A Real Time Control Strategy for Optimisation of an Economised Indirect Multi-Temperature Transport Refrigeration System

Shane Smyth
University College Dublin

Donal P. Finn
University College Dublin

Barry Brophy
University College Dublin

A REAL TIME CONTROL STRATEGY FOR OPTIMISATION OF AN ECONOMISED INDIRECT MULTI-TEMPERATURE TRANSPORT REFRIGERATION SYSTEM

S. Smyth, D.P. Finn* and B. Brophy.

School of Electrical, Electronic and Mechanical Engineering,
University College Dublin, Belfield, Dublin 4, Ireland
*Tel: +353 1 716 1947 email: donal.finn@ucd.ie

ABSTRACT

This paper describes an approach for control of an economiser cycle based on the use of economiser pressure as the primary control parameter. In the study, the economiser cycle was used to optimise a multi-temperature indirect (IDX) transport refrigeration system, where hydronic secondary loops were utilised. In transport refrigeration applications, IDX systems can offer the potential to address a number of important environmental and control issues associated with direct expansion (DX) systems. IDX systems may also give rise to reduced capacity and COP through increased compressor pressure ratios associated with the hydronic secondary circuit and power requirements of the liquid secondary pumps. One approach by which this issue can be addressed is through use of an economiser cycle, which provides a mechanism for performance enhancement by augmenting the refrigeration effect of the primary refrigerant, in the primary to secondary heat exchanger of these systems. Previous work ascertained that by control of the mass-flow injection ratio, an economiser cycle can be used to optimise indirect multi-temperature systems for a wide range of diverse operating conditions. This method of control necessitates mass-flow instrumentation which is impracticable for field applications. An alternative method of control described here, is based on a more easily measured economiser pressure, thereby eliminating the requirement of mass-flow instrumentation.

1. INTRODUCTION

Multi-temperature transport refrigeration systems can consist of up to three separate compartments, incorporating independent control of each compartment set-point temperature, thereby facilitating simultaneous transport of multiple cargos. Multi-temperature systems can within the cold-chain, offer the potential to reduce transit journeys, operating costs and environmental emissions (Tassou et al., 2009). At present, direct expansion of HFC refrigerants in individual remote evaporators are exclusively used in multi-temperature systems. These systems frequently operate under non-uniform, non-design conditions, thereby giving rise to pressure control issues. Furthermore they require large refrigerant charges with consequent leakage risks. Despite considerable developments in multi-temperature systems over the last decade, increasing environmental awareness coupled with ongoing legislative developments concerning the use of HFC refrigerants, are expected to challenge the continued use of these systems in the coming years.

Indirect (IDX) systems have been suggested as an alternative technology to direct expansion refrigeration. These systems have received considerable interest in a number of stationary applications such as industrial refrigeration applications (Rivet, 2003), ice rinks and supermarket systems (Hinde et al., 2009). IDX systems consist of a compact sealed chiller unit, which cools a secondary heat transfer liquid circulating through individual fan-coil units. The secondary heat transfer liquid can either be an environmentally benign single or multi-phase fluid to facilitate heat transfer between conditioned space and chiller. The presence of a heat exchanger between primary and secondary loops contributes to an additional temperature difference which can result in a reduced evaporator pressure relative to the DX baseline. Under these conditions, the IDX system may give rise to capacity and COP penalties. As a result, it has previously been found that indirect system optimisation is often necessary for comparable performance with a DX baseline (Terrell et al., 1999). A number of component and system optimisation measures have been proposed in the literature. Heat exchanger optimisation for supermarket systems has been discussed by Haglund-Stignor et al. (2007). Similar approaches have concerned optimisation of pumps (Kazachki and Hinde, 2006), secondary coolants (Melinder, 1997; Aittomaki and Lahti, 1997; Sawalha and Palm, 2003) and flow regimes (Clarke and Finn, 2008). Optimised single temperature IDX systems have resulted in comparable performance and reduced environmental impact relative to baseline DX systems (Horton and Groll, 2003).

