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Experimental Feasibility Investigation of the Use of Indirect Refrigeration Systems (Secondary Loops) in Climate Controlled Cargo Transport Systems

by

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This paper examines the performance of a prototype indirect (secondary loop) refrigeration system developed as an alternative design to existing direct expansion multi-temperature systems in transport refrigeration applications. The work is framed in the context of an increased demand for multi-temperature transport refrigeration systems, coupled with the environmental and control issues that arise as a direct consequence of this demand. Multi-temperature transport refrigeration systems almost exclusively utilise direct expansion (DX) refrigeration for climate control of the multi-compartments. Although incremental design changes have been evident in the past decade, considerable environmental and control issues continue to exist with these systems. IDX systems may offer a solution to the shortcomings of DX systems in transport refrigeration applications, however a number of key issues remain unaddressed. Foremost amongst these include: the requirement of a secondary fluid with acceptable thermophysical properties across multiple ATP conditions, good material compatibility with minimal environmental impact; effective fluid-system integration, and packaging issues such as minimisation of fluid volume, fluid weight and choice of pumps and heat exchangers. For single temperature stationary systems, the design can be optimised based on a uni-temperautre design criterion, however for transport refrigeration systems, a number of challenging issues arise due to the non-uniform, non design operating conditions frequently encountered. In order to examine these issues, a multi-temperature experimental test facility was designed and built with the capability to test both systems at standard ATP test conditions . Evaluation was based on side-by-side testing of the direct and indirect systems for a range of cold room set point temperature set point combinations between -20˚C and 0˚C.

1. INTRODUCTION

The increased demand for refrigerated transport systems over the past decade can be viewed in the context of an increase in worldwide consumption of perishable foodstuffs (Tassou et al. 2009). In addition, heightened consumer expectations concerning produce quality and availability, as well as commercial demands for faster and more reliable time frames for product transportation, have accelerated this trend. In the transport refrigeration sector, these developments, when considered with current energy usage and possible over reliance on direct expansion (DX) refrigeration systems, have prompted concern regarding the future sustainability of existing technologies (Chopko and Stumpf 2003, Spence et al. 2004). This is particularly true for multi-temperature transport refrigeration systems, which constitute an important sector within the global cold chain. Multi-temperature transport refrigeration systems are increasingly being used in urban distribution environments, where they offer more flexible scheduling thereby giving potential reductions in associated transit journeys, operating costs and environmental emissions (Tassou et al. 2009). These systems enable simultaneous transport of chilled, fresh and frozen produce and generally deploy two or three direct expansion evaporators for climate control of individual trailer compartments, all of which operate in conjunction with a single compressor-condenser unit (Figure 1). Multi-temperature systems must be capable of providing independent temperature control between -30˚C and +25˚C in any of the individual compartments, regardless of ambient temperature.

