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<td><strong>Publication date</strong></td>
<td>2011-03</td>
</tr>
<tr>
<td><strong>Conference details</strong></td>
<td>1st IIR International Cold Chain Conference: Sustainability and the Cold Chain, Cambridge, UK, 29-31st March 2010</td>
</tr>
<tr>
<td><strong>Publisher</strong></td>
<td>The International Institute of Refrigeration</td>
</tr>
<tr>
<td><strong>Item record/more information</strong></td>
<td><a href="http://hdl.handle.net/10197/4744">http://hdl.handle.net/10197/4744</a></td>
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INFLUENCE OF MASS-FLOW INJECTION RATIO ON AN ECONOMISED INDIRECT MULTI-TEMPERATURE TRANSPORT REFRIGERATION SYSTEM

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ABSTRACT

Refrigerant leakage associated with multi-temperature direct expansion (DX) systems in transport refrigeration applications has lead to increased interest in alternative refrigeration concepts. One alternative design approach aimed at reducing refrigerant charge and simplifying system control, involves the use of an indirect (IDX) refrigeration circuit. Recent investigations, concerned with the deployment of indirect systems for supermarket applications have shown that penalties in cooling capacity and COP can exist under certain operating conditions. These performance deficiencies are attributed to a number of factors including: additional power requirements of secondary circulation pumps, temperature glide associated with the secondary working fluid, and increased compressor pressure lift due to the additional primary to secondary heat exchanger. One design strategy, aimed at offsetting the disadvantage of reduced refrigeration capacity is to incorporate an economiser circuit into the primary cycle of the IDX system. Economiser cycles can enhance the refrigeration effect of the primary refrigerant in the primary to secondary heat exchanger of the indirect system. The net effect can result in increased refrigeration capacity, if appropriate mass flowrates of refrigerant are expanded through the economiser circuit. However, to date, there is little evidence of the use of an economiser cycle in more complex refrigeration systems, such as multi temperature transport systems, which are subject to significant variation in boundary conditions and operate frequently under transient conditions. This paper examines for a multi-temperature indirect transport refrigeration system, the influence of injection ratio on system performance under ATP conditions as well as the sensitivity of system parameters to different injection ratios. It is found that system capacity and COP can be modulated by regulation of injection ratio. An optimum injection ratio exists which allows maximisation of system capacity and COP respectively. This optimum injection ratio appears to be dependent on set-point and ambient conditions.

1. INTRODUCTION

Direct expansion of primary refrigerants is exclusively used in multi-temperature transport refrigeration applications. Multi-temperature transport refrigeration systems are gaining increased attention due to potential reduction in required transit journeys, operating costs and environmental emissions (Tassou et al., 2009). Considerable environmental and control issues arise however, regarding their deployment. The wide range of operating temperatures, in addition to non-design conditions results complex control issues, which when combined with large refrigerant charges and consequent leakage risk, has given rise to increased environmental concerns. More recently, promising results have been noted with use of indirect refrigeration in stationary systems (Sawalha and Palm, 2003; Hinde et al., 2009). Single temperature optimisation (Kazachki and Hinde, 2006) has resulted in excellent performance for supermarket applications. It has previously been found that system optimisation measures are necessary to achieve comparable performance with baseline DX systems across the ATP test range (Smyth et al., 2007).

