2. Operation of a thermostatic expansion valve

Fig. 1 shows a schematic view of an internally equalised, liquid charged, thermostatic expansion valve (TEV) of a type used in commercial refrigeration systems, Dossat (1978). With regard to a refrigeration system, the functions of such valves are:

(i) Maintain a required pressure difference between the condenser and evaporator.
(ii) Meter refrigerant flow to the evaporator at (ideally) the same rate as it is drawn off by the compressor.
(iii) Protect the compressor from liquid ingestion by maintaining an acceptable degree of superheat in the suction line; normally set to be between 5 K and 10 K.

The operation of this type of valve and other types of TEV are described by ASHRAE (1994). The same reference gives the capacity of TEV to be,

$$\dot{Q}_e = C_o\sqrt{\rho_{c,f}(P_c - P_e)}\left[h_{e,g} - h_{c,f}\right], \quad (1)$$

where C_o is a constant dependant on valve geometry. Valves that use the same refrigerant in their sensing bulb as used in the refrigerator are termed 'straight-charged'. In practice it is common to charge a sensing bulb with a refrigerant or a blend of refrigerants that has a higher saturation pressure than the refrigerant in the refrigerator evaporator at a lower saturation temperature. This prevents excessive compressor suction vapour superheat at low evaporator temperatures. Valves which use a different refrigerant to that in the refrigerator system are know as 'cross-charged'. To model cross-charged valves at steady-state it is necessary to know the saturation pressure–temperature relationship of the refrigerant or blend of refrigerants contained by the sensing bulb. For commercial reasons this information is not normally made available by valve manufacturers and therefore, for the purposes of refrigeration system modelling it is usual to assume that a TEV is straight charged. However, as shown later in this paper it was interesting to observe that the model described here, which assumes a straight-charged liquid valve, appears to

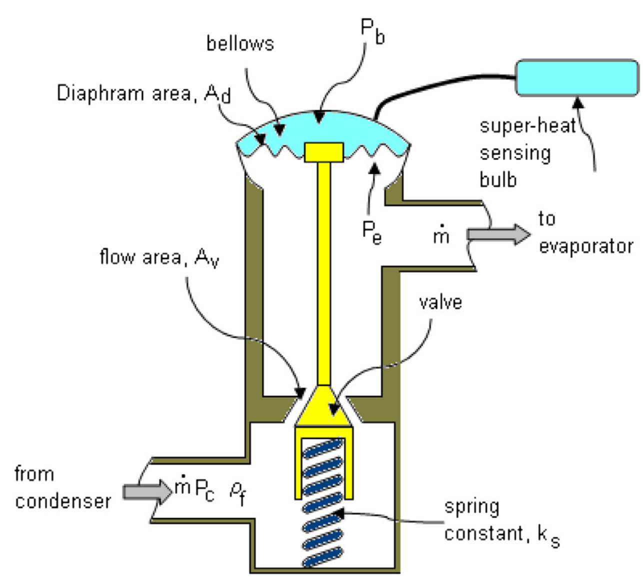

Fig. 1 – Schematic view of TEV model.

satisfactorily correlate the performance of a proprietary cross-charged valve within its normal operating temperature range.

Cross-charged valves, like 'gas-charging' valves discussed later, provide the added benefit of limiting the maximum operating pressure (MOP) in the evaporator. This can help to protect compressor drive motors from being overloaded during start-up by preventing excessively dense vapour being drawn from the evaporator when the air entering the evaporator might be significantly warmer than the design condition, (Dossat, 1978).