Indirect systems have been analysed in transport refrigeration applications where they have been deployed in multi-temperature applications (Smyth et al., 2010). Multi-temperature transport refrigeration systems operate under a wide range of ambient and set-point temperatures resulting in additional challenges compared with single temperature stationary systems. Indirect systems are potentially suitable for operation with multi-temperature transport refrigeration applications. However, the wide range of ambient and set-point temperatures encountered by these systems gives rise to capacity and COP penalties under certain operating conditions, whilst comparable performance at other operating conditions is achievable (Smyth et al., 2007). It has been shown elsewhere by the authors, that the inclusion of an economiser cycle (Fig. 1) can be used to augment capacity and COP in indirect systems, even across a wide range of ambient and set-point temperatures, to give comparable performance under certain conditions with baseline DX systems. The economiser cycle must however be optimised to operational conditions by regulation of mass-flow injection ratio. It was previously found that an optimum injection ratio exists for different combinations of ambient and set-point temperatures in a multi-temperature indirect transport refrigeration system (Smyth et al., 2010). Modulation of the injection ratio, by control of the economiser refrigerant mass flow rate, allows maximisation of capacity and COP. However this method requires the use of massflow instrumentation, which is generally impracticable for field installations. This paper describes therefore an alternative method for control of economiser cycle, based on use of the economiser pressure as the control parameter by which system capacity and COP can be optimised.

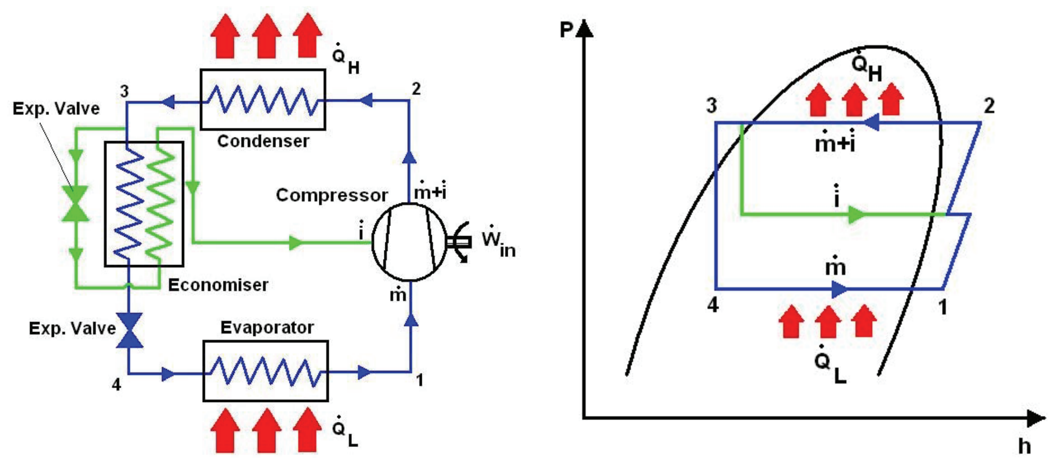

Figure 1. Economiser Cycle. (Scale not representative of actual cycle examined).

2. APPROACH

It has been shown elsewhere, that economiser cycles can be used to optimise indirect systems for multi-design conditions (Smyth et al, 2010). Economiser cycles can enhance the refrigeration effect of the primary refrigerant in the primary to secondary heat exchanger, if appropriate mass flowrates of refrigerant are expanded through the economiser circuit. Control of the injection ratio at optimum level requires the use of massflow instrumentation, which is generally impracticable for field installations. Moreover the deployment of instrumentation, where noise-vibration and harshness (NVH) frequently arises militates further against direct mass flow measurement, therefore an alternative method of system control is required. Rugged pressure measurement instruments are considered to be more suitable for this application, thereby motivating the need for an alternative control strategy to facilitate deployment of a pressure control system and eliminating the need for refrigerant mass-flow measurement.

It was previously found that IDX system performance (capacity and COP) can be independently optimised by modulation of the mass injection ratio. To facilitate development of an algorithm based on pressure control, the relationship between economiser pressure and injection ratio must be determined. In this paper, experimental investigations were conducted to examine the relationship between economiser pressure and injection ratio (INJR) in a multi-temperature indirect refrigeration system. The individual economiser pressures for maximising capacity and COP for each individual ATP test point were determined (ATP, 2003). The influence of evaporator and condenser saturation conditions on optimum economiser pressure were examined by experimental evaluation of individual parameters. These individual optima permitted identification of key trends. The control equations were determined by curve fitting to the experimental data representing the optimum economiser pressure for capacity and COP maximisation. The control equations facilitated development of a real-time control algorithm for implementation into the economised IDX system. Subsequent experimental evaluation with the control algorithm show that the proposed algorithm provides equivalent performance relative to a mass-flow controlled economised cycle. Furthermore, it was found that the control algorithm can be used to maximise capacity and COP of the IDX system without the requirement for mass-flow instrumentation.