A key issue that arises in operation of an IDX system is the inherent penalty in system capacity (Terrell et al., 1999) and coefficient of performance (Kazachki and Hinde, 2006) compared with the direct expansion system. This is due to the presence of the additional heat exchanger and power requirements of the secondary pumps. Rivet (2003) notes that if no optimisation strategy is implemented for indirect systems in a particular application, only the direct impact will be reduced due to elimination of refrigerant leaks. This issue has been addressed in the literature for single temperature stationary applications, where optimisation at component level is discussed (Kazachki and Hinde, 2006). Other research aimed at component optimisation includes;
heat exchangers (Haglund-Stignor et al., 2007), secondary coolants (Melinder, 1997; Aittomaki and Lahti, 1997; Sawalha and Palm, 2003), and pumping power (Kazachki and Hinde, 2006). In multi-temperature transport applications, variable and non-design operating conditions typically prevail, thereby requiring both component and system optimisation. The presence of an additional temperature difference across the IDX system results in a lower evaporator pressure (relative to DX systems) for similar chamber set-points which adversely affects system capacity and COP. Considering the indirect system to comprise of distinct primary and secondary loops, most studies to date concern secondary loop optimisation. Little work has been carried out on primary cycle optimisation as part of an overall indirect system. The strong influence of primary cycle parameters (Kazachki and Hinde, 2006) on indirect system performance, would suggest considerable optimisation potential may exist from a primary cycle perspective. Where deployment of larger compressors and heat exchangers has been successful in stationary applications, these options are not always feasible in transport applications where packaging and weight constraints apply (Morley, 2003). Rivet (2003) and Kazachki and Hinde (2006) suggest that liquid subcooling may have considerable potential in primary side optimisation strategies, however little data exists to date on its contribution to indirect refrigeration systems.

Control of liquid subcooling is generally achieved through optimisation of condenser parameters. However for transport applications, additional challenges arise where packaging constraints and variable ambient conditions make control more challenging. Alternative means of achieving subcooling are generally based on mechanical subcooling cycles (Khan and Zubair 2000) or economiser cycles (Rivet, 2003). These novel designs have been proposed for stationary applications where increased capacity and COP has been observed. Khan and Zubair (2000) found that the mechanical subcooler cycle can be optimised for a particular set point condition. Winandy and Lebrun (2002) notes that increased capacity and COP are possible using an economiser cycle under certain circumstances. Neither concept has been investigated for multi-temperature indirect systems for transport applications.

This results in a number of key issues arising particularly with regard to the parasitic losses (Kazachki and Hinde, 2006) that arise from indirect system deployment. Is it possible, under variable operating and set-point conditions, that an economiser cycle can improve the capacity and performance of an indirect refrigeration system. However to what extent can system performance be improved under the range of ambient and setpoint temperatures typical of a transport environment, is still an open question. The wide range of operating and setpoint conditions would likely necessitate a control strategy of the subcooling system to optimise system performance for the expected boundary conditions and multi-compartment set-point temperatures. The use of economiser cycles (Figure 1) has been noted to give improved performance at low evaporator pressures (Winandy and Lebrun, 2002). This concept has to date primarily been deployed mainly in large-scale industrial screw compressors with multiple injection ports. The recent emergence of small-scale scroll compressors with vapour injection ports has resulted in considerable scope for installation other application such as indirect transport applications. The relatively recent introduction of this technology has resulted in little published literature, with the result that only single condition design procedures exist. Therefore considerable potential exists for optimisation of an economiser cycle in this application.

![Economiser Cycle](image.png)

Figure 1. Economiser Cycle
Control of economiser cycles is generally achieved by means of an auxiliary expansion valve to modulate refrigerant flow. Current design practice (Copeland, 2009), recommends the use of a fixed temperature difference approach with a thermostatic expansion valve. It has previously been found that this procedure results in non-optimum operation (Beeton and Pham, 2003), particularly at off-design conditions. The wide range of operating conditions of transport refrigeration systems would suggest considerable scope for optimisation of an economiser cycle for this application, which hitherto has not been investigated.

2. APPROACH

In this paper, experimental studies which examine the influence of injected mass flow ratio (INJR) on cooling capacity and COP of an economised indirect multi-temperature transport refrigeration system (Figure 2) are reported. The INJR is the ratio of the injected refrigerant mass flowrate (i) (Figure 1) to the main suction mass flowrate (m) (Winandy and Lebrun, 2002). Injection mass flow ratios from 0% to 50% were examined in this work. All tests were carried out with reference to ATP specification for a Class C refrigerated vehicle (ATP, 2003). A comprehensively instrumented multi-temperature indirect refrigeration test facility has been developed for performance evaluation of indirect refrigeration systems with direct expansion systems. An economiser circuit was incorporated into the primary loop of this test facility as per Figures 1 and 2. It consisted of a compressor with a vapor injection port and a liquid line heat exchanger. Refrigerant mass flowrate regulation was achieved by a computer controlled electronic expansion valve (EEV).

