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Analysis of the expansion valves used for controlling refrigerant feed into delicatessen cabinets in supermarkets

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Abstract

Delicatessen cabinets can be found in most supermarkets and are used to retail and display unwrapped chilled foods, including fresh and cooked meats, salads and pies. High operating costs for these cabinets have been regularly reported and this is mainly due to evaporation losses which occurs whilst unwrapped food is on display. One of main reasons for this has been cited to be low humidity within the display area, however, the cause of this is not clear.

The role of the expansion device in the poor performance of the delicatessen cabinet is examined in this paper. This includes the results from a practical and theoretical investigation into the behaviour of thermostatic expansion valves (TEV) in the delicatessen cabinet. This showed standard expansion valve designs to be unstable and to constrain the level of humidity achieved in the display area. Alternative expansion devices are reviewed and the performance of the short tube restrictor device modelled. This has shown that this device will overcome the shortcomings of the traditional expansion devices and will significantly reduce the operating costs of the delicatessen cabinet.

Keywords: Cooling; Delicatessen; Cabinet; Drying; Expansion device; Retail display; Refrigeration; Short tube restrictor; Supermarket; TEV; Unwrapped food; Weight loss

Nomenclature

A	Area	m^2
C_p	Specific heat capacity	kJ/kg K
K_D	Mass transfer coefficient	kg/m^2 Pa s
H	Enthalpy	kJ/kg
$\dot{m}$	Mass flow rate	kg/s
P	Partial pressure	Pa
T	Temperature	°C
U	U-value	kW/m^2 K

Subscripts

a	Air
b	Boiling refrigerant
e	Evaporating water
f	Food surface
i	Refrigerant inlet
o	Refrigerant outlet
s	Subcooled refrigerant

1. Introduction

Delicatessen cabinets have been used for many years to display for sale unwrapped perishable foods in grocers shops (Griffiths, 1951). In recent years the sale of chilled foods has grown rapidly in Western countries, in some countries by about 5% per year (Bøgh-Sørensen & Olsson, 1990). One of the reasons given for this growth is that many consumers believe that chilled food is fresh food. There has also been an increase in demand for convenience and lightly processed ready meals of between 20 to 40% per annum (Bailey, 1989).

This combined with a diversification in the eating habits of the consumer resulting from internationalism has produced an increased demand for a range of premium, ethnic, convenience, fresh and unwrapped chilled delicatessen foods (Meffert, 1990). As these foods have a high added value the major supermarket chains have been keen to encourage this and in recent years the use of delicatessen cabinets has expanded (Lyons & Drew, 1985).

This greater emphasis has highlighted the deficiencies of delicatessen cabinets and exposed their high operating costs (James & Swain, 1986). Despite this, the design of delicatessen cabinets does not appear to have changed since the early 1950s. This paper addresses one aspect of the delicatessen cabinet design; the choice of expansion device used to control the refrigerant feed to the evaporator.

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This paper initially addresses the role of the delicatessen cabinet and discusses the reasons for high operating costs. The role of the traditional thermostatic expansion valves (TEVs) in delicatessen cabinets is described and their performance in the cabinet is shown both practically and theoretically to be a contributing factor to the high operating cost. Alternative expansion devices are reviewed and a device is proposed that will overcome the shortcomings of the traditional expansion devices. The performance of this device is evaluated using a computer-based model together with the economics of this change.

2. Refrigeration in the delicatessen

Delicatessen cabinets have two roles. Firstly, they display the food attractively and secondly they refrigerate the food. This prevents the foodstuff becoming unfit for human consumption and preserves it as near as possible to its fresh initial state. Any deterioration that causes a change in the appearance, odour, taste or weight of fresh food immediately reduces its commercial value and thereby represents an economic loss.

The range of foods that are displayed in the delicatessen cabinet is large and the foods are categorised into specific groups. Fig. 1 shows the percentage of the cabinet that is typically occupied by each food group (Graham, 1995) and from this it can be seen that chilled meats (including cooked, fresh, continentals and patés) dominate.

The changes that can occur with time to the chilled foods sold in delicatessens are numerous. Deterioration and eventual spoilage of perishable food are caused by a series of complex micro-biological, chemical and physical changes that take place in the foodstuff following harvesting and/or manufacture (Dossat, 1981). For the range of unwrapped chilled foods on display, Maidment (1998) found that over the retail display life, the normal reason for commercial loss with all delicatessen foods was surface drying. This affects the food in two ways; firstly it leads to colour/textual changes that are undesirable and necessitate trimming, secondly, surface drying leads to significant weight loss from the food. As

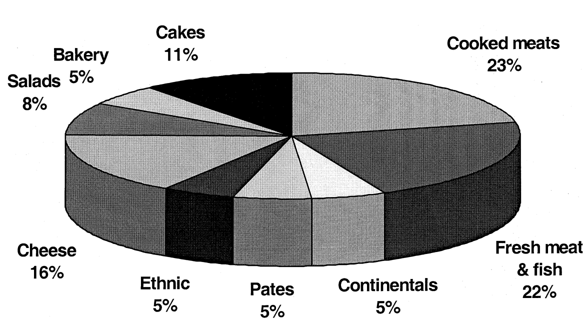

Fig. 1. The proportion of food products retailed from the delicatessen cabinet in the typical supermarket.

most of these products are sold on a weight basis a small percentage weight loss can result in a significant loss of profit. Typically in the delicatessen this can be as high as 5% of the total mass retailed (Stephens, 1995), where profit margins are less than 10%.

The design and operation of a typical delicatessen cabinet is shown in Fig. 2. From this it can be seen that the food is displayed on a base plate and is cooled by cold air passing over it. The air is then recirculated through the cabinet and is cooled by a refrigerated plate fin evaporator.

Food exposed to air loses moisture by evaporation and the drying of a typical food product is described in Fig. 3 (Radford, Herbert & Lovettet, 1976). After a short settling down period at point A, the food surface comes into equilibrium with the refrigerated air stream and the constant rate-drying period begins. This continues until point B is reached, when the rate of transfer of moisture from the product to its surface becomes less than the loss from the surface, and the surface begins to dry out. At point B the rate of drying then reduces and the period known as the falling rate period is entered. Point C shown on the graph represents the critical or equilibrium moisture content.