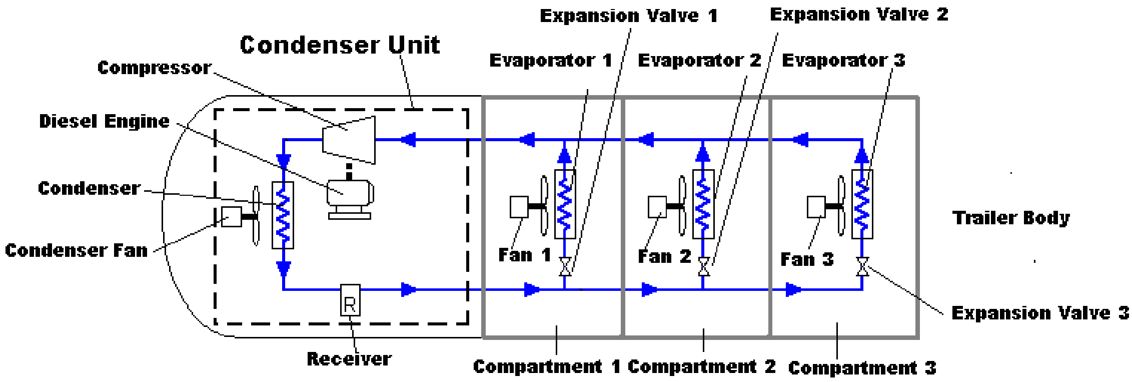

Figure 1. Direct Expansion (DX) Multi-Temperature Transport Refrigeration Unit

Direct expansion of refrigerant within the conditioned compartments necessitates considerable refrigerant distribution line lengths with consequent requirement for large refrigerant charges, which can give rise to increased refrigerant leakage risks, additional pressure and system losses and inherent control challenges. Pressure control issues associated with multi-temperature DX refrigeration also militate against these systems resulting in considerable potential for alternative design concepts (Horton and Groll 2003). From the context of environmental protection, stricter legislation concerning refrigerant charges is likely to militate against the continued use of DX technology in the transport refrigeration sector. As a result, increased interest in alternative temperature control concepts in cargo transportation systems has been evident in recent years. Despite the existence of clear challenges and the strong need to reduce environmental impact and improve temperature control and flexibility, previously reported research in this area is not widely reported in the literature. Notable examples include the hybrid diesel-electric system of Chopko and Stumpf (2003) and the novel air cycle system of Spence et al. (2004).

In recent years ,indirect (IDX) refrigeration systems have been been suggested as a possible alternative approach to DX technology (Kazachki and Hinde 2006). This concept has been utilised in recent years in supermarket systems (Horton and Groll 2003), ice rinks and industrial applications (Rivet 2003). Noted advantages include reduced refrigerant charge (Horton and Groll 2003), better control (Kazachki and Hinde 2006), simpler defrost and improved reliability (Hinde et al. 2009). Kazachki and Hinde (2006) argue that recent advances in indirect system technology, have resulted in IDX technology becoming the most reliable solution to address increased regulatory legislation, particularly in the area of primary refrigerant leakage. IDX systems operate on the principle whereby a single phase secondary fluid is used in conjunction with multiple remote chillers, whereby heat can be exchanged with a single direct expansion primary refrigeration circuit using a secondary to primary heat exchanger (Figure 2). The requirement of an additional heat exchanger and liquid secondary pumps can give rise to a consequent reduction in capacity and COP, necessitating optimisation to minimise these potential losses and achieve comparable performance to DX systems (Wood et al. 1996).

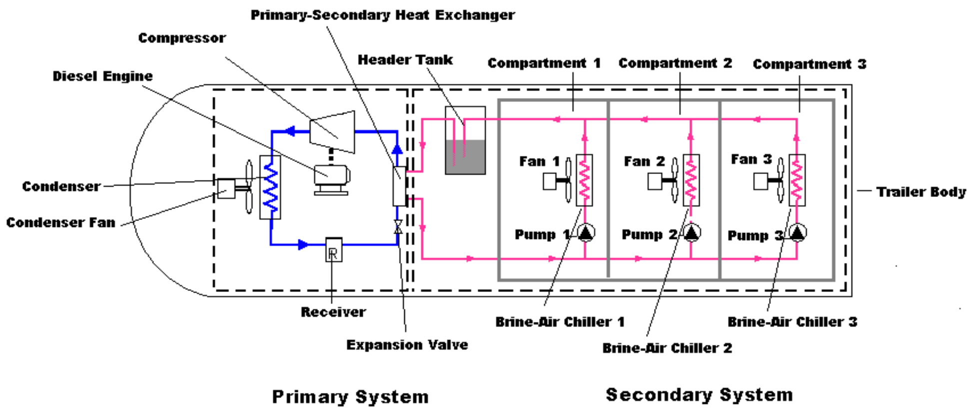

Figure 2. Indirect Multi-Temperature Transport Refrigeration System.