Under normal operation the quantity of refrigerant contained in the sensing bulb of a gas-charged or MOP (maximum operating pressure) valve is significantly smaller than that contained by a liquid-charged valve, whether it is cross-charged or straight-charged. As the sensing bulb of a MOP valve is heated its rising temperature causes the saturation pressure of the liquid within the valve to increase. Thus, within the normal control range of the valve the bulb-pressure, (P_b) and bulb-temperature are both saturated and related logarithmically and the valve behaves like a conventional cross- or straight-charged valve. However, above a predetermined saturation temperature the valve is designed so that all the liquid within the sensing bulb will have evaporated. With only vapour in the sensing bulb the controlling pressure becomes almost directly proportional to the absolute temperature of the bulb ($P_b \propto T_b$) as the containment volume of the valve is approximately constant. The effect this produces is to close the valve when the evaporator temperature is greater than that determined by MOP setting of the TEV. With the valve closed the saturation temperature in the evaporator falls rapidly when the compressor is started and this results in a significant reduction in start-up torque.

For gas-charged or MOP valves it important that the bellows is always warmer than the sensing-bulb in order to prevent any liquid contained in the bulb from migrating to the bellows causing the valve to malfunction.

Another cause of malfunction can result from the use of highly dynamic evaporators such as those that contain relatively small amounts of refrigerant, such as plate-type evaporators, or systems that can have short ON-OFF cycling times. Such evaporators might be found in air conditioning systems. The response of a refrigeration system to changes in air-on temperature, or chilled water-on temperature, is normally dampened to a greater or lesser extent by the thermal mass and liquid volume of the evaporator. However, if this 'natural' damping is insufficient, because of a small liquid volume for example, then 'artificial' damping may be introduced through the addition of 'ballast' to the sensing bulb. This increases the time constant (τ) of the valve by raising its thermal mass.

3. A theoretical mass flow model

The following describes theoretical models for liquid or gas, straight or cross charged thermostatic expansion valves that are either internally or externally equalized.

Equation (1) adequately models the capacity of a TEV when the degree of evaporator superheat is fixed, which may be assumed to be for steady-state performance at a given system design condition. However, under part-load or transient operation the degree of superheat can change and doing so will alter the capacity of the valve. This occurs both when the evaporator saturation temperature increases or decreases from its design-point condition and when the refrigerant flow through the TEV does not equal that through the compressor. This flow mismatch causes the evaporator to become either flooded or starved of refrigerant.

From Equation (1) the refrigerant flow through a TEV is given by,

$$\dot{m} = C_1\sqrt{\rho_{c,f}(P_c - P_e)} \tag{2}$$

Referring to Fig. 1, the constants of proportionality, C_o and C_1, in Equations (1) and (2), are functions of the flow area, A_v. To determine the nature of this function a force balance is applied to the spring-bellows system:

$$(P_b - P_e)A_d = k_s(x_o - x)$$

Where,

x_o = length of spring when fully extended
x = length of compressed spring

In the above force balance the effect of pressure differential across the valve port is assumed to be negligible compared with that of the bellows.

Letting $\Delta x = (x_o - x)$, when the valve is closed the force reaction of the valve seat $= k_s\Delta x_o$

Where the subscript 'o' is used here to denote values at the point of valve opening.

For the valve to be open,

$$(P_b - P_e)A_d > k_s\Delta x_o$$

Therefore, for the valve to just begin to open,

$$(P_b - P_e)A_d = k_s\Delta x_o$$

The bulb pressure at which the valve just begins to open can be adjusted by tensioning the control spring. This has the effect of setting the minimum degree of superheat in the compressor suction line where the sensing bulb is located, as shown in Fig. 1. The superheat temperature required to just cause the valve to begin to open is known as the 'static superheat setting' (SSS).

Therefore, when $(P_b - P_e)A_d > (P_b - P_e)_o A_d$ then, $k_s\Delta x > k_s \Delta x_o$

Therefore,

$$(\Delta x - \Delta x_o) = \left[(P_b - P_e) - (P_b - P_e)_o\right]\frac{A}{k_s} \tag{3}$$