![Diagram](image)

Figure 2. Indirect Multi-Temperature Systems for Transport Refrigeration

3. EXPERIMENTAL

The vapour injected economiser cycle, as installed into the primary cycle of the prototype indirect refrigeration system, is shown in Figure 3. A hermetic scroll compressor with a displacement volume of 17.1 m³/hr at 50Hz was installed onto the apparatus. This compressor was equipped with an auxiliary suction port for connection to the economiser cycle. The apparatus was completed in accordance with the design procedure of Beeton and Pham (2003). This necessitated installation of a subcooling heat exchanger, auxiliary expansion valve and associated pipework. A stepper motor modulating valve was incorporated in place of the thermostatic expansion valve. HFC-R404A is deployed in the primary circuit, with 50% V/V aqueous ethylene glycol as the secondary coolant.
The electronic stepper-motor expansion valve enabled continuous modulation of the injected refrigerant flow from zero to maximum. Condenser conditions were maintained at the desired level by means of a proportional-integral-derivative (PID) controlled heat unit. Compressor speed was controlled at 50Hz for all tests and fans and pumps were run at rated speed. Instrument calibration is carried out in accordance with NIST (2009) and ASHRAE (2005). Instrument data is recorded by means of a 16 bit data acquisition system based on a LabVIEW platform.

4. TEST MATRIX

Side by side testing was investigated at ATP conditions of -20°C, -10°C and 0°C. The performance parameters of the system are studied in injection ratio increments of 5% from zero INJR to the maximum possible. Both chambers were supplied with secondary coolant in parallel. Temperature control within the test chambers was implemented by means of electrical resistance heaters incorporating feedback control to maintain setpoint temperature. Box and condensing conditions are maintained to within +/-0.5°C for the duration of the tests.

5. RESULTS

All investigations were carried out at a constant condensing condition with a water inlet temperature of 22°C and flowrate of 15 L/min. At a set-point temperature of -20°C for both compartments, refrigeration capacity at the evaporator was observed to increase with increasing INJR ratios up to approx 35% (Figure 4), where a maximum refrigeration capacity was observed, before decreasing for higher INJR ratios. For INJR ratios between 30 and 40%, evaporator cooling capacity was observed to increase by approximately 21% to 4.8kW, with reference to a non-economised capacity of 3.96kW. Figure 6 shows that for the optimum INJR of 33.5%, compressor electric power consumption increased from 4.26kW to 5.01kW (17.6%), and this resulted in a slight increase in COP from 0.93 to 0.96 (3%) (Figure 7). As INJR is increased beyond 40%, the cooling capacity is observed to gradually decrease (Figures 4 and 5), whereas compressor power continues to increase (Figure 6). This resulted in an increase in condenser heat rejection (Figure 8), however the cooling COP decreased (Figure 7). Tests were also carried out at other ATP set-point conditions (-10°C and 0°C). For the -10-10°C condition, an increase in evaporator capacity was noted up to 25% INJR, where a maximum capacity of 6.39kW was noted. Thereafter, a decrease in evaporator capacity is seen as the INJR is increased
to the maximum. For a box temperature of 0.0°C, capacity increase is observed with increasing injection ratio up to 17%, and thereafter a decrease in capacity is noted. By increasing the injection ratio further, it is not possible to increase the evaporator capacity, as the refrigerant saturation temperature in the economiser is increased. Therefore the log mean temperature difference is degraded, resulting in reduction of refrigerating capacity.

