Refrigeration Control with Varying Condensing Pressures

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ABSTRACT

Allowing the condensing pressure of direct expansion refrigeration equipment to float (vary freely) with ambient temperature can lead to substantial energy savings. This paper presents experimental results of the performance of thermostatic expansion valves (TEV) under a range of condensing pressures, system loads and inlet quality. A theory is developed to explain the performance of the valves under these conditions. It is concluded that condensing pressures can be allowed to vary freely across a wide range of conditions provided that the TEV is correctly sized and that an envelope of safe operating conditions is defined to ensure satisfactory performance at all times.

INTRODUCTION

Direct expansion refrigeration equipment has been traditionally designed to maintain an artificially high condensing temperature irrespective of ambient conditions. However for a given evaporator duty and temperature, the lower the condensing temperature the lower will be the compressor power, and the higher will be the refrigerating capacity. This effect is demonstrated in Figure 1 which shows the theoretically calculated refrigerating capacity and coefficient of performance over a range of condensing temperatures for a typical air-conditioning system. It can be seen that, for most efficient operation, the system should run at as small a condensing temperature as possible and that substantial energy savings will result in so doing. Consequently full advantage should be taken of low ambient temperatures by allowing the condensing temperature to float (vary freely) with ambient temperature.

The most usual reason given for maintaining a fixed condensing temperature is that it maintains an adequate pressure drop across the liquid expansion device - usually a Thermostatic Expansion Valve (TEV) - for its correct operation (1). There is thus a need to establish to what extent, if any, the performance of a TEV is impaired by operating with a reduced pressure drop. The need to keep the pressure drop across the valve small to minimise energy use is now becoming recognised (2) but there is little data on the range of conditions that a TEV can be expected to perform over.

A TEV consists of an automatically adjustable orifice which modulates in response to a temperature sensor at the evaporator exit in an attempt to maintain a constant refrigerant superheat. If the valve is fully open any reduction in pressure drop across it will result in an inadequate supply of refrigerant to the evaporator, giving a reduction in refrigerating capacity and an increase in superheat. The overall state of affairs is illustrated in Figure 2 and shows how the equilibrium refrigerant mass flow rate varies as a function of pressure difference \((P_c - P_e)\) across the system. \(P_c\) is the condensing pressure and \(P_e\) is the evaporating pressure. Consider the curve labelled "design orifice". At low pressure differences i.e., up to a pressure difference of \(\Delta P_1\), the refrigerant flow is determined by the fully open orifice size of the TEV. The mass flow in this region is, to a first approximation, proportional to the root of the pressure difference. When the pressure difference is greater than \(\Delta P_1\) the flow is determined by the capacity of the compressor and the size of the orifice in the TEV will be reduced automatically to maintain the set superheat. The operating line set by the compressor slopes slightly downwards due to the reduction in volumetric efficiency as the condensing pressure is increased. From Figure 2 it is seen that the refrigerant flow and therefore the refrigerating capacity is small at low pressure differences i.e., low condensing temperatures, and thus it could be argued that the condensing temperature must be maintained high to avoid this. However it can be seen in Figure 2 that if the lowest attainable pressure difference for the system where \(\Delta P_2\), then
the problem of the valve restricting the capacity would be overcome by selecting a valve whose fully open characteristic where described by the curve labelled "large orifice". Unfortunately there is an upper limit to the size of TEV that can be used because of the problem known as "hunting". This is the tendency for a TEV to become unstable when the orifice is close to its fully closed position i.e., at high condensing temperatures and/or part load conditions, resulting in a cyclic overfeeding and underfeeding of the evaporator. The most serious consequence of this can be refrigerant liquid being admitted to the compressor leading to possible mechanical failure particularly in the case of reciprocating compressors. The limit of this unstable region is ill defined. Manufacturers recommend that in order to avoid this region the capacity should not be reduced to less than 30% of the declared capacity (3). Included in Figure 2 is a "hunting limit" defined by the "large orifice" closed down to 30% of its area. Part load conditions can be obtained by reducing the swept volume rate of the compressor and this operating condition is also illustrated in Figure 2. Clearly using a large orifice TEV to allow operation at a low pressure differences will not necessarily extend the operating range of the plant because the TEV may now hunt at low capacities and/or high pressure differences. One of the purposes of the work described here was to investigate experimentally the arguments addressed above and to test if they are reasonable.