Fig. 2 shows the geometry of the valve stem in relation to its seat when the valve is open. Referring to Fig. 2, the flow area, A_v, is given approximately by,

$$A_v = \pi.d.a$$

The distance 'a' is proportional to the valve stem position, $(\Delta x - \Delta x_o)$, and therefore,

$$A_v = C_2(\Delta x - \Delta x_o)$$

Where C_2 = constant of proportionality

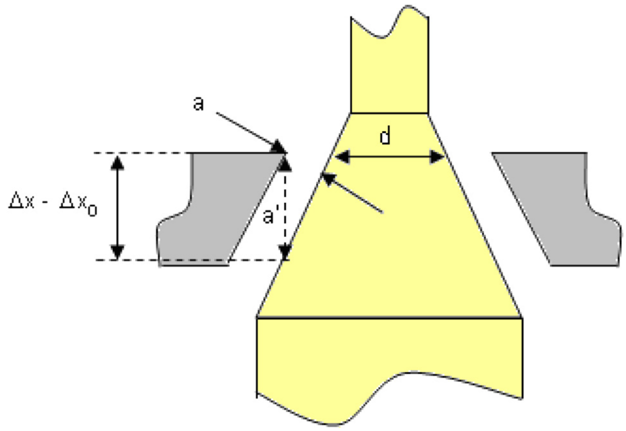

Fig. 2 – Flow area through valve, A_v.

Substituting this result into Equation (3) and collecting together all the constant terms together gives,

$$A_v = \beta\left[(P_b - P_e) - (P_b - P_e)_o\right]$$

In this paper β is referred to as the valve flow area constant. Increasing β causes the capacity of a valve to increase and vice versa. In other words, if a system has a TEV fitted that has a β-value larger than what should normally be fitted then the valve would tend to always over feed evaporator with refrigerant. This would cause the evaporator temperature to constantly vary; rising and falling in a sinusoidal way which is referred to as hunting.

Substituting the above result for the constant, C_1, in Equation (2) and letting $\alpha = (P_b - P_e)_o$ gives the mass flow through the valve:

$$\dot{m} = \beta[(P_b - P_e) - \alpha]\sqrt{\rho_{c,f}(P_c - P_e)} \text{ for } \alpha \le (P_b - P_e) \le \delta, \tag{4a}$$

where α is the pressure equivalent of static superheat setting (SSS) and δ is the value of $(P_b - P_e)$ when the valve is fully opened.

$$\dot{m} = 3.04939\times10^{-7}[(P_b - P_e) + 20.12477]\sqrt{\rho_{c,f}(P_c - P_e)}(\text{kg s}^{-1}) \text{ for } \alpha \le (P_b - P_e)\rangle\delta \tag{5}$$

$$\dot{m} = \beta[\delta - \alpha]\sqrt{\rho_{c,f}(P_c - P_e)} \text{ for } (P_b - P_e)\rangle\delta \tag{4b}$$

$$\dot{m} = 0 \text{ for } (P_b - P_e) \le \alpha \tag{4c}$$

If a gas-charged or MOP valve is to be modelled, once the liquid in the sensing bulb has evaporated and if further volume change in the bellows is neglected, then,

$$P_b = P_b(\rho_b^*, T_b) \text{ for } P_b \ge P_{\text{mop}}, \tag{4d}$$

where, ρ_b^* = saturated vapour density at a pressure equal to P_{mop}.

The value of α is determined by the minimum degree of superheat set by the valve, and δ is the pressure equivalent to the degree of superheat needed to fully open the valve. When a TEV is full-open the mass flow becomes a function of $(\rho_f(P_c - P_e))^{1/2}$ as given in the ASHRAE Handbook (1994).

4. Determination of the valve constants

4.1. Values of α and β for a model valve

For a theoretical model of a valve, the value of α the pressure equivalent to the static superheat setting) in Equation (4a) is set to equal the pressure difference, $(P_b - P_e)$, required for zero mass flow at the required minimum degree of superheat temperature. At that condition the valve is just closed or just starting to open. In terms of temperature, this pressure difference will equate to the static superheat setting. The value of the valve area flow constant, β can be determined using values of $\dot{Q}_e$, $(P_b - P)_e$, ρ_f and $(P_c - P_e)$, taken at a refrigerator's known design-point conditions. If a sensing bulb is charged with the same refrigerant as the evaporator it controls then the value of P_b in Equation (4) equals the saturation pressure of the refrigerant at the desired superheat temperature at the location of the sensing bulb in the compressor suction line.