In the delicatessen cabinet moisture losses from food normally occur in the constant rate period (A to B) (Maidment, 1998). The rate at which food loses moisture during this period is given by Dalton's law

$$\dot{m} = K_D A(P_f - P_a). \tag{1}$$

Maidment (1998) investigated the effect of the environment surrounding the food on display in the delicatessen and determined a set of optimum conditions for minimum weight loss, a summary of the findings are shown in Table 1.

This paper considers the importance of maintaining high humidity only. The importance of air temperature, air velocity, radiant heat and conduction on weight loss is discussed elsewhere (Maidment, 1998; Maidment, Missenden, James & Tozeret, 1998).

Humidity is important, as the vapour pressure term P_a (in Eq. (1)) is a function of the saturated vapour pressure of the air and the relative humidity. Whilst a high RH is desirable, delicatessen cabinets available on the market have been reported to achieve only moderate humidities typically from 47% to 69% (James & Swain, 1986).

Dehumidification is caused by the evaporator surface temperature or apparatus dew point being well below the temperature of the air passing through it (Wile, 1966). Whilst there are a number of potential causes for this excessively low surface temperature, according to James and James (1987), inefficient heat exchanger performance is partly because these evaporators utilise TEVs, which produce a zone of poor heat transfer in the evaporator, as they attempt to provide superheating of the vapour of 5–10 K.

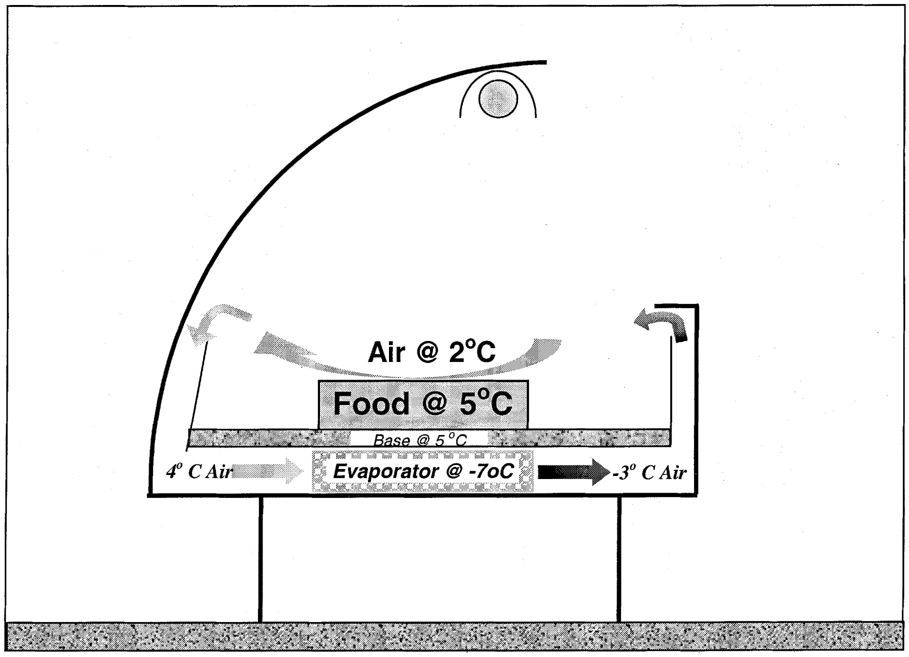

Fig. 2. Operation of a typical delicatessen cabinet.

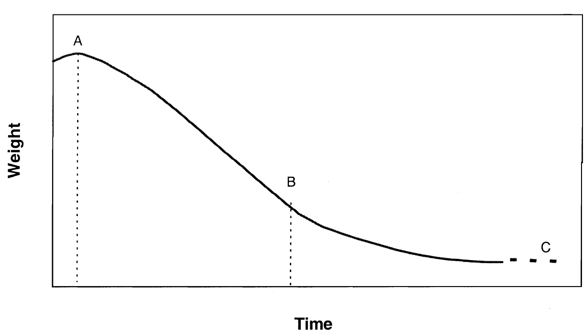

Fig. 3. Typical drying curve for food.

3. TEV operation in the delicatessen

The refrigeration system used to cool the air in the delicatessen cabinet is shown in Fig. 4.

The purpose of the TEV in this system is two fold; to reduce the pressure of the liquid refrigerant and to regulate the flow of refrigerant to the evaporator to satisfy the load and avoid returning liquid refrigerant to the compressor. The TEV achieves this by controlling a superheat temperature at the evaporator outlet. The TEV is a proportional controller, which responds to the difference between the pressure of the refrigerant at the position that the pressure-sensing connection is made and the pressure developed in the temperature-sensing remote bulb located at the outlet of the evaporator. The bulb is normally charged with refrigerant. A variable spring setting in the valve is provided to adjust the required superheat of the vapour leaving the evaporator.

Table 1
Thermal criteria specified for food storage

Optimum environmental criteria for unwrapped food
Minimum air temperature: 0–2°C
Minimum air velocity: <0.2 m/s
High Relative humidity: >90%
Minimum radiant heat
Low base temperature 0 to −2°C

3.1. Performance of the TEV in the delicatessen cabinet

The performance of the TEV in the delicatessen was investigated using a purpose built experimental apparatus. The experimental apparatus developed consisted of a traditional delicatessen cabinet (George Barker Series 2000, model no. BUHM) with a direct expansion plate fin evaporator, solenoid and TEV. This was connected to the purpose built refrigeration circuit, which included a condensing unit consisting of a hermetic compressor, air-cooled condenser and liquid receiver. The refrigerant employed was R22 and the rated capacity of the cabinet was 1.1 kW at an evaporating temperature of −7°C.