3. EXPERIMENTAL

The experimental apparatus, shown in Fig. 2 is described in detail elsewhere (Smyth et al., 2010). All tests were carried out with reference to ATP specification for a Class C refrigerated vehicle (ATP, 2003). A commercial multi-temperature DX installation charged with R404A was installed alongside the prototype indirect system. For comparison purposes, a common condensing unit utlising a water cooled condenser was deployed. A vapour injected economiser cycle was installed into the primary cycle of the prototype indirect refrigeration system. A scroll compressor was installed with pressure transducers on the suction (Psuc), discharge (Pdis) and economiser (Pecon) lines. The prototype indirect system incorporated a single phase hydronic secondary loop. The primary cycle economiser cycle consisted of a subcooling heat exchanger, auxiliary expansion valve and associated pipework. Refrigerant flow modulation was implemented via a stepper motor expansion valve. HFC-R404A was deployed in the primary circuit, and a 50% V/V aqueous ethylene glycol solution was utilised as a secondary coolant. An electronic stepper-motor expansion valve was utilized on the economizer circuit and was interfaced to a computer-based controller, thereby enabling implementation of the control algorithms. Condenser conditions were maintained by means of a PID controlled heat unit. Compressor speed was controlled at 50 Hz for all tests and fans and pumps were operated at rated speed. Instrument calibration is carried out in accordance with ASHRAE (2005).

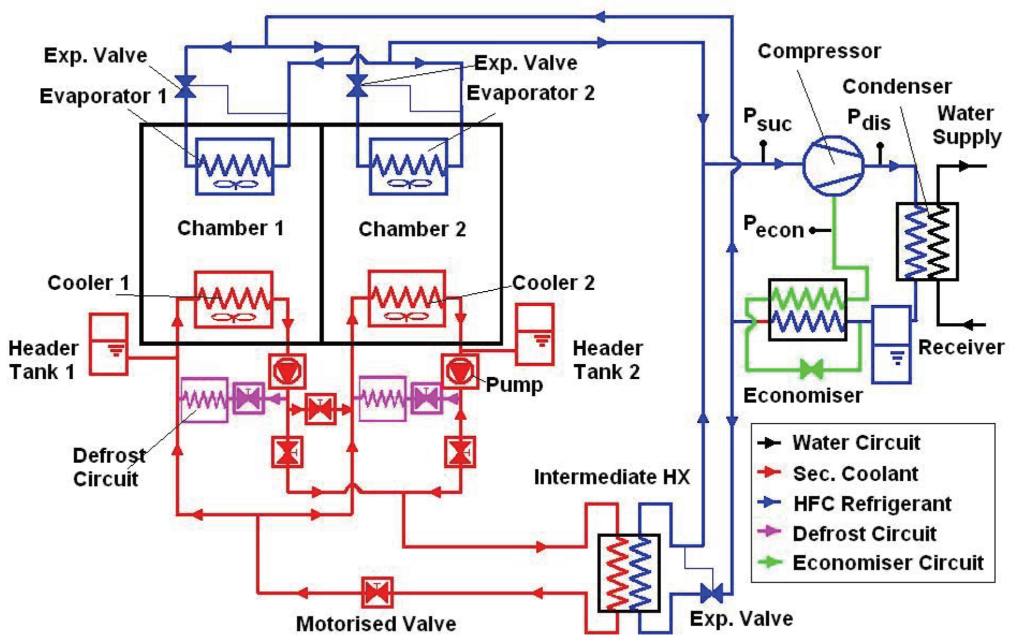

Figure 2. Test Rig Schematic with Economiser Circuit

4. TEST MATRIX AND EXPERIMENTAL APPROACH

Side by side testing of the algorithm controlled and the manually controlled economised system was investigated at ATP conditions of -20,-20°C, -10,-10°C and 0,0°C, where both chambers were supplied with secondary coolant in parallel. Chamber and condensing conditions were maintained to within ±0.5°C for the duration of the tests. Each ATP condition was examined for three individual condensing conditions of +18°C, +22°C and 25°C (condenser water inlet temperature). The performance of the system was studied for injection ratio increments of 5% from zero injection ratio to the maximum possible injection ratio. The economiser pressure was recorded for each injection ratio increment and the optimum economiser pressures recorded with respect to suction and discharge pressures, as well as condensing and set-point temperatures.