4.2. Values α and β for a real valve

If a model of a proprietary valve is required then values for α and β can be determined using information provided by manufacturer's data sheets. Fig. 3 shows comparative data for a Danfoss TS2; an internally equalised valve with a cross-charged bulb using Danfoss capsule 03 and working with R404a. The values of α and β were calculated using a linear regression curve of $\frac{\dot{m}}{\sqrt{\rho_f(P_c - P_e)}}$ against $(P_b - P_e)$. The resulting characteristic equation in this example is,

In this case, the results in Fig. 3 show that Equation (4a) provided an accurate correlation of the manufacturer's data. It is thought reasonable to assume that this would be true in any other case, although the correlation should be checked for each. In Fig. 3 the error in mass flow between the correlation and model is less than +/− 2%.

5. Part-load performance characteristics of a TEV

The capacity of a thermostatic expansion valve is defined by ASERCOM, (2005) as,

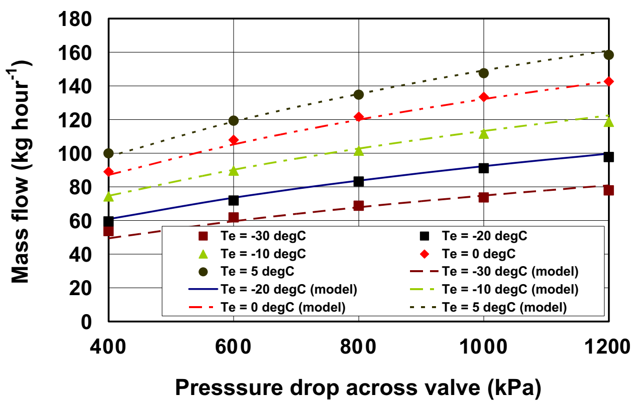

Fig. 3 – Comparison between manufacturer's data for a Danfoss TS2 (internally equalised, cross-charged bulb), Capsule/Orifice 03, working with R404a, and the model of a TEV defined by Equation (5).

$$\dot{Q}_e = \dot{m}(h_{e,g} - h_{e,i}), \tag{6}$$

where $h_{e,g}$ is the enthalpy of saturated vapour leaving an evaporator and $h_{e,i}$is the enthalpy of wet-vapour entering.

Fig. 4 shows how the capacity of a TEV changes with superheat temperature. In Fig. 4, the static-superheat setting is normally set at the rating condition by the valve manufacturer, although this can be adjusted by varying the tension in the control spring, shown in Fig. 1. Based on the ASERCOM (2005) proposals, static-superheat would be set between 3 and 4 K. The 'opening-superheat' is the superheat needed to open the valve between the just-closed position and that needed to ensure the capacity is equal to the valve's nominal capacity at the rating conditions. ASERCOM (2005) propose that this degree of superheat should not be greater than 5 K. Therefore, for the valve to be fully open the degree of superheat should not exceed 8–10 K. ASERCOM (2005) standard rating conditions are, $T_{e,sat} = 4$ °C, $T_{c,sat} = 38$ °C with 1 K of sub-cooling. At these conditions the rated differential pressure across a TEV should equal the difference between the saturation pressures in the condenser and evaporator.

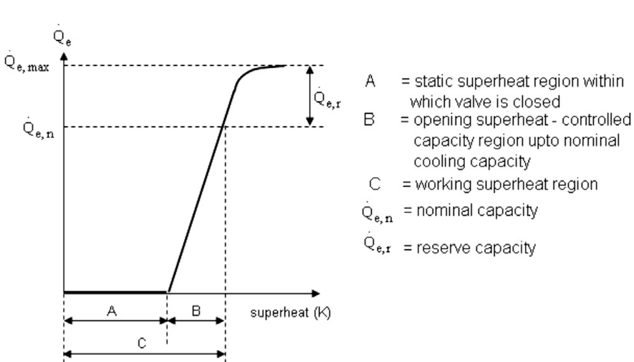

Fig. 4 – Capacity-superheat operating characteristic of a typical TEV (ASERCOM, 2005).