The TEV provided with the cabinet was Danfoss TX2 internally equalised TEV with an orifice number OO. Whilst this was designed to control a superheat of between 5–7 K at the evaporator outlet, this was unachievable at any superheat setting. The valve was either unstable with superheat varying by between 0–12 K as shown in Fig. 5 or stable with saturated gas at the evaporator outlet, as in Fig. 6. Whilst, the valve

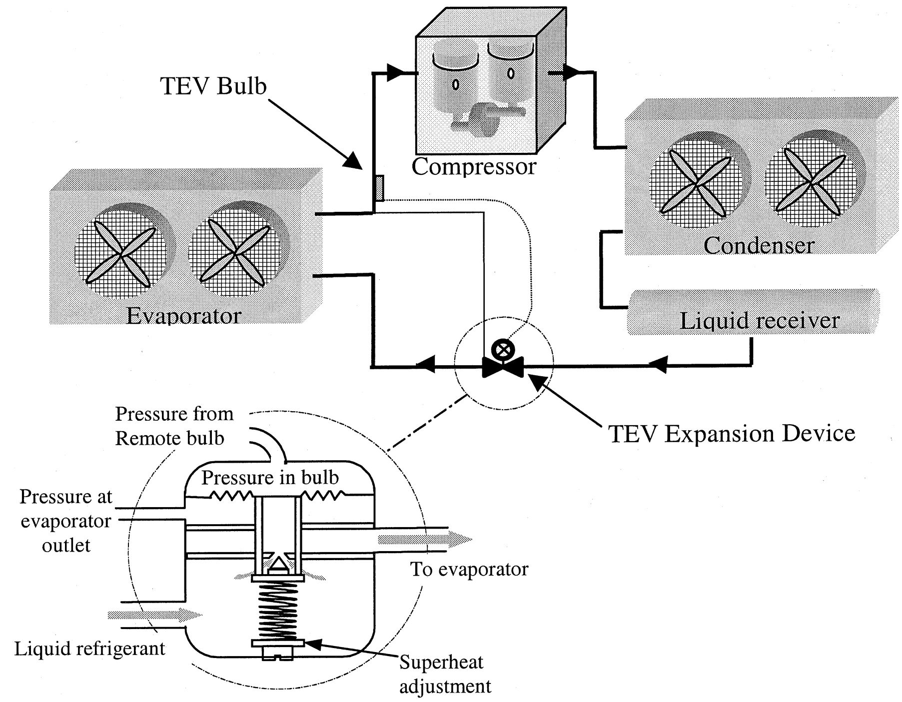

Fig. 4. A simple refrigeration system with a thermostatic expansion valve used to control the evaporator feed.

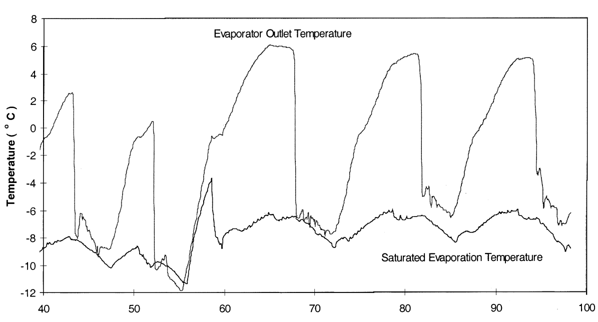

Fig. 5. Unstable operation of Danfoss TX2 valve.

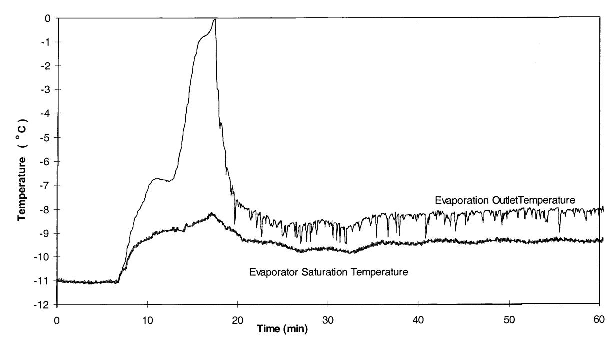

Fig. 6. Stable operation of Danfoss TX2 valve with saturation at evaporator outlet.

maintains a stable evaporator outlet temperature only by allowing the wet refrigerant to leave the evaporator.

3.2. Major factors influencing TEV behaviour

This undesirable oscillating behaviour influencing TEV performance has been theoretically investigated by James and James (1987). They used a mathematical model to show that thermal resistance between the bulb and evaporator outlet tube, the vapour velocity at evaporator outlet and the vapour volume in the bulb and head all influence valve stability.

Thermal resistance between bulb and tube was significant as changing this resistance resulted in a different response time. Broerson and van der Jagt (1980) who also investigated the stability of the TEV and evaporator arrived at similar conclusions and showed that stable operation occurs if this resistance is greater than 4 K/W.

The suction vapour velocity also changed the heat-transfer coefficient between the suction vapour and the tube wall. Finally, the influence of the volume occupied by the vapour in the TEV body was investigated by increasing vapour volume in the model; this increases the quantity of vapour required to produce a specific pressure rise. The results demonstrated that a smaller vapour volume produced a quicker response time.

The reasons for valve instability have been also reviewed by ASHRAE (1994) which reported a number of potential causes for valve instability. Firstly, it was stated that valve instability could be due to an excessive evaporator pure time delay leading to a continuous overshooting of the valve. This occurs because as the bulb signals for a change in refrigerant flow the refrigerant must traverse the entire evaporator before a new signal can reach the bulb. Other possible factors for valve instability were valve over sizing, incorrect positioning of the bulb, excessive evaporator pressure drop, incorrect refrigerant charge, excessive liquid line pressure drop and interference from the environment surrounding the bulb.

3.3. Reasons for valve instability

The factors for valve instability were investigated using the experimental plant and it was established that instability was due to valve sensitivity, and this was demonstrated by increasing thermal resistance between the bulb and the sensing pipe, with the addition of PTFE tape. The results of this are shown in Fig. 7. By examining the mass flow rate before and after the application of PTFE tape we can see that the stability improves. With 2 pieces of tape the valve begins operating like a sophisticated controller rather than an on/off control. The reason for this is that the PTFE tape changes the resistance term of the time constant of the bulb and reduces its sensitivity. However instead of tending to operate as a proportional controller the additional PTFE tape causes the sensor to average its output and the valve begins to operate with integral control.