5. RESULTS

Optimum injection ratios exist for both capacity and COP and these appear to be dependent on chamber set-point and condenser boundary conditions. The relationship between economiser pressure and injection ratio is shown in Fig. 3. The economiser pressure is noted to increase with increasing mass flowrate of injected economiser refrigerant. The minimum pressure for each operational condition when no economiser action is present (0% INJR rates - Fig. 3) is defined as the saturated injection pressure (SIP), which is measured using the economiser pressure transducer. A saturated injection temperature (SIT) for each individual SIP value can also be determined. Closer analysis revealed that both the SIP and SIT values were dependent on the compressor operating condition in addition to the evaporator and condenser boundary conditions. In this work, the SIP was found to be strongly dependent on the compressor suction pressure, due to the close proximity of the economiser injection ports to compressor suction port (see Fig. 4). The SIP was measured experimentally for different combinations of ambient boundary conditions and setpoint conditions and a regression line was fitted as shown in Fig. 4.

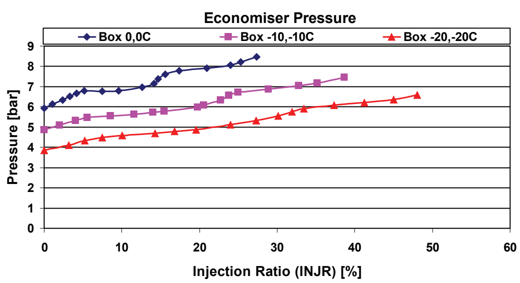

Figure 3. Economiser Pressure

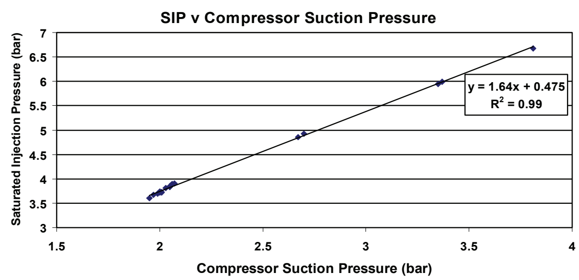

Figure 4. Saturated Injection Pressure

Examining Fig. 5, it was found that a unique economiser pressure exists for each ATP test point at which the evaporator capacity is maximised. For the -20-20°C condition, a steady increase in evaporator cooling capacity relative to economiser pressure is noted up to a peak of approximately 6 bars. The decrease thereafter results from high refrigerant saturation temperature (at higher pressures) and its influence in reducing the subcooling effect. Capacity maximisation is therefore achieved by maintaining the injection pressure at 6 bars. Similar trends are evident for -10,-10°C and 0,0°C

The evaporator COP is illustrated in Fig. 6. It can be that as the injection ratio was increased up to its maximum value, that an initial increase in COP was achieved, followed by a decrease and then an increase again. Compressor power was found to increase with injection ratio (Smyth et al, 2010), therefore the initial COP improvement arises from greater enhancement in capacity relative to compressor power. As injection ratio is increased further, the influence of increasing compressor power results in a reduction in COP. Further capacity augmentation relative to compressor power results in a second peak in COP. Referring to Fig. 6, it can be seen that a choice of two COP optima exist. The low-pressure optima are useful for minimisation of compressor power, whereas the high pressure optima are necessary where high capacity is required.