The maximum capacity of a valve may be greater than the nominal capacity at the rating conditions, thus providing an amount of reserve capacity. ASERCOM (2005) does not specify a limit on reserve capacity. However, the characteristic in Fig. 4 suggests this might be as much as 50%.

The significance of the characteristic shown in Fig. 4 is the linear relationship between capacity and opening superheat within the control region and the reserve capacity above the nominal capacity. Any model of a TEV must, therefore, reflect both these characteristics. Fig. 5 shows the results for the theoretical steady-state model, defined by Equations (4a)–(4c), for an internally equalised, straight-charged, TEV operating on R134a with a nominal capacity of 10 kW at the rating conditions. The results in Fig. 5 show that the correlation satisfies this requirement.

6. Transient responses of a TEV to changes in superheat

The flow predicted by Equations (4a)–(4d) represent steady-state values for given condenser and evaporator conditions. In practice the valve response will not be instantaneous and there will be a delay in response between a change in superheat temperature and a corresponding change in refrigerant flow through the valve whenever the saturation conditions in the evaporator and condenser alter. It is reasonable to assume that for most cases (excepting those requiring the use of a ballast type sensing bulb discussed earlier) the response time of a valve will be much shorter than that of evaporator and condenser to changes in operating conditions. This transient behaviour of the evaporator-valve system is assumed here to be due only to the thermal resistance between the bulb and suction pipe and thermal capacity of the bulb.

$$\overline{MC}\frac{dT_b}{d\theta} = UA_b(T_{e,sh} - T_b) \tag{7}$$

Integrating Equation (7) between time θ and $\theta + \Delta\theta$, and assuming $(T_{e,sh} - T_b)_\theta$ is known at $\theta = \theta$ gives,

$$(T_b)_{\theta+\Delta\theta} = (T_{e,sh})_{\theta+\Delta\theta} - (T_{e,sh} - T_b)_\theta \exp\left(-\frac{\Delta\theta}{\tau}\right), \tag{8}$$

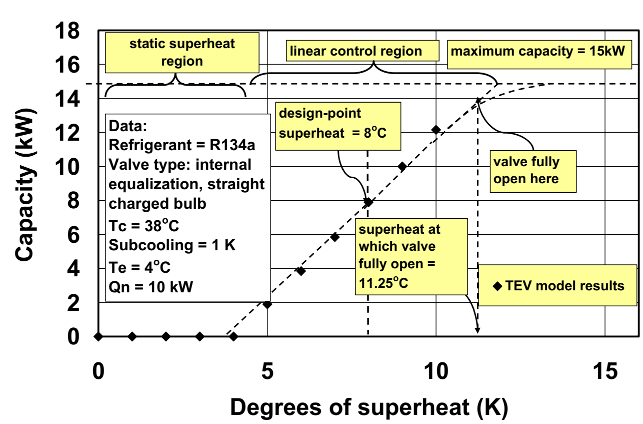

Fig. 5 – Results of the theoretical model for a TEV operating on R134a with a nominal capacity of 10 kW at the ASERCOM (2005) rating conditions.

where $\tau = \frac{MC}{UA}$ is the response time for the valve. The result shown in Equation (8) assumes:

1) Valve stem stiction, friction and inertia are negligible.
2) The temperature of the liquid within the bulb and the body of the bulb are equal.
3) Momentum changes of the liquid in the liquid-line leading to the TEV are negligible.

Therefore, at any time, $(\theta + \Delta\theta)$, the bulb temperature may be calculated using Equation (8). The bulb pressure can be determined using a saturation pressure-temperature correlation and thus the refrigerant mass flow through the TEV can be determined from Equations (4a)–(4d), depending on the type of valve and the degree of superheat.