Valve sensitivity is also dependent on the time constant of the heat exchanger, the fluid flow lag between the valve and the superheat sensing point, and the damping characteristics of the valve. The evaporator time constant and the fluid flow length were fixed by the geometry of the existing design. However, the damping characteristics and bulb time constant were an inherent feature of the valve type and therefore could be changed if a different valve type was used. A study identified two valve alternatives:

1. The Parker TEV model CAVW was selected for its stability. The suppliers claimed that as a ball valve with much greater movement over its full range it has much greater stability than the alternative needle type mechanism. The bulb size was similar to the TX2 valve although the full range capacity was slightly higher at 1.6 kW. The results were disappointing and the valve responded unstably under all conditions.
2. A prototype Danfoss TUAE valve was also obtained with a much closer capacity to that required. The valve was fitted with a 'O2' orifice, giving a capacity of 1.15 kW. The valve also had a considerably smaller bulb giving a reduced time constant compared to the TX2 valve and consequently faster response time. The valve had an external equalising connection and although the pressure drop across the evaporator was not excessive, this connection helped to realise a lower superheat value. The results with this valve are shown in Fig. 8, and although the valve still modulates, the degree of modulation is much reduced with a superheat of 9 K (±1 K) being maintained.

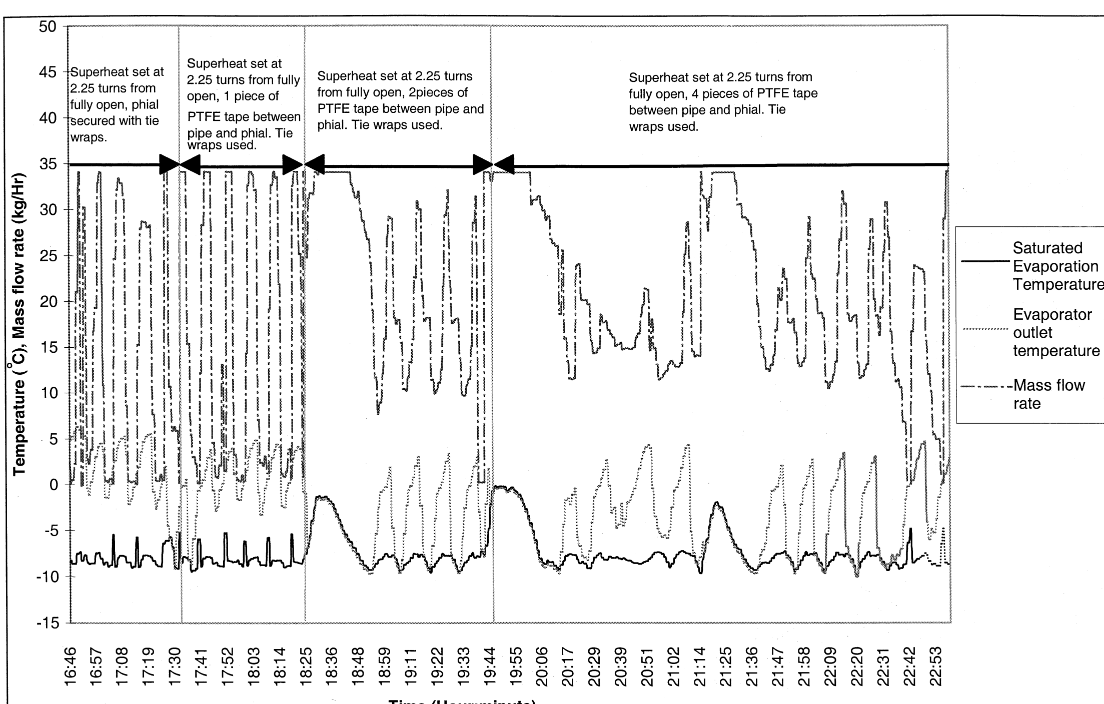

Fig. 7. Effect of thermal resistance on valve stability.

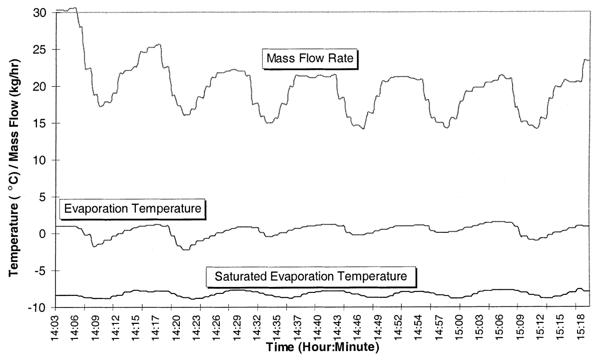

Fig. 8. Operation of the Danfoss TUAE expansion valve.

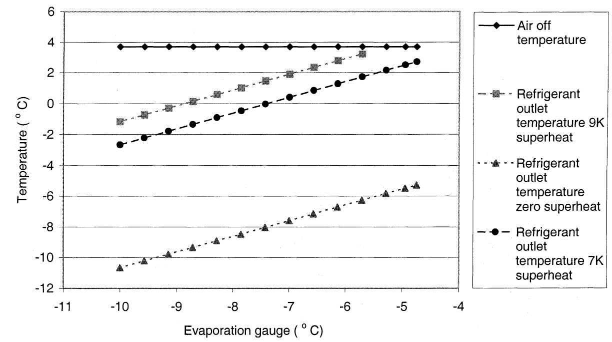

Fig. 9. Graph showing the refrigerant outlet and air inlet conditions for zero and positive superheat devices at different evaporating pressures.

The stability of the valve could be further improved by reducing the response time of the valve by using a bulb with a smaller time constant or reducing the fluid flow lag between the valve and the superheat sensing point. This could be achieved by either changes to the coil geometry or to the damping characteristics of the valve.

4. Alternative expansion devices for the delicatessen

Whilst controlling superheat in this way has proved satisfactory in many other applications, as previously discussed this device underutilizes the evaporator surface area and therefore results in a lower evaporating temperature, apparatus dew point and leaving air humidity than other devices.