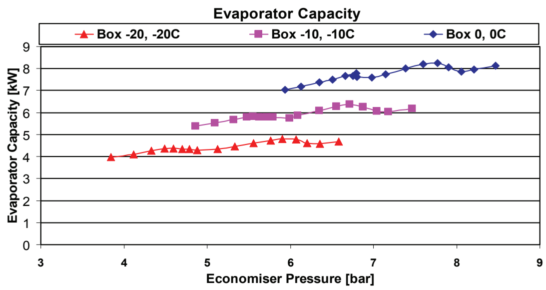

Figure 5. Evaporator Capacity

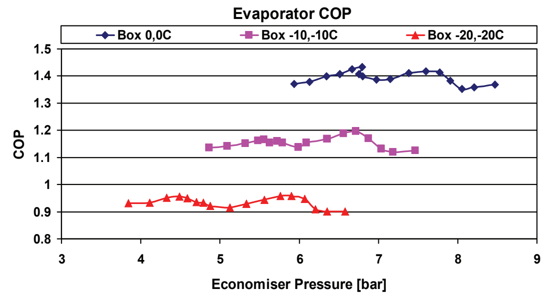

Figure 6. Evaporator COP

6 ALGORITHM DEVELOPMENT

Maximisation of system capacity or COP, as shown in Figs. 5 and 6, requires knowledge of the optimum economiser pressure, which is a function of the saturated injection pressure (SIP). As was shown in Fig. 4, a correlated relationship can be established between the SIP value and the compressor suction pressure. Figs. 7 and 8 show that the economiser pressure required for optimisation of capacity or COP can be correlated with the saturated injection pressure. Depending on the requirement of optimum capacity or optimum COP, the appropriate correlation can be used as a control equation for the economiser pressure. Therefore, for control purposes, the SIP is first determined, which is then used to determine the optimum economiser pressure depending on the requirement for maximum capacity or COP.

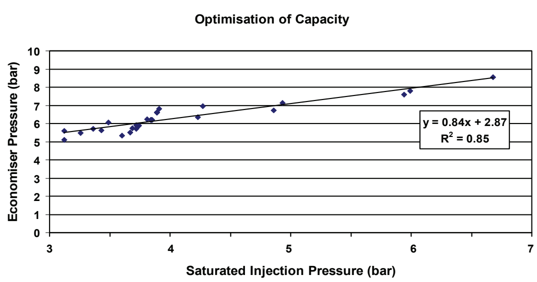

Figure 7. Control Equation for Optimum Capacity

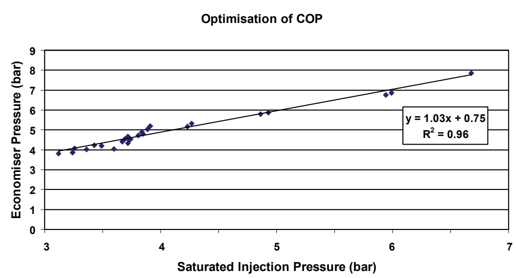

Figure 8. Control Equation for Optimum COP

A software-based algorithm was developed using LabVIEW and incorporates the regression equations shown in Fig. 4, 7 and Fig. 8 (NI, 2009). This algorithm consists of a single-input, single output control loop, coupled with a calculation routine to determine the optimum economiser operating pressure based on a real-time measurement of the suction pressure. Knowledge of the suction pressure allows the appropriate saturated injection pressure to be determined, from which the optimum economiser pressure can be calculated for either capacity or COP optimisation. Modulation of the economiser pressure at the optimum value is used to continuously correct the economiser EEV signal, thereby optimising the economiser injection ratio.

7. ALGORITHM TEST RESULTS

The control algorithm was implemented for a multi-temperature indirect system for ATP set-point temperatures of -20,-20ºC, -10,-10ºC and 0,0ºC. Condenser water inlet temperatures were controlled at +18ºC, +22ºC and +25ºC. DX system performance, subject to a condenser water inlet temperature of +22ºC, was used as a baseline comparison. The chamber and the evaporator capacity are shown in Figs. 9 and 10. Differences between the chamber and evaporator capacities arise from the additional energy dissipation associated with the secondary pumps. IDX chamber capacity varied from 76% to 84.5% of DX baseline. For evaporator capacity this range was between 88% and 90%, relative to the DX baseline (Fig. 11). The IDX capacity differences relative the DX baseline was observed to decrease with increasing chamber setpoint temperature.

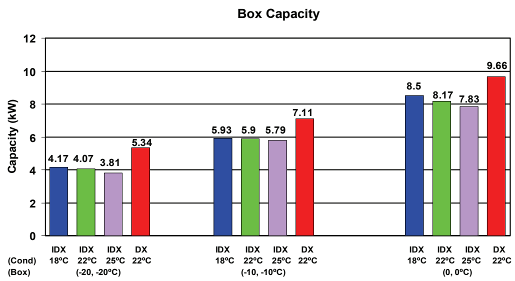

Figure 9. Chamber Capacity

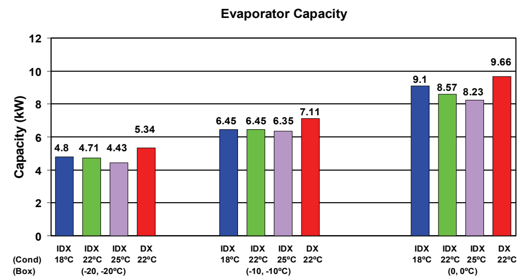

Figure 10. Evaporator Capacity

COP performance based on box capacity is shown in Fig. 11, and is less than COP performance based on evaporator capacity (Fig. 12). The additional heat exchanger and associated temperature glide required for the IDX system, coupled with the additional power requirements of the secondary pumps resulted in lower COPs for the indirect system. Fig. 12 shows that, based on evaporator capacity, COP performance to within 80-86% was achieved for the algorithm controlled economised IDX system when benchmarked against the DX baseline. The corresponding chamber COP was within 65-74% of DX COP.