7. Model validation

The theoretical model described by Equations (4a)–(4d) and Equation (8) was incorporated VCRS simulator, (Eames et al., 2010). Fig. 6 shows a comparison between experimental and theoretical results for the VCRS software validation case study on a 6.2 m^3 air blast refrigerated food cooler described by Eames et al. (2012). The time-constant value for the TEV model in this case was 20 s. The refrigerator supplied 13 kW of cooling at its design condition. The system included a 7 kW defrost heater which was timed to come on for 20 min every 4 h. The 'saw-tooth' characteristic of the graph lines in Fig. 6 is due to the 1.5 kW evaporator fans remaining on when the compressor was shut-down by a thermostat in the cooler chamber. The comparative results shown in Fig. 6 serve to indicate the potential usefulness of the VCRS model.

As the VCRS software was designed to model transient responses of refrigeration systems to changes in inputs, such as ambient weather, refrigerator door opening, loading the refrigerator with food stuff and so on, the system was particularly useful for testing the TEV model described here.

The characteristic equation for the steady state operation of the valve was,

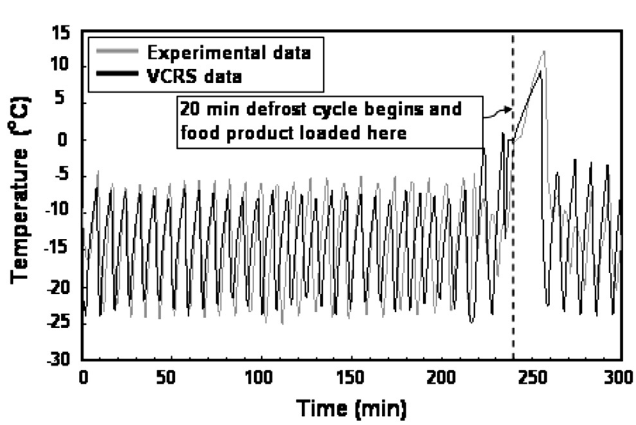

Fig. 6 – Comparative experimental and model data from the VCRS software: Variation in evaporator saturation temperature for a 13 kW (rated), 6.2 m^3 refrigerated air blast cooler.

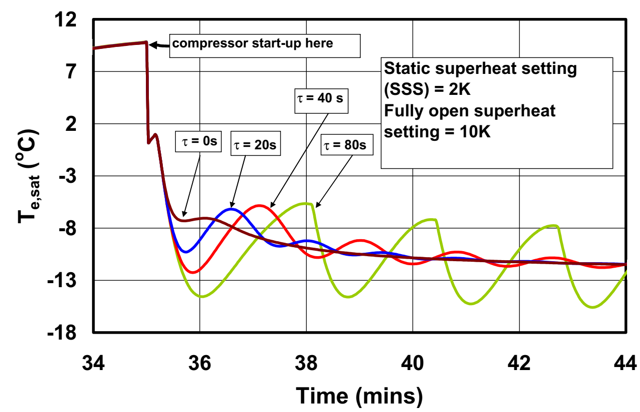

Fig. 7 – Showing the VCRS simulator predictions for the evaporator temperature of a large chill-store during start-up showing the effects of changing the time constant of a correctly sized straight liquid charged, internally equalised, thermostatic expansion valve.

$$\dot{m} = 1.775605 \times 10^{-6}[(P_b - P_e) + 34.9]\sqrt{\rho_{c,f}(P_c - P_e)}\,(\text{kg s}^{-1})$$

This equation is based on the following design point conditions:

Refrigerant = R404A
Evaporator heat rate = 13 kW
Evaporator temperature = 5 °Csat
Evaporator superheat = 5 K
Minimum superheat = 2 K
Condenser temperature = 40 °C
Condenser subcooling = 5 K

8. Results and discussion

The TEV model described in this paper was used in the VCRS software to study the performance effects of under-damping,