The other area of TEV operation which this report will examine is that of the effect, on performance, of the presence of refrigerant vapour (flash gas) in the liquid at entry to the TEV. Any flash gas present in the liquid line will increase the mean specific volume of the refrigerant and could thus be expected to reduce the capacity of the expansion valve. A given mass of vapour will occupy a larger volume at low pressure than at high pressure and will consequently have a larger affect on the valve's performance - another reason cited for maintaining an artificially high condensing temperature (4).

**TEST EQUIPMENT**

The test equipment comprised a laboratory refrigeration plant with a design refrigerating capacity of 10 kW (2.8 ton) at an evaporating temperature of 0°C (32°F). The refrigerant used was R22. The evaporator was a plate heat exchanger with a basic rating of 1.7 kW/K (0.34 ton/°F) and the cooling load was provided by electric heaters in the recirculating chilled water line. The heaters could be thermostatically controlled to maintain a constant return temperature to the evaporator at full and part load conditions. The condenser was water cooled with a shell and tube configuration. The compressor was a two cylinder reciprocating open type with a water cooled head. Part load conditions were obtained by using different pulley combinations to vary the compressor speed supplemented by an evaporator pressure regulator valve. Two sizes of TEV's, with nominal capacities of 10 kW (2.8 ton) and 7.5 kW (2.1 ton) were tested. A microprocessor controlled valve - activated by a stepper motor - having an orifice of a similar size to the smaller TEV was also tested.

The liquid line from the condenser to the expansion valves was made as short as possible to minimise the built in pressure drop and so avoid possible vapour formation. However, in order to investigate the effects of the liquid line pressure drop on the performance of the expansion valve, a needle valve was incorporated in the liquid line so that pressure drops of various amounts could be effected. A sight glass was placed immediately in front of the expansion valve to give a visual indication of the presence of vapour.

All of the tests were performed with an evaporating temperature of 0°C (32°F) representative of typical air-conditioning systems. Most other types of refrigeration systems will have lower evaporating temperatures and consequently will always have a larger pressure drop available at the expansion valve. The water temperature onto the evaporator was maintained at 12°C (54°F) throughout. At part load conditions the water flow to the evaporator was adjusted, if necessary, to balance the load and prevent the evaporating temperature rising.
VALVE PERFORMANCE IN THE ABSENCE OF FLASH GAS

The results of the tests at the full and part load duty for each of the TEVs are shown in Figures 3 and 4. Included in these graphs are performance curves taken from the manufacturer’s catalogue. From Figure 3 it is clear that TEVs will control at duties considerably in excess of the nominal duty - 50% higher in this case.

Two regions of the results are important for this investigation. The first one is at low condensing temperatures when the desired refrigerating capacity cannot be maintained. The second is at part load conditions and high condensing temperatures where expansion valve instability is possible. With respect to the first region it can be seen that both the TEV’s behaved broadly as expected from the discussion in the introduction. The capacity of both valves was found to be significantly larger than their declared capacities and allows the system to maintain the set load at condensing temperatures much lower than that predicted from the declared data.

The second area of importance was that at high condensing temperatures and low load where instability was possible. It was expected that instability would occur when the expansion valve was near the closed condition i.e. when a combination of high pressure difference and low refrigerating capacity prevailed. This was, in fact, broadly the case and for both the TEV’s instability was observed only at the lowest capacity operating point i.e., 25% full load for the larger TEV and 33% for the smaller TEV. Instability was present for much of the operating range but diminished as the condensing temperature was reduced. The instability was greater in the larger valve at 25% load than in the smaller one at 33% load. The form of the instability, as observed from the liquid line flow meter, was different depending on the system’s operating pressure difference. At the low pressure differences variation in flow rate consisted of a fairly high frequency component (at approximately 0.5 Hz) superimposed upon a low frequency component with a frequency of approximately a minute or so. At these low frequencies the change in flow rate and corresponding change in refrigerating capacity was easily measurable and varied between limits of approximately 10% of the values indicated in the Figures. As the pressure difference was increased it was noted that the low frequency component diminished to zero but the high frequency component remained. At no time during the test programme did the instability have any serious adverse effect on the compressor.