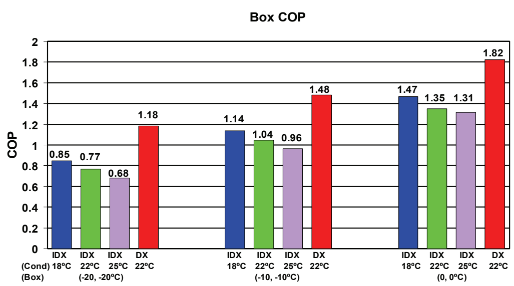

Figure 11. Overall (Chamber) COP

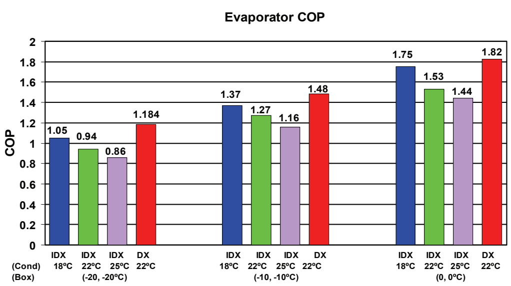

Figure 12. Evaporator COP

8. COMPARISON WITH INJECTION RATIO CONTROL STRATEGY

The pressure control algorithm was benchmarked against a system using manual control of injection mass flow rate as described in Smyth et al. (2010). Examining the chamber capacity in Figure 13, it was found that the pressure control controlled system using the capacity maximisation algorithm was within 4.7% of the conventional injection ratio controlled system at -20,-20ºC. At -10,-10ºC and 0,0ºC, the difference between pressure control algorithm and injection ratio control was reduced to 1.7% and 1.6%. If the evaporator capacity for these set points are considered, the differences between injection ratio controlled system and pressure controlled system are approximately 4.5% at -20,-20ºC, 1.98% at -10,-10ºC and 1.6% at 0,0ºC. Examining the COP values in Fig 15 revealed a slight increase in COP at the chamber. At -20,-20ºC, the COP was increased by 4% from 0.737 to 0.768. At -10,-10ºC, the corresponding increase in COP was 8% and at 0,0ºC was 7%. For -10,-10ºC, a corresponding increase in COP was also noted at the chamber. At -10,-10ºC, the COP was increased by 1.6% at the chamber. At 0,0ºC, the corresponding increase in COP at the chamber was 1.3%. At -20-20ºC, the COP was decreased at the chamber by 9.6%. For this test point, the COP optimisation algorithm maximises COP without a significant degree of capacity enhancement. When this is coupled with high pumping power and compressor lift coupled with the slightly augmented capacity was found to reduce the COP at the box. This was only found to occur at -20,-20ºC and was largely due to the poor thermophysical properties of the liquid ethylene glycol secondary coolant at low temperatures. Although COP can be optimised independently of capacity, the compressor design was based on low-side injection principle, which is more suitable for capacity enhancement (Beeton and Pham, 2003). Therefore as expected, a reduced degree of COP enhancement was observed relative to the DX baseline.