A separate theoretical study showed that this device constrained the maximum evaporating temperature that could be used and it therefore limited the maximum humidity that may be achieved within the display area. This theoretical study was carried out using a computer based mathematical model, which has been described in detail elsewhere (Maidment, 1998). This programme was written in Engineering Equation Solving software (EES, 1996), to enable the delicatessen cabinet design to be optimised and it allowed the implications of a number of design variables to be studied.

Using this model the implications of different heat exchanger geometries and surface areas on cabinet operating cost were investigated. This showed that the maximum evaporator surface area and evaporating temperature was constrained by the superheat required to maintain the stability of the TEV. This can be seen in Fig. 9, which shows that with a superheat of 9 K the existing TEV system constrains the maximum evaporating temperature that may be achieved to approximately −5.5°C, since the refrigerant outlet temperature approaches the air off temperature. This would limit the maximum relative humidity with this system to approximately 65%. This Figure also shows that even if the superheat required to maintain stability is reduced, the evaporating temperature is constrained below −5.0°C.

For this reason a review of expansion device types was carried out to identify if a more appropriate device existed for this application. The valve options considered included the capillary tube, short tube restrictor, the fixed orifice, the constant pressure expansion valve, HP & LP float valves, a number of types of TEV and 3 types of electric expansion valves. A comparison of these is shown in Appendix A. This indicates that the capillary tube, short tube restrictor and thermistor type electric expansion valve (TEEV) are preferable to a TEV for high heat transfer as they will maintain a minimum of superheat. Unlike the TEV operated system, these expansion devices do not have to operate at a low evaporating temperature to provide a sufficient temperature difference for stable superheat. This is shown in Fig. 9. Indeed, the evaporating pressure is constrained only by the cooling load, and consequently, these alternative devices will allow heat transfer to be optimised and higher humidities to be achieved. In addition, they will also make better use of the evaporator surface area, as no superheating section will be required.

Whilst previous experimental work (Deng, 1991) has shown TEEV's to control effectively the refrigerant quality near to the saturation condition for direct expansion evaporators. The capillary/short tube restrictor (or "choked flow restrictor") options are preferable to the TEEV as they are a simpler, cheaper and are a passive means of expansion and maintaining dry gas at the compressor suction. To demonstrate the suitability of this type of expansion device a separate investigation was carried out and this is described in Section 5.

5. The suitability of choked flow restrictors

5.1. Description and operation of choked flow restrictors

Choked flow restrictors are normally applied in critically charged refrigeration systems such as domestic refrigerators, and they are not known to have been applied to a central supermarket refrigeration system. The two forms of choked flow restrictor are the capillary tube and the short tube restrictor.

5.1.1. Capillary tube restrictors

A capillary tube is typically 1–6 m long with an inside diameter generally from 0.5 to 2 mm (Stoecker & Jones, 1988). Liquid refrigerant enters the capillary tube, and as it flows through the tube, the pressure drops due to friction and acceleration of the refrigerant. Some of the liquid flashes into vapour as the refrigerant flows through the tube.

The capillary relies on the principle that liquid refrigerant passes through it more readily than vapour. The refrigerant mass flow rate through a short tube is therefore strongly dependent on the upstream pressure and sub-cooling. Flow rate will also increase with lower outlet pressure down to a critical value, below which the flow rate does not change. The critical value occurs where sonic velocity is achieved at the exit, so that if the capillary tube is selected for a choked flow condition then the flow rate will actually be independent of the saturated evaporating temperature. This produces a significant advantage over alternative fixed restrictors which are dependent upon the outlet pressure.

5.1.2. Short tube restrictors

In recent years, short tube restrictors have become widely used in commercial air conditioning units. They offer the same advantages of low cost, high reliability and the choked flow phenomenon as the capillary alternative, however they have the added advantage of ease of inspection and replacement. They are also more compact than capillary tubes and are typically 10–13 mm in length, with a length-to-diameter (L/D) ratio of between 3 and 20. Whereas capillary tubes have an L/D ratio much greater than 20.

There are generally two basic designs for short tube restrictors: stationary and moveable. The latter consist of a piston, which is moveable within a housing and is shown in Fig. 10. A moveable short tube restrictors offers restriction for the refrigerant flowing in one direction, however in the opposite direction the restriction is reduced as the piston moves away from its seat allowing greater flow to be achieved. In the cabinet either fixed or moveable designs could be fitted depending on the defrost system used. If electric defrost is utilised the fixed restrictor would be suitable, whereas if hot/cool gas defrost is used then the moveable type would allow greater gas flow rates reducing defrost times.

5.2. Application of choked flow restrictors in the supermarket

To be applied successfully the choked flow restrictors must meter the refrigerant correctly under all load conditions. This means that they must be sized to ensure that the vapour at the compressor suction is superheated under all load conditions. However they must also meter sufficient refrigerant to satisfy the cooling load under steady state and pull down following defrost.

To assess their suitability, their performance under part and full load conditions has been theoretically investigated based upon a typical Sainsbury's store as described in Section 5.3.

5.3. Refrigeration system description – typical supermarket

The refrigeration systems currently installed in the supermarket utilise central systems, a schematic of this is shown in Fig. 11.

In the supermarket the food is grouped by product storage temperature into chill and frozen food categories and separate central refrigeration systems are provided for each. Normally to minimise the risk of failure there are two individual central systems serving each of the food groups. With this arrangement each system will consist of up to four compressors which are operated in parallel as a package unit with a liquid receiver and multi-station manifolds for individual liquid, suction and hot/cool gas connections. Each package includes a

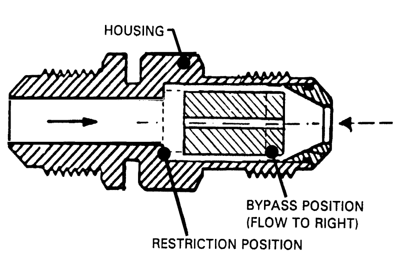

Fig. 10. Short tube restrictor.