Under no conditions was instability observed in the electronic valve. This would suggest that in this respect it is superior to the TEV because, at the same conditions, the smaller TEV which had a similar measured capacity to the electronic valve, was sometimes unstable. The stability of the system is largely dependent upon the gain of the valve (i.e. the change in valve stroke to the change in superheat signal). For a TEV this is more or less a constant fixed by the design of the valve. In the electronic valve the gain can take on a wider range of values, usually determined automatically by the control system. The control system is thus able to select a value for the gain low enough to ensure stability over a much wider range of operating conditions than can the TEV (S).

There were no adverse effects on the system due to running at reduced pressures. In fact the system was seen to be running much more quietly and smoothly than when operating at high condensing temperatures.

Figure 5 shows the effect of varying condensing pressure on the evaporator exit superheat and the system COP for the larger TEV operating at full capacity. Both are measured values. As expected the superheat remains substantially constant as the condensing temperature is reduced until a point is reached (at around 13 - 20°C (60 - 68°F) for this valve - see also Figure 4) where the valve becomes fully open and can no longer control. There after the superheat increases towards the water inlet temperature. The COP increases with decreasing condensing temperature, as predicted, but only whilst the valve is controlling. Once it is fully open the COP begins to fall due to the reduction in refrigerating capacity as the evaporator becomes starved of liquid. In a system without an evaporator pressure regulator the evaporating pressure will begin to fall once the valve is fully open and thus slightly more liquid would be fed to the evaporator than was the case in the experiments reported here. The rise in superheat and fall in COP would then be expected to be less rapid than those shown in Figure 5.
THE EFFECT OF FLASH GAS

To a first approximation the refrigerant mass flow rate, \( \dot{m}_r \), through an orifice with a pressure difference of \( P_c - P_e \), is obtained from Bernoulli's equation i.e.,

\[
\dot{m}_r = C_d A (2(P_c - P_e)/v_1)^{0.5}
\]

(1)

Where \( C_d \) is the discharge coefficient, \( A \) is the orifice area and \( v_1 \) is the specific volume of the liquid refrigerant. Now if there is a pressure drop of \( \Delta P \) in the liquid line then, providing the refrigerant remains liquid, the new refrigerant mass flow rate, \( \dot{m}_p \), can be calculated from equation 1 with \((P_c-P_e)\) replaced by \((P_c-\Delta P-P_e)\). Hence the fractional reduction in mass flow, \( M_p \), due to a pressure drop of \( \Delta P \) is:

\[
M_p = \frac{\dot{m}_p}{\dot{m}_r} = (1 - \Delta P/(P_c - P_e))^{0.5}
\]

(2)

If the pressure reduction causes the refrigerant state to pass into the wet region then vapour will be formed and there will be an additional reduction in mass flow rate. The amount of vapour formed i.e., the refrigerant quality, can be found by proceeding as follows. The refrigerant leaves the condenser as a liquid with a specific enthalpy \( h_c \) and, if there is no heat transfer or work performed, the enthalpy and mass flow rate remain constant. Then if there is a pressure drop of \( \Delta P \) in the liquid line the quality \( x \) of the refrigerant entering the expansion valve is given by

\[
x = (h_c - h_l)/(h_g - h_l)
\]

(3)

where \( h_l \) and \( h_g \) are the specific enthalpies of saturated liquid and vapour at the reduced pressure \( P_c - \Delta P \).

The mean specific volume of this mixture will then be

\[
v_m = v_g x + (1 - x)v_l
\]

(4)

where \( v_l \) and \( v_g \) are the specific volumes of the liquid and vapour at the reduced pressure. Thus if \( v_1 \) is the specific volume of the liquid at \( P_c \) and if it is assumed that the effect of the vapour on mass flow depends only on the mean specific volume, then the fractional reduction of mass flow as a result of the formation of vapour is

\[
M_v = (v_1/v_m)^{0.5}
\]

(5)

and the total fractional reduction in mass flow as a result of the reduction in pressure \( \Delta P \) when this results in the refrigerant entering the wet region is

\[
M_t = M_p M_v
\]

(6)

Experimental results showing the behaviour of the smaller TEV when flash gas is formed due to a liquid line pressure drop are shown in Figure 6. The tests were performed at a constant condensing temperature with the pressure drop before the TEV progressively increased by using the needle valve. The increasing quantity of vapour could be seen in the sight glass which was immediately before the expansion valve. Included in Figure 6 are the predicted values for \( M_t, M_p \) and \( M_v \) from the analysis given above. Considering the assumptions made in obtaining equation 6 the agreement between the measured values and the calculated \( M_t \) is surprisingly good. It is clear from Figure 6 that vapour formation has the dominant effect on valve capacity over the likely range of pressure drops to be encountered in practice. The amount of vapour formed is, of course, independent of the evaporator conditions.