A key issue regarding IDX systems concerns the choice of secondary circuit components and parameters such as working fluid, pumps, valves and heat exchangers. Thermophysical properties and data tables (Melinder 1997) describing temperature dependent profiles of specific heat capacity, dynamic viscosity, density and thermal conductivity are commonly available. In industrial, stationary and supermarket systems, a secondary fluid can be selected on the basis of most favourable thermophysical properties at a single design condition. In transport refrigeration applications, non-design, non-uniform operating conditions are frequently encountered. This results in a number of coinciding issues concerning selection of a suitable secondary coolant for indirect multi-temperatre transport applications which must be considered. These issues concern the variability of fluid properties at variable ATP (ATP 2003) conditions, material compatibility issues, environmental impact, effective fluid system integration and packaging constraints such as minmisation of volume, fluid weight and choice of pumps and heat exchangers. To date, no published literature exists concerning the suitability of any particular secondary coolant under ATP conditions. Terrell et al. (1999) found that the individual thermophysical properties do not adequately describe the suitability of a secondary coolant for a particular application.

Based on experience of IDX systems in supermarket applications, it would appear that there is potential for deployment of the IDX concept in multi-temperature transport applications. Design of efficient IDX systems for transport refrigeration applications raises a number of challenging issues including the requirement of comparable capacity and COP, effective system integration and control as well as system packaging issues such as minimisation of volume, weight, noise and vibration. In multi-temperature transport applications, variable and non-design operating conditions frequently prevail, thereby requiring both component and system optimisation, at different design set-points and a wide variety of operational conditions. To date, no published literature exists concerning the feasibility of an IDX approach in multi-temperature transport refrigeration systems. In the absence of previous research on this issue, the development of an experimental test facility, capable of evaluation of an indirect refrigeration concept in transport refrigeration system was required, which is the subject matter of the current paper.

2. TEST PROTOCOLS

Performance data for DX transport refrigeration units can be specified according to either ARI or ATP test protocols. The ARI standard rating condition of 35°F (+1.7°C) box temperature and 100°F (37.7°C) ambient temperature was used by Chopko and Stumpf (2003) for evaluation of all-electric transport refrigeration units. The alternative ATP rating condition specifies two box set-point temperatures of 0°C and -20°C in conjunction with an air-on condenser temperature of +30°C (ATP 2003). Under ATP Category C (ATP 2003), a standard multi-temperature refrigerated trailer can contain either two or three compartments each with independent control of individual box set-points between +10°C and -20°C. A two zone configuration was utilised in this research, with the capability of operation in either DX or IDX mode.

3. EXPERIMENTAL TEST RIG

Figure 3 shows a schematic representation of the multi-temperature installation used to emulate the ATP test conditions outlined in Table 2. To maintain independent control of each compartment, the DX system has two independent parallel evaporators, whereas the IDX system uses individual remote chillers (air fan coil units). Each remote chiller was required to be supplied with secondary coolant from a single evaporator (intermediate heat exchanger) by means of an independent pump. The parallel circuit mode was used to facilitate independent temperature control within each box, whereas the use of the series flow circuit could be optionally used when different temperature set-points were required in each compartment. The use of separate circulation pumps for each circuit also facilitated independent defrost as required for each compartment. Figures 3(b)/(c) show the system. The DX evaporators were rated for a semi-trailer unit of volume 72.6 m^2, with a specified cooling rating per evaporator of 3 kW at -20°C and 5 kW at 0°C box set points. Air volume flow rate, based on three evaporators for dry coil conditions at 0 Pa static pressure difference, is 5000 $m^3.hr^{-1}$. For partially frosted coils, with a back pressure of 450 Pa, the airflow is reduced to 3600 $m^3.hr^{-1}$.