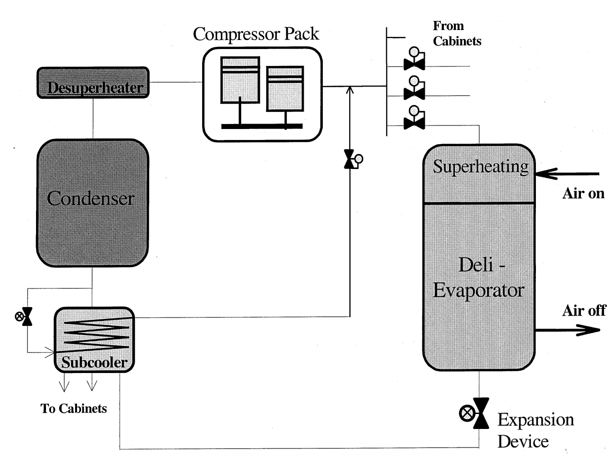

Fig. 11. Diagram of delicatessen refrigeration system model.

hot gas de-superheater for heat recovery and a flash type liquid sub-cooler. The degree of sub-cooling is controlled by a TEV and the evaporated refrigerant returns to the compressor suction via an evaporator pressure regulator (EPR), whose function is to control a constant evaporator pressure.

In the typical supermarket the delicatessen cabinets are connected to chill system no. 1. The cabinets, their design cooling loads and evaporating temperatures operating on chill system no. 1 are shown in Table 2.

The chill temperature compressor packs are connected on site to heat reclaim coils which are in parallel for warm air heating and to a remote air cooled condenser. The condensing pressure is controlled by fans operated in stages and with star/delta speed control. Also, it is assumed that there is a significant pressure drop and temperature rise in the suction line even though the pipe is insulated. As a result the gas superheat at the compressor suction is approximately 20 K at the design condition.

5.4. Analysis of suitability of short tube restrictors

In order to assess the suitability of the short tube restrictor it was first necessary to select a restrictor to satisfy the cooling duty requirements under the minimum flow condition. For a restrictor designed for choked flow operation the minimum flow condition will in principle occur at the minimum condenser saturation temperature and minimum sub-cooling. As these conditions may not necessarily coincide, the initial investigation aimed to establish the minimum flow condition.

For the typical supermarket the minimum condenser saturation temperature will occur at the setting used for the head pressure control which is typically 30°C. The minimum sub-cooling was determined by examining the energy balance around the sub-cooler, as shown in Fig. 12, where

$$Q = \dot{m}_b \Delta H_b = \dot{m}_s C_p (T_i - T_o) \approx UA((T_i/2 + T_o/2) - T_b). \quad (2)$$

By assuming that the UA and C_p are constant irrespective of liquid temperature, this equation can be reduced to

$$\text{Sub-cooling} = (T_i - T_o) \approx ((T_i/2 + T_o/2) - T_b)/\dot{m}_s. \quad (3)$$

As the boiling temperature (T_b) is fixed by the EPR and the subcooled mass flow rate ($\dot{m}_s$) has been shown by other researchers not to fluctuate significantly in similar applications (Energy Efficiency Office, 1994), sub-cooling can be shown to be primarily dependent on the refrigerant temperatures (T_i and T_o). Thus if these temperatures are high which will occur at high condenser pressures, then the subcooling will be high. It can also thus be stated that the minimum condenser pressure and minimum sub-cooling are coincidental and will produce the minimum flow condition. Substitution of real values into the above equation shows that a minimum sub-cooling of 10 K occurs at the minimum condenser pressure.

To establish the short tube restrictor selection it was also necessary to estimate the mass flow required to satisfy the cooling load requirements in each delicatessen cabinet. As it was undecided whether the cabinet would be operated with or without evaporator defrost and both options would require different mass flow rates it was necessary to analyse these cases separately.

In case 1, it was assumed that the evaporator did require defrosting and the short tube restrictor was selected to meter the same refrigerant flow rate observed from the experimental test plant following a defrost cycle.

In case 2 it was assumed that the delicatessen cabinet does not require defrosting and therefore the mass flow rate of refrigerant required must satisfy the cooling load at the minimum delivery pressure and at the lowest subcooled condition.

Table 2
Breakdown of the cooling loads for the chill system no. 1

No. off	Description	Total duty (kW)	Saturated evaporating temperature (°C)
1	Meat chiller	2.2	−6
1	Provisions cold store	3.67	−5
1	Produce cold store	4.77	+3
2	Fish multi-deck	9.27	−8
1	Fish counter	1.312	−10
2	Meat counter	2.012	−10
1	Bakery serve-over	1.1	−5
1	Cream cake multi-deck	2.4	−8
4	Fresh meat multi-deck	15.2	−8
5	Fresh meat multi-deck	23.22	−8
2	Dairy multi-deck	8.06	−8
2	Milk roll in	11.84	−10
2	Fresh meat multi-deck	8.47	−8
4	Delicatessen cabinets	2.82	−5
3	Produce multi-deck	15.24	−7
	Total	111.6	

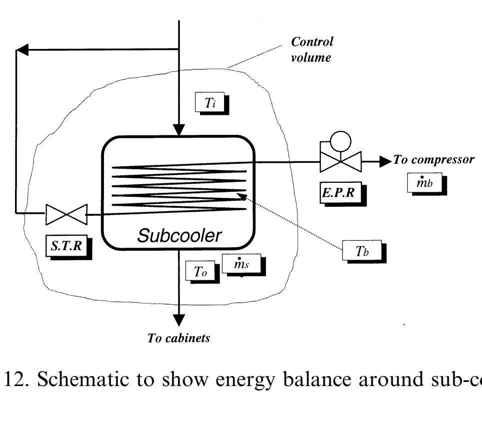

Fig. 12. Schematic to show energy balance around sub-cooler.

Using a computer program developed by James (1995), short tube restrictors were selected for the delicatessen cabinet to achieve 0 K superheat at the refrigerant outlet condition at the minimum flow condition. As the refrigerant states used for selection in both cases were similar, their cycles may be represented by the same pressure enthalpy chart, as shown in Fig. 13. From this it can be clearly seen that whilst the refrigerant condition leaving the delicatessen cabinets is saturated the refrigerant condition at the pack suction is significantly superheated. This is because the saturated refrigerant gas returning to the compressor package from the delicatessen cabinet gains heat through the suction line and is mixed with superheated refrigerant returning from other cabinets and the sub-cooler.