Figure 7 is a plot of the calculated fractional reduction in mass flow rate, \( M_t \), through a fully open valve at a constant evaporating pressure of 0°C for various liquid line pressure drops. This shows that the reduction in mass flow rate resulting from a pressure drop is always significant but is much more so as the condensing temperature is reduced. The graph applies to any size of valve, the normal operating point when the liquid line pressure drop is zero being given by \( M_t = 1 \).
The effects of a fixed pressure drop from the saturated state are plotted for a range of condensing temperatures in Figure 8. The theoretical predictions for this condition were obtained using Equation 6 assuming that at the saturated state the TEV was fully open. These predictions are also shown in the Figure and the correspondence between theory and experiment is excellent.

The only other observation made about the effects of flash gas was during testing at the low duty conditions where hunting occurred. It was found that introducing a pressure drop into the liquid line to cause vapour to form stabilised the valve’s behaviour by effectively reducing its capacity. It has however been reported elsewhere that flash gas produced in long liquid lines can produce severe hunting of TEVs (6). This could clearly be the case if the liquid and vapour do not remain well mixed but rather arrive at the valve as slugs of alternatively liquid rich and vapour rich refrigerant.

CONCLUSIONS

1. It has been shown that thermostatic expansion valves (TEV) can operate satisfactorily with a wide range of pressure differences across them. The minimum pressure drop is set by the fully open valve. The fully open condition of the TEV is well defined by a simple square root relationship between flow and pressure drop.

2. Any TEV is likely to hunt (i.e. become unstable) at low duties and/or high condensing pressures. This region of operation should be avoided. However the region where hunting occurs is not well defined and is probably system dependant. It is suggested (based on manufacturer’s claims and the present study) that operating conditions where the TEV would be less than 30% fully open should be avoided. There is a need for better understanding of the conditions causing hunting.

3. The electronic stepper motor driven valve tested showed no signs of hunting over the range of test conditions.

4. Flash gas caused by pressure drops and/or heat gains in long liquid lines reduces the capacity of the valve. This reduction is well predicted by the simple theory presented. Flash gas reduces the range of safe operating conditions for a TEV. However it is suggested that the presence of flash gas is not itself a cause of hunting unless there is separation of the two phases. If phase separation is likely or if the range of operating conditions is too limited, consideration should be given to eliminating the flash gas. Flash gas is unlikely to be a problem in short liquid lines.

5. There is little justification for maintaining an artificially high condensing temperature. The system was found to operate more quietly at lower condensing temperatures and gave substantial energy savings provided that the liquid flow was not restricted by a fully open TEV.

6. TEVs have excess capacity above their rated capacity. Use of this excess capacity should be encouraged to allow a wider range of safe operating conditions. Manufacturers could help by quoting valve coefficients for their valves.

7. The TEV should be chosen to meet the full system capacity at the minimum pressure drop across it. An envelope of safe operating conditions should then be defined set by the fully open and hunting limits. If the plant is to operate beyond this range consideration should be given either to using more than one TEV and switching between them using a solenoid valve or to using an electronic valve.
REFERENCES

1. Whitman and Johnson, Refrigeration and air conditioning technology Delmar Publishers Inc. 1988
2. ASHRAE 1988 Equipment Handbook, Chapter 18
3. Sporlan Valve Company, Selection Guide

FIG. 1 REFRIGERATING CAPACITY AND COEFFICIENT OF PERFORMANCE FOR SYSTEM WITH AIR COOLED CONDENSER.
FIG. 2 REFRIGERANT MASS FLOW RATE AS A FUNCTION OF PRESSURE DIFFERENCE

FIG. 3 SYSTEM PERFORMANCE WITH A 7.5 kW CAPACITY TEV
Fig. 4 System performance with a 10 kW capacity TEV

Fig. 5 Superheat and coefficient of performance at design capacity with a 10 kW capacity TEV
Fig. 6 Comparison between theoretical and experimental mass flow ratio.

Fig. 7 Theoretical effect of pressure drop on mass flow over range of condensing temperature.
FIG. 8 EFFECT OF PRESSURE DROP OVER RANGE OF CONDENSING TEMPERATURE.