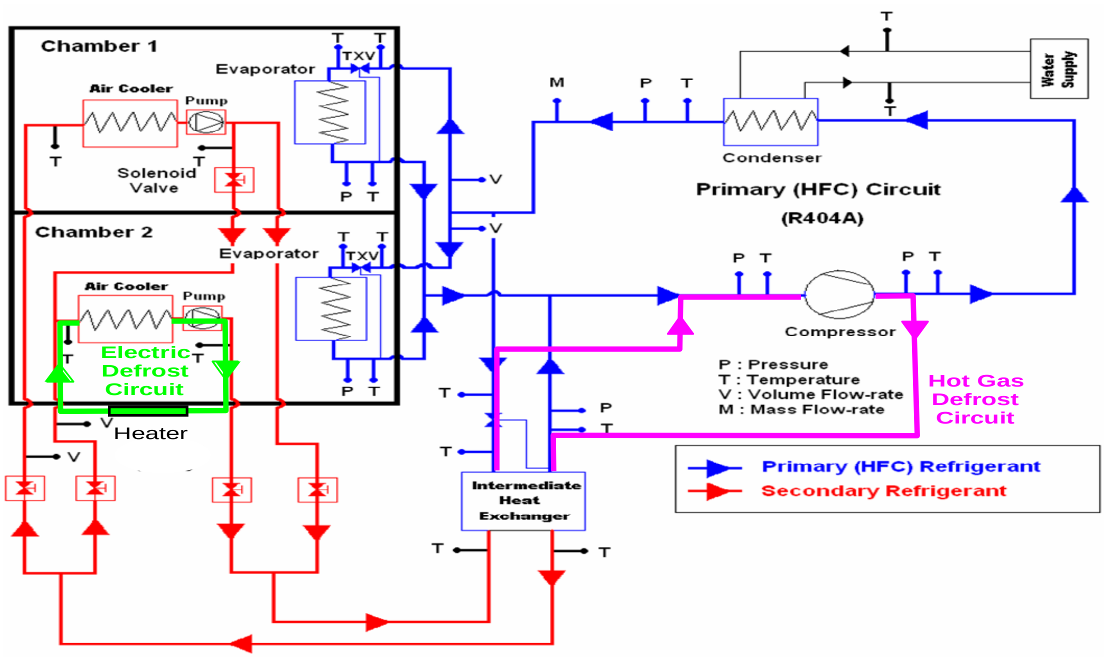

Figure 3(a). Schematic diagram of DX and IDX systems.

Figure 3 (b)/(c). Transport Refrigeration Test Rig: (a) Primary system with twin insulated compartments to rear. (b) Inside insulated compartments with associated DX evaporators (middle) and IDX remote coolers (top/bottom).

4. IDX SYSTEM DESIGN CONDITION

The single point design approach described by Wood et al. (1996) and later by Kazachki and Hinde (2006) was used in this study. This is based on utilisation of a single temperature difference for specification of evaporator saturation temperatures, heat exchanger duty, secondary fluid flow rates, pump duty and primary and secondary line sizes. For the IDX system, the ATP set-point temperature (-20,-20°C) was utilised along with an ambient condensing condition of +30°C, thereby ensuring that the IDX system capacity would match the 6 kW capacity of the DX system. A flow rate of 29 L.min^{-1} with a 5 K secondary fluid glide was chosen, as this enabled the use of standard copper piping for the secondary circuit. A 60 plate brazed-plate heat exchanger with a design LMTD of 5 K was used as the primary to secondary heat exchanger, so as to deliver the cooling load of 6 kW at an evaporating temperature of -30°C using R404A as the primary refrigerant. Each of the compartment air chillers had a capacity of 3 kW at an ATP condition of -20°C with 5 K temperature difference on the liquid side and air-side approach. For the initial prototype air chillers, each air cooler consisted cross flow heat exchanger consisting of 96 tube passes in 6 columns of 16 rows of tubes. The coolant flow circuit configuration consists of 3 circuits of 32 passes each resulting in an overall working volume of 8.4 litres and a pressure drop of 67 kPa at the design flow rate of 0.24 L.s^{-1}. The 600 x 560 mm flow duct contains a wavy fin pack with a depth of 182 mm. The fin pack consists of 0.15 mm thick fins spaced at 3.5 mm intervals giving an overall surface area of 23.8 m^2. This design necessitates an air flow rate of 1620 m^3.hr^{-1} resulting in an air side pressure drop of 1.2 Pa. The average air velocity required to satisfy the design condition was 1.34 m.s^{-1}. The induced air flow was provided by means of a 350 mm duct mounted axial fan. This fan was selected from performance rating curves for a flow rate of 1620 m^3.hr^{-1} at an air side static pressure drop of 1.2 Pa. A centrifugal type wet pump fitted with an integral variable speed drive (VSD) which permitted modulation of volume flow rate by variable speed control was deployed. The pump motor was designed for a maximum speed of 80 Hz and a maximum power consumption of 375 W. A warm liquid defrosting system was implemented on the IDX system consisting of an auxiliary heated coil loop. A schematic of this warm liquid loop is shown in Figure 2. The defrost loop consisted of an inline heater assembly containing a 3 kW resistive heating element. The electrical resistance heater was controlled by means of a variable power supply to accurately modulate heat flux and defrost energy. This arrangement was implemented on both boxes, thereby allowing independent defrost of the individual boxes.