To assess the viability of these selections it was necessary to ensure that under the maximum flow condition the refrigerant entering the compressor suction was in a superheated state. The maximum flow condition was established by applying the same principles used in determining the minimum flow condition. This established that the maximum flow occurred at the highest condenser pressure and sub-cooling. The values determined were 45°C and 22 K, respectively. Under these operating conditions the mass flow rate passing through the restrictor was recalculated for each case using the selection programme and the values used are shown in Table 3.

The effect of these flow rates on suction superheat at the compressor pack were analysed using a mathematical model of the refrigeration system, detailed previously in Section 4.

5.5. Results of investigation

The solution to the case 1 is shown on the P–H chart in Fig. 14. From this it can be seen that although the refrigerant condition at the delicatessen cabinet outlet is wet, the refrigerant at the pack suction is significantly superheated. This is partly due to heat gain through the suction line insulation and also because the refrigerant is mixed with superheated gas returning from other cabinets and the sub-cooler. There is no reason why this should not be acceptable in the supermarket since liquid flood back with TEV systems is already tolerated (Lawrence & Lawson, 1997). This occurs because the TEV bulb heats up during the off cycle and thus when the refrigerant flow restarts, the valve opens fully. Whilst liquid refrigerant leaves the evaporator, the thermal capacity of the suction line is high enough to ensure that dry gas exists at the suction. The refrigerant conditions for case 2 were similar to those for case 1, although to avoid defrost entirely it was necessary to assume that the evaporator operated at a saturation temperature of 3°C.

Table 3
Mass flow rates through restrictor under minimum and maximum flow conditions

Case	Mass flow rate (kg/s) at maximum flow condition	Mass flow rate (kg/s) at minimum flow condition
1	0.02828	0.01571
2	0.04755	0.01585

The results of the two ‘choked flow restrictor’ cases were compared with the existing TEV solution. These are shown in Table 4.

From this it can be seen that in both cases the final refrigerant condition at the compressor pack has significant superheat, and therefore both cases would be acceptable. Indeed, further analysis showed that if defrost could be avoided other cabinets could be safely operated with choked flow restrictors.

Table 4 also shows the comparative COPs at full and part load, and the full load compressor swept volume required for each case. As COP and capacity are unaffected by the choice of expansion device short tube restrictors may be effectively used to optimise humidity within the delicatessen cabinet.

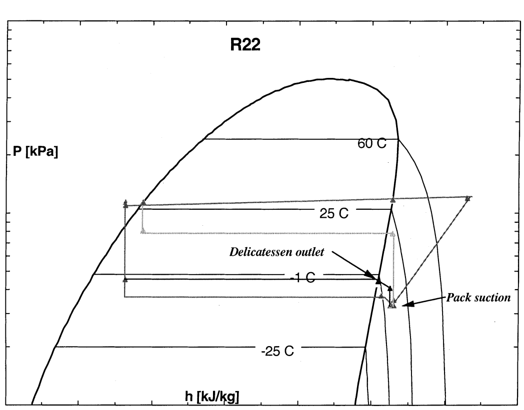

Fig. 13. The design case for a choked flow restrictor shown on the pressure enthalpy chart.

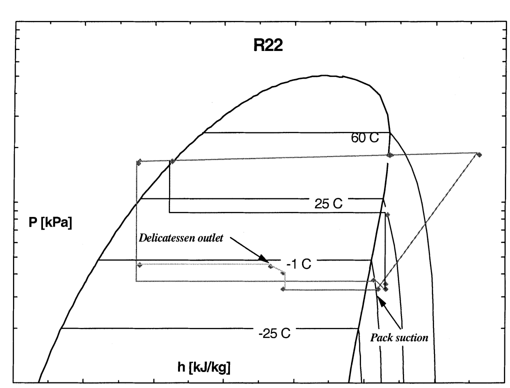

Fig. 14. The design case for a choked flow restrictor shown on the pressure enthalpy chart.

Table 4
Results of performance investigation for choked flow restrictors

	Existing design	Restrictor case 1	Restrictor case 2
Defrost required in delicatessen cabinet	Yes	No	Yes
Evaporating temperature at delicatessen	−7°C	−3°C	−7°C
Pack suction superheat – full load	22 K	21.3 K	21.2 K
Pack suction superheat – part load	21 K	17.2 K	11 K
Compressor swept volume	0.725	0.725	0.725
COP at minimum pressure ratio	3.245	3.246	3.246

5.6. Integration of additional controls

The short tube restrictor may be integrated into the existing cabinet design and operate in conjunction with a liquid line solenoid maintaining cabinet output in response to changes in air temperature with on/off control. However, the use of a more sophisticated control such as a stepper valve or pulse width valve may be a beneficial addition to the choked flow restrictor since a modulating valve will indirectly control the refrigerant quality leaving the evaporator during steady state cooling. To avoid the necessity of fitting both the short tube restrictor and control valve on site, manufacturers could incorporate a short tube restrictor into their existing valve designs.

Furthermore, an additional safety cut out could be fitted to the common pack suction as a fail safe device. In the unlikely event of the common suction superheat falling below 5 K, the central monitoring system could close either a solenoid or a modulating valves until a higher superheat were reached.

6. Conclusions

The TEV is the traditional expansion device used in the supermarket. However, when operated on a standard delicatessen display cabinet the TEV is unstable because it is unable to respond to superheat changes in time. Whilst improvements to its stability maybe achieved the TEV has been shown to be an ineffective and inefficient means of expansion device.

The alternative expansion devices for the delicatessen cabinet have been reviewed, and this showed that a choked flow restrictor would provide the simplest, most efficient and lowest cost solution. A model of the supermarket refrigeration system model has been developed and used to show that a choked flow restrictor maybe theoretically successfully applied in the supermarket to satisfy part load and full load conditions. This will realise a higher humidity within the delicatessen cabinet which will result in lower food weight loss and consequently higher profits for the retailer.