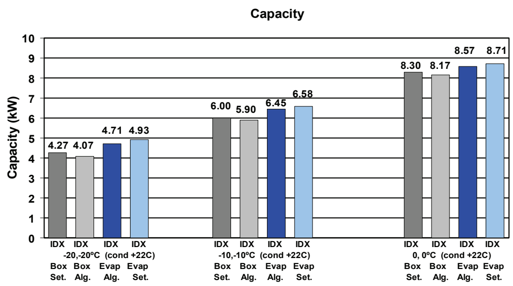

Figure 13. Chamber Capacity

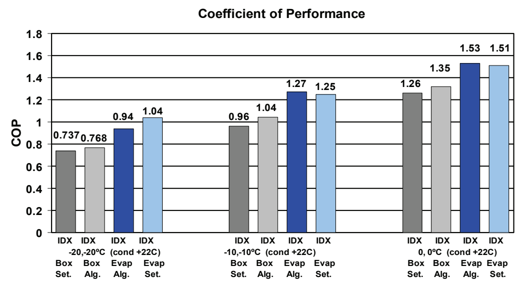

Figure 14. Coefficient of Performance

9. CONCLUSIONS

It was found that use of a pressure control algorithm is a suitable means of controlling the economiser cycle. The pressure control algorithm can be configured to control the system to optimise either capacity or COP of the indirect system. The capacity augmentation compared with contemporary injection ratio control. The algorithm provides satisfactory control of optimal injection pressure for maximisation of capacity and COP. For some test point conditions (-20-20ºC), better performance may be achieved by maximising capacity rather than COP, however this finding is influenced by the compressor and secondary coolant used in this study. Moreover, it would appear that pressure control approach is suitable for economiser installations, where practical issues constraining the installation of mass-flow instrumentation may apply.

ACKNOWLEDGEMENTS

This research was carried out under the auspices of the Innovation Partnership Programme (Project IP-2008-0539), which is administered by Enterprise Ireland. The financial support of Enterprise Ireland and ThermoKing Ltd is gratefully acknowledged.

REFERENCES

Aittomaki, A., Lahti, A., 1997. Potassium formate as a secondary refrigerant *Int. J Refrig,* 20(4), pp276-282.

ASHRAE, 2005. Fundamentals Handbook, *SI Edition*, ASHRAE Inc.

ATP, 2003. Agreement on International Carriage of Perishable Foodstuffs, *UNECE*, An.1, (3), pp12-34.

Beeton, W.L., Pham, H.M., 2003. Vapour-Injected Scroll Compressors, *J. ASHRAE,* April 03, pp22-27.

Clarke, R., Finn, D.P., 2008. The influence of secondary refrigerant air chiller U bends on fluid temperature profile and downstream heat transfer for laminar flow conditions, Int J. Heat and Mass Transfer, 51(3)4 pp724-735.

Haglund Stignor, C., Sunden, B., Fahlen, P., 2007. Liquid side heat transfer and pressure drop in finned tube cooling coils operated with secondary refrigerants, *Int J. Refrig,* 30, pp1278-1289.

Hinde, D., Zha, S., Lan, L., 2009. Carbon Dioxide in North American Supermarkets*, J. ASHRAE,* 51(2), p18-26.

Horton, W.T., Groll, E.A., 2003. Secondary Loop Refrigeration in Supermarket Applications – A case study, ICR0345 *Proc 21st IIF/IIR Int Cong. Refrig*, Washington DC, USA.

Kazachki, G. S., Hinde, D.K., 2006. Secondary Coolant Systems for Supermarkets, *J. ASHRAE*, 48(9), pp34-46.

Melinder, A., 1997. Thermophysical properties of liquid secondary refrigerants, *IIF/IIR* (Paris), 1997.

NI, 2009. LabVIEW Software, *National Instruments Corp.*

Rivet, P., 2003. Indirect versus direct systems in medium and large cooling plants, ICR0609, *Proc 21st IIF/IIR Int. Cong. Refrig*, Washington DC, USA.

Sawalha, S., Palm, B., 2003, Energy consumption evaluation of indirect systems with CO2 as a secondary refrigerant in supermarket refrigeration, ICR0434, *Proc IIF/IIR Int Cong Refrig,* Washington DC, USA.

Smyth, S., Brophy, B., Finn, D.P., Timoney, D.J., 2007. Performance Evaluation of Secondary Loop Systems for Multi-Temperature Transport Applications, ICR-D2-734, *Proc 22nd IIR Int Cong Refrig*, Beijing, China.

Smyth, S., Finn, D.P., Brophy, B., 2010, Influence of mass-flow injection ratio on an economised indirect multi-temperature transport refrigeration system, Proc 1st IIF/IIR Cold Chain Conference, Cambridge, UK.

Tassou, S.A., De-Lille, G., Ge, Y., 2009, Food transport refrigeration – Approaches to reduce energy consumption and environmental impacts of road transport, *Appl Th Eng,* 29, pp1467-1477.

Terrell, W.J., Mao, Y., Hrnjak, P.S., 1999. Evaluation of secondary fluids for use in low temperature supermarket applications, *ACRC CR-15,* University of Illinois, Urbana Champaign, IL, USA.