5. ENERGY BALANCES

The ATP directive outlines a standardised procedure to measure box loss, capacity and COP of a refrigerated transport system (ATP 2003). Experimental analysis was conducted within a controlled environmental compartment where the internal temperature was maintained by a metered quantity of electrical heat. Calibrated calorimeters in accordance with ATP Annex C were used for this research with independent control of individual box temperature between to -20°C and +10°C. In this work, twin compartments were used, which were constructed from aluminium-cladded rigid polyurethane foam (PU) composite. Box 1 has dimensions 3.64 m x 3.1 m x 2.42 m and box 2 with dimensions 2 m x 3.1 m x 2.42 m (L x W x H). The same compartments were used for both DX and IDX analysis. Temperature measurement within the boxes was by means of six T thermocouples installed at various locations within the boxes. The temperature readings combined with the experimentally determined box wall UA values allow determination of the heat transmission loss (or box loss) through the box walls during testing. Each box was fitted with two PID controlled electrical resistance heaters to maintain a steady box temperature to ±0.5 K. Separate analysis of the energy balances on the IDX and direct expansion systems were examined on the DX and IDX systems. Each system was examined using a case study examining the independent energy balances and uncertainty associated with the absolute values. The uncertainty analysis was carried out using a Taylor expansion of the component uncertainties associated with each measurement (Table 1). The case study for both systems is the low temperature ATP operating condition (-20°C) with a condensing condition of +22°C (water inlet temperature).

Component	Energy Balance	Equation	Nominal Value	Uncertainty
Compressor	Electrical Side	$P_{comp} = V_{comp} . I_{comp}$	-	±0.11 kW
Condenser	Water Side	$\dot{Q}_{cond\_water} = \dot{m}_{cond\_water} \, Cp_{cond\_water} \, \Delta T_{cond\_water}$	8.97 kW	±0.342 kW
	Refrigerant Side	$\dot{Q}_{cond\_refrig} = \dot{m}_{refrig} \, \Delta h_{cond}$	8.85 kW	±0.116 kW
Intermediate Heat Exchanger	Refrigerant Side	$\dot{Q}_{evap\_IDX\_refrig} = \dot{m}_{refrig\_evap} \, \Delta h_{evap\_IDX}$	4.39 kW	±0.071 kW
	Coolant Side	$\dot{Q}_{evap\_IDX\_sec} = \dot{m}_{sec\_coolant} \, Cp_{coolant} \Delta T_{evap}$	4.23 kW	±0.185 kW
DX Evaporator	Refrigerant Side	$\dot{Q}_{DX\_evap} = \dot{m}_{refrig\_DX\_evap} \, \Delta h_{DX\_evap}$	3.27 kW	±0.132 kW
	Box Side	$\dot{Q}_{box} = \dot{Q}_{elec\_heater} + \dot{Q}_{box\_loss}$	-	±0.06 kW
IDX Air Cooler (Air Chiller)	Coolant Side	$\dot{Q}_{remote\_cooler} = \dot{m}_{sec\_coolant} \, Cp_{coolant} \Delta T_{remote\_cooler}$	1.59 kW	±0.095 kW
	Box Side	$\dot{Q}_{box} = \dot{Q}_{elec\_heater} + \dot{Q}_{box\_loss}$	-	±0.06 kW

Table 1. Energy Balance Equations.