A more sophisticated control incorporating a stepper valve or pulse width valve has been proposed. It may be a beneficial addition to the choked flow restrictor and to avoid the necessity of fitting both on site, it is suggested that valve manufacturers could incorporate a short tube restrictor into their existing valve designs.

Finally, the short tube may allow lower energy consumption as higher evaporating temperatures may allow defrost to be avoided.

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Appendix A. Comparison of possible expansion valves

Type	Will it work at design conditions	Will it work at part load conditions	Advantages and disadvantages
Capillary tube	(a) The capillary tube would be ideally suited to this application if defrost can be avoided. It may then be sized for steady state conditions. (b) If defrost is necessary it may still be applicable but the capillary would be sized	(a) Without defrost – one of the features of capillary tubes is if they are sized for 'choked flow' at a part load suction pressure they will deliver the same mass flow rate should the suction pressure fall. The mass flow rate will remain constant irrespective of suction pressure. As delivery pressures will vary	*Advantages* (1) Low cost; (2) low maintenance; (3) ease of commissioning; (4) no superheat zone required; (5) evaporating temperature is not limited by superheat and therefore high humidities are possible; (6) there

Type	Will it work at design conditions	Will it work at part load conditions	Advantages and disadvantages
	to avoid compressor slugging but achieve pull down following defrost.	however, to be acceptable, the capillary must ensure that the refrigerant is well into the superheated zone before entering the compressor suction. (b) With defrost – whilst the operation of the tube will be similar to the non-defrost case the tube will be sized for pull down operation. This will make the steady state selection more critical to avoid liquid return to the compressor.	will not be an energy penalty at low loads as pack suction pressure is controlled independently of evaporating pressure. *Disadvantages* (1) Capillary likely to clogg; (2) difficult to remove foreign body following blockage.
Short tube restrictor	Essentially the short tube restrictor and the capillary are very similar in operation.	The same design issues for the capillary are relevant to short tube restrictors.	*Advantages* (1) Low cost; (2) low maintenance; (3) ease of commissioning; (4) no superheating zone required in the evaporator; (5) evaporating temperature is not limited by superheat and therefore high RH is achievable.
Fixed orifice	This expansion device will operate in a similar way to both capillary and short tube restrictor.	The disadvantage compared with the other restrictors is that the fixed orifice can not achieve a choked flow condition and therefore its capacity will vary with changes in both upstream and downstream pressure.	*Advantages* (1) Low cost; (2) low maintenance; (3) ease of commissioning; (4) no superheating zone required in the evaporator; (5) evaporating temperature is not limited by superheat and therefore high RH is achievable. *Disadvantages* (1) Orifice is likely to clogg.
Constant pressure expansion valve	This valve regulates the refrigerant mass flow entering the evaporator to maintain a constant evaporator pressure. The valve is only applicable to constant load situations.	Not suitable – At part loads it would tend to open and at high loads it would close. As a result it would not be suitable for use in the delicatessen cabinet.	Not suitable
HP and LP float valves.	HP floats operate on critical charge systems, which do not occur in the supermarket. Although the LP float system charge is not critical it is unlikely to be a cost effective solution as each evaporator would require a liquid level control pot (£ 500+)	Not suitable	Not suitable
TEV- straight liquid charged and TEV- crossed liquid charged	With this valve, the bulb always controls the valve operation even when a colder valve head or capillary tube. This is because the liquid charge is such that the bulb will always contain liquid. The minimum superheat the straight liquid valve can control increases with decreasing evaporating temperature. This can increase the exchanger surface required. However, at −5°C evaporating temperature, superheats of 5 K should be possible. The cross-charged valve is said to control lower superheat and has anti-hunting characteristics.	Stable operation should result if valve is sized for both full and part load scenarios.	*Advantages* (1) Low cost familiar technology. *Disadvantages* (1) Compared to passive expansion systems, the TEV will require greater evaporator surface for superheating; (2) with this form of superheat control the maximum evaporating temperature possible will be −4°C which will restrict RH to <70% (at 3°C air temperature); (3) slightly increased capital cost.
TEV- adsorption charged	Adsorption charged valves are unaffected by the surrounding environment of the valve. They have a comparatively slow response time and are said to control stable superheats of 4 K.	Stable operation should result if valve is sized for both full and part load scenarios.	*Advantages* (1) Stable operation? *Disadvantages* (1) Compared to passive expansion systems, the TEV will require greater evaporator surface for superheating; (2) with this device the maximum evaporating temperature is −4°C. This restricts RH to <70%

Type	Will it work at design conditions	Will it work at part load conditions	Advantages and disadvantages
			(at 3°C air temperature); (3) slightly increased capital cost.
Electric expansion valves – heat motor type.	This type of valve responds to the output from a bimetal heater in the evaporator outlet. Such that if the gas is superheated the thermistor opens the valve until the gas becomes saturated. A condition close to saturation can be controlled. In close-coupled systems the valve may fail to eliminate the risk of some liquid droplets returning to the compressor, however this is very unlikely in the supermarket due to large natural superheat.	Stable operation should result if valve is sized for both full and part load scenarios.	*Advantages* (1) Small superheat zone required in the evaporator; (2) evaporating temperature is not limited by superheat control and therefore high RH is achievable. *Disadvantages* (1) Slightly increased capital cost.
Electric expansion valves – pulse width type.	Valve is fixed orifice with a pulsed on/off solenoid valve. Valve controls a temperature differential between the evaporator saturation temperature and the gas temperature at the exit, and can control superheat of approximately 4 K.	Stable operation should result if valve is sized for both full and part load scenarios.	*Advantages* (1) Valve does not require a solenoid valve. *Disadvantages* (1) Compared with passive systems, the TEV will require greater surface for superheating; (2) with this device the max evaporating temperature possible is −3°C which restricts RH to <75% (at 3°C air temperature); (3) slight increase in capital cost.
Electric expansion valves – stepper motor type.	A stepper motor is used	As pulse width type	As pulse width type

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