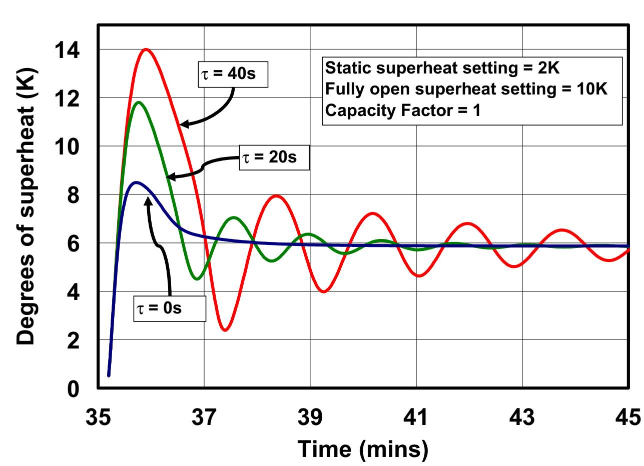

Fig. 8 – Showing the effect on the variation in superheat degrees (K) with time caused by changes in the time constant of a TEV.

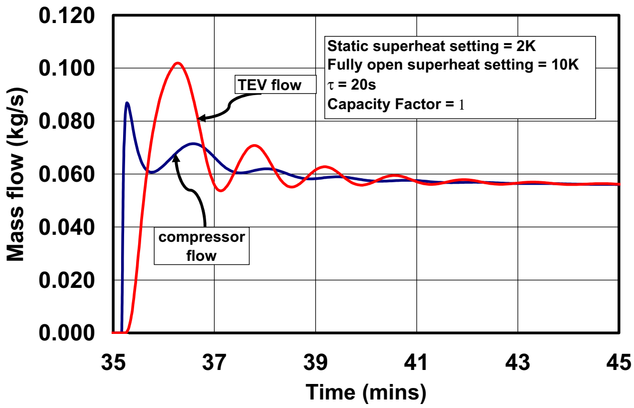

Fig. 9 – Showing the variations in valve and compressor flows (kg s^{-1}) with time for a design-point capacity TEV with a time constant of 20 s.

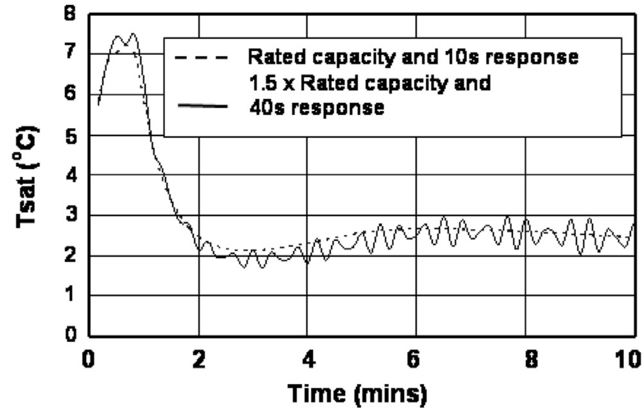

Fig. 11 – VCRS predication for the variation in evaporator temperature under 'normal' and hunting conditions.

caused by incorrect sizing of the TEV and inappropriate location and fitment of the sensing bulb, which are thought to increase the time constant of the valve.

Fig. 7 shows that as the time constant of a valve increases the control system becomes progressively under-damped. The results show that the initial amplitude and wave length of the temperature oscillations both increase with the time-constant. However, these oscillations gradually disappeared as the system came into balance in the manner of an under-damped control system. In this case the reason for the temperature oscillations is thought to be repeated increases and decreases in liquid level in the evaporator, which is reflected by the variation in super-heat temperature as shown in Fig. 8. When the degree of super-heat is large the TEV over-feeds the evaporator causing refrigerant to enter faster than the compressor can remove it, which causes the saturation pressure in the evaporator to rise as shown in Fig. 9. If the liquid content increases too rapidly the TEV control to over-shoots the balance condition, reducing the degree of super-heat and closing the TEV completely and starving the evaporator of refrigerant. A system with a correctly fitted TEV would quickly come into balance so that the degree of superheat is held constant and flow through the TEV equals that through the compressor.