6. TEST MATRIX

The wide range of operational temperatures encountered in transport applications necessitated use of a reduced test matrix which evaluated key ATP test conditions. The test matrix is detailed in Table 2. Side by side testing of overall system performance, where different secondary coolants were utilised was investigated for chamber set-point temperatures of -20°C and 0°C. An intermediate set point temperature of -10°C was also investigated in this instance. The prototype test rig was fitted with a configurable flow control system to enable series and parallel flow of coolant as required. The test matrix contains 4 groups of tests, representing the main flow configurations of the secondary circuit. Secondary coolant was configured for parallel flow (Groups1 & 2), single zone flow (Group3) or series flow (Group 4). Side by side testing was carried with zones at a single temperature (Group1 & 3) or at different temperatures (Group 2 & 4). Four secondary coolants were examined alongside the baseline DX system. The four secondary coolants examined were Ethylene Glycol (50% V/V), Hydrofluoroether 7200 (HFE), Potassium Formate/Sodium Propionate (40% V/V) and Potassium Formate (50% V/V). All experimental testing was carried out with condenser water at an inlet temperature of +22°C and a flow rate of 15 L/min. Compressor speed was maintained at 50Hz and fans and pumps were run at rated speeds.

	Test No.	Zone 1 (°C)	Zone 2 (°C)
Group 1	1	-20	-20
Single Temperature	2	-10	-10
Parallel Flow	3	0	0
Group 2	4	-20	0
Multi-Temperature	5	-20	-10
Parallel Flow	6	-10	0
Group 3	7	-20	N/A
Single Temperature	8	0	N/A
Group 4	9	-20	-10
Series Flow	10	-20	0
	11	-20	10

Table 2. Test Matrix.

7. RESULTS

The results obtained show that the choice of secondary working fluid can have a significant influence on both system capacity and coefficient of performance (COP). A distinction is made at this stage between the evaporator capacity and the refrigeration capacity. Since the indirect system contains an additional temperature stage across the intermediate heat exchanger, two capacities can be defined. The evaporator capacity represents the capacity at the intermediate heat exchanger, whereas the refrigeration capacity represents the capacity at the compartment. For the DX system these two concepts are equivalent. The evaporator capacity facilitates performance comparison of the primary cycles, whereas the refrigerating capacity represents the useful available cooling capacity.
Total refrigeration cooling capacity is shown in Figure 4, whereas Figure 5 depicts evaporator capacity. The difference between refrigerating capacity and evaporator capacity in the IDX system is due to heat gains arising from internal energy dissipation arising from pump work input. This results in degradation of the log mean temperature across the heat exchanger, with consequent loss of cooling capacity. This can be significant in the case of high-viscosity secondary coolants such as ethylene glycol (Figures 4 & 5), where these heat gains are considerable. (Inspection of Figure 13 indicates that the total pumping power for ethylene glycol is approx 680W at -20-20°C which amounts to 18% of total refrigeration capacity (3.82kW)). Moderate capacity is evident for HFE at low temperatures, however a more pronounced decrease is evident at higher temperatures. The low viscosity of HFE permits relatively high flow rates even at low temperatures, however its low specific heat capacity results in poor performance at high temperatures relative to the other aqueous secondary working fluids. To overcome these shortcomings, HFE is often operated in the turbulent regime to enhance heat transfer, this was investigated in this work, however only with considerable difficulty. Even with a pressurised system, the low viscosity of HFE resulted in cavitation of the liquid pumps at higher speed. Nevertheless, it is concluded that HFE operated acceptably for low temperature operation, however for higher temperature conditions, turbulent flow was found to be necessary in order to overcome the limitations of its low specific heat capacity.

Ethylene glycol provided good cooling capacity at higher temperatures. High specific viscosity can inhibit performance at low temperatures resulting in low flow rates, high temperature glide and excessive pumping power. The additional pumping power required to overcome the higher viscosity at lower temperatures is ultimately dissipated in the flow, which results in degradation of refrigeration capacity. It can be concluded that ethylene glycol provides good capacity at higher temperatures, however, performance is compromised at low temperatures. Pure potassium formate was found to provide good performance across the operating range, and was broadly similar to potassium formate/sodium propionate solution. Slightly better performance was noted for the potassium formate/sodium propionate solution due to the moderately better thermophysical properties of this secondary coolant. This difference is largely due to the potassium formate concentration (40% Vs 50%) and different corrosion inhibitor mixture deployed. It can be seen that capacity is reduced for series flow relative to parallel flow. This is due to the increased temperature difference across the secondary circuit due to the series flow regime. This increased the evaporator mean temperature difference, resulting in considerable reduction on evaporator pressure (Figure 11).