Returning to Figure 5, the variation of refrigeration capacity with economiser pressure is depicted. For each ATP condition, a unique economiser pressure exists. The minimum pressure at each operational condition (Figure 5) corresponds to the Saturated Injection Pressure (SIP) for the compressor, based on an INJR of zero. A unique SIP exists for each operating condition. By increasing the injection ratio, an increase in economiser pressure is noted. An optimum economiser pressure exists whereby capacity is optimised. This optimum pressure depends on the particular ATP test point in question.

The increased evaporator capacity coupled with the increase in compressor power consumption will also result in an increase in condenser capacity. This is due to the increased refrigerant mass-flowrate resulting from the increased injection ratio. It can be seen from Figure 8 that with increasing injection ratio, an increase in condenser capacity is evident. Although not the case with the evaporator capacity, it is evident here that the condenser capacity is maximised when the injection ratio is maximised. It should be noted however, that the condenser capacity will also be limited by the maximum capacity of the condenser heat exchanger.

The condensing COP is depicted in Figure 9. It can be seen that despite an increasing injection ratio, a relatively constant condensing COP is evident. A slight increase in condensing COP can be seen between 0-10% INJR and at the optimum injection ratio for optimum capacity 25-30%. Therefore since two optimum condenser COP conditions exist, the choice of condenser COP will depend on power availability at the compressor.
The condensing pressure is depicted in Figure 10. By increasing the condenser capacity using injection ratio, the total mass flowrate is increased with consequent increase in condenser duty. Since a constant water inlet temperature is maintained, the condensing pressure will increase with injection ratio due to the increased condenser duty. The condensing pressure follows approximately the same trend as the condenser capacity, provided the water inlet temperature and flowrate is maintained at a constant level and the limit of the condenser heat exchanger capacity is not exceeded.

The relationship between economiser pressure and injection massflow is depicted in Figure 11. By increasing the mass-flowrate of injected refrigerant into the economiser, the economiser pressure is increased along with the saturation temperature. This eventually limits the subcooling effect of the economiser cycle at high injection ratios. It can be seen that where the injection massflow is zero, the economiser pressure is not zero but approaches the saturated injection pressure (SIP). This corresponds to the saturated injection temperature and is dependent on the compressor design and condenser and evaporator conditions.

The evaporator COP is illustrated in Figure 12. It can be seen that by continually varying the injection ratio up to its maximum, that an initial increase in COP is achieved, followed by a decrease, and then an increase again. For high setpoint temperatures (0,0°C), the COP is maximised at low injection ratios. For low setpoint temperatures (-20-20°C), the COP is maximised at high injection ratios. Therefore a choice exists, depending on the required capacity and available power requirements whether the COP is maximised at low capacity and low compressor power or at high capacity and high compressor power.
Figure 12. Evaporator COP

Figure 13 Compressor Discharge Temperature

Figure 13 depicts the compressor discharge temperature with increasing injection ratio. It can be seen that by increasing the injection ratio up to 5%, an initial increase in discharge temperature is observed. The additional massflow of refrigerant results in an increase in compressor power (Figure 6). The initial increase in compressor discharge temperature is due to the increased superheating of suction gas in the hermetic casing due to the increase motor power. This increase in compressor discharge temperature is observed to drop continuously when the injection ratio is increased beyond 5%. The low temperature of the injected stream absorbs heat from the compression process, and reduces the discharge temperature accordingly. As the injection ratio is varied, the suction massflowrate remains constant (Wang 2009a, Wang 2009b). Therefore by increasing injection ratio, the total mass flowrate will increase. As expected, the compressor power consumption increases continuously with injection ratio (Figure 6).

6. CONCLUSIONS

Optimal injection ratios exist for both capacity and COP and these appear to be dependent on setpoint conditions. Moreover, it would appear that optimal injection ratios are different than in cooling mode compared to heating mode. In order to maximise the effectiveness of an economiser cycle for this application, careful system integration and optimisation are likely to be required in any practicable system.

7. ACKNOWLEDGEMENTS

The financial assistance of Enterprise Ireland and ThermoKing Ltd. is gratefully acknowledged.
8. REFERENCES

16. 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

1st IIR International Cold Chain Conference, Cambridge, UK, 2010