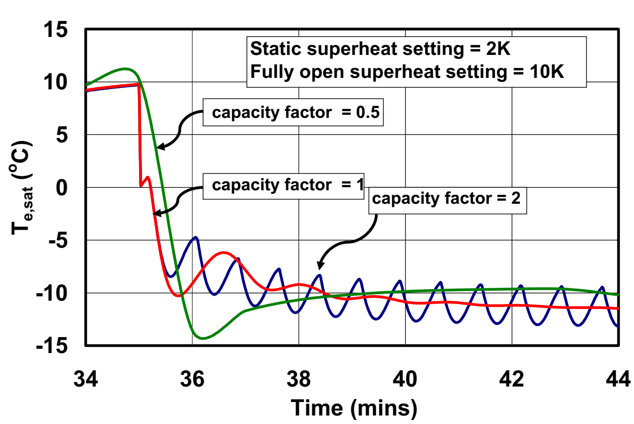

Fig. 10 – Showing the variation in evaporator temperature with time for a range of valve flow capacities. The results show that if a valve is oversized the evaporator temperature will 'hunt'.

Mithraratne and Wijeysundera (2002), reporting the work of Broersen (1982), suggest a time-constant of 20 s for a TEV. Although this value may be acceptable for modelling a whole refrigerator, in practice there may be occasions when less responsive valves are better, such as for evaporators using in air conditioning systems (Danfos, 2005).

8.1. *TEV hunting*

TEV hunting can be caused by number of operational, design or manufacturing faults. The most common faults are thought to be incorrect positioning and over-sizing of the valve. Fig. 10 compares the predicted results for a valve which has twice the required flow capacity (Capacity Factor = 2) with those for a correctly sized valve (Capacity Factor = 1) and an undersized valve (Capacity Factor = 0.5). These results show that over sizing a TEV causes the flow control to be under-damped, which causes the evaporator to be alternately over and under-fed with liquid refrigerant. As the liquid level rises the degree of superheat falls, causing the valve to move towards its closed position and slowing the flow into the evaporator. If the compressor draws refrigerant from the evaporator at a greater rate that it is supplied by the TEV then the liquid level will fall raising the degree of superheat in the suction-line and, in turn, increasing the flow through the TEV. This causes the liquid level in the evaporator to rise causing the TEV to close again and so on. Similar effects occur when a TEV is incorrectly positioned so that its response time is increase. Fig. 11 shows results produced by the VCRS for a correctly sized TEV and an oversized, unresponsive TEV. It is interesting to note that Mithraratne and Wijeysundera (2002) have found that the amplitude and frequency of the hunting oscillations could be reduced by increasing the time constant of the TEV bulb. A possible reason for this apparently contradictory result may lie in the type of evaporator used by Mithraratne and Wijeysundera in their experiments. As stated at the end of Section 2, if an evaporator is highly dynamic it can be necessary to slow the response of the controlling TEV by adding thermal ballast at the sensing bulb in order to prevent hunting.

By varying the response time and capacity for the TEV model and other design-point settings in the VCRS, other effects can be simulated. These include an undersized TEV, a punctured sensing bulb, and a partially blocked condenser.

9. Conclusions

The development of both steady-state and transient models for thermostatic expansion valves, of the types commonly found in commercial refrigeration systems, have been derived. Both models can be used without the need for manufacturer's data, which makes them useful when writing generalised transient or steady state models of vapour compression refrigerator systems. Notwithstanding the general nature of the models described by Equations (4a)–(4c), these were tested against manufactures data and the results were found to provide a close correlation, (Fig. 3). The equations were also tested against the ASERCOM (2005) TEV characteristic and again close similarity was found. The fidelity of the models within a complex refrigeration system, under realistic loadings, was tested using the VCRS software, Eames et al. (2012). Comparison between the prediction and measurements were excellent. The VCRS software results enabled the influence of TEV response time, over capacity and hunting to be investigated.

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