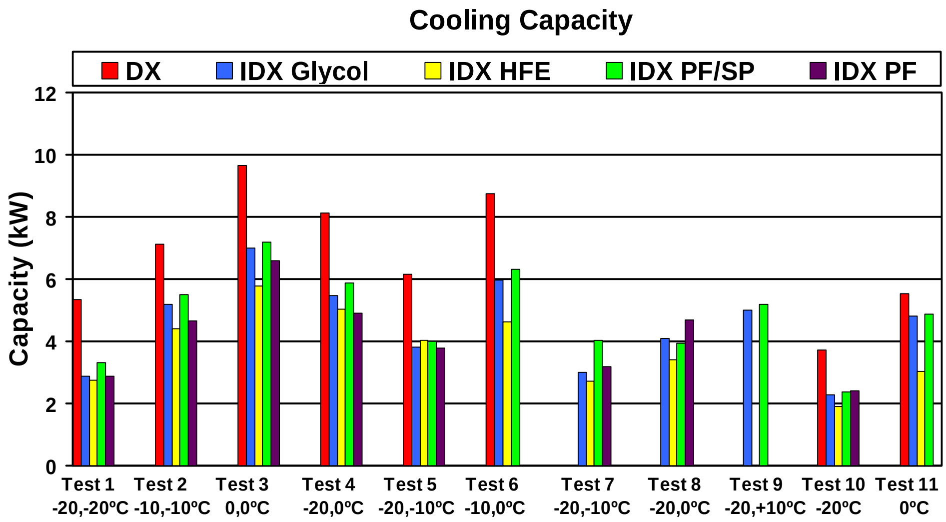

Figure 4. Cooling Capacity.

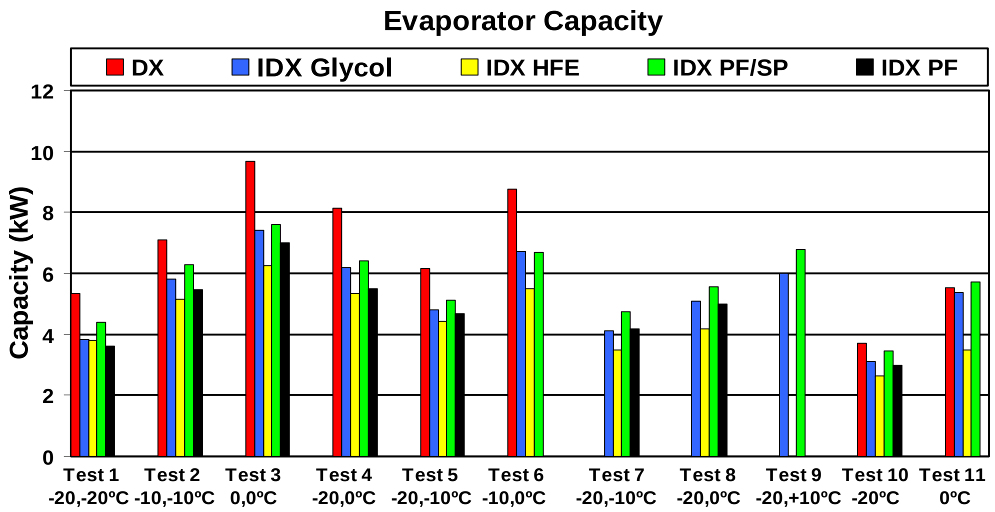

Figure 5. Evaporator Capacity.

The electrical power consumption associated with the primary cycle compressor is depicted in Figure 6. It can be seen for all IDX tests that the compressor power is higher than the baseline DX tests. This is due to the lower refrigerant specific volume at the compressor suction due to close proximity of evaporator suction to compressor suction. The presence of suction–liquid heat exchangers on the DX evaporators results in increased refrigerant specific volume at the compressor suction which reduces compressor power consumption.

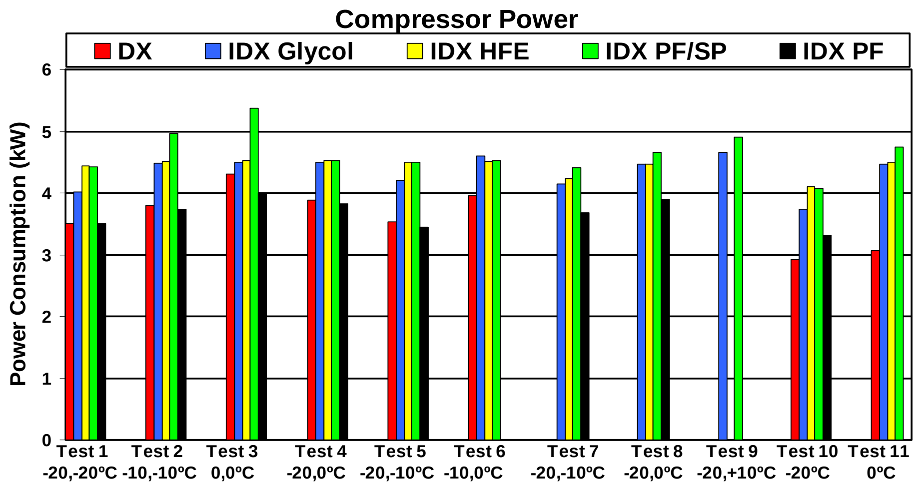

Figure 6. Compressor Power.

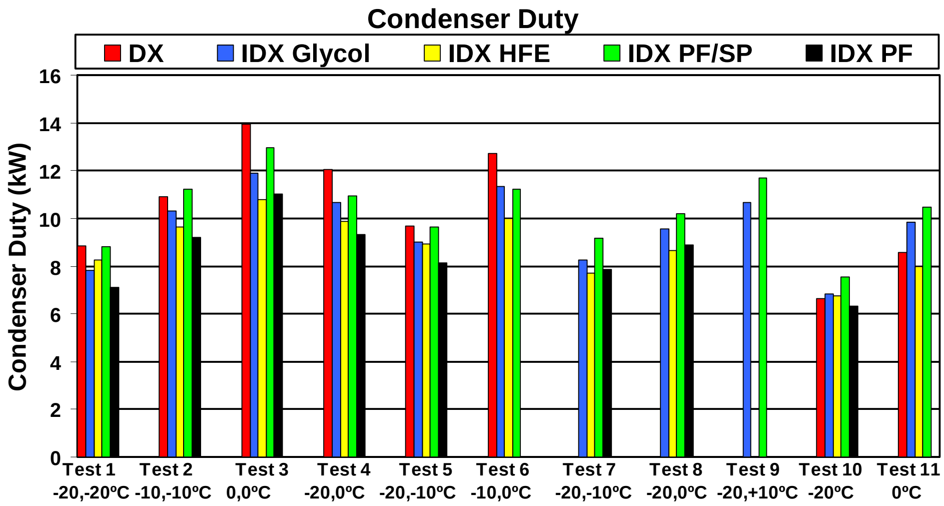

Figure 7. Condenser Duty.

Compressor power consumption was also found to be influenced by the thermophysical properties of the secondary coolant (see Figure 6). For secondary coolants with high specific heat capacity, higher compressor power was evident due to the increased mass flow rate of the primary refrigerant through the evaporator. Conversely for low specific heat capacity fluids, compressor power consumption was lower. The additional power supplied by the liquid pumps resulted in an increased condenser load (see Figure 7). It could be suggested in this case, that the internal energy supplied to the secondary coolant by the pumps is beneficial to the condensing capacity (Figure 7). However it must also be noted that as a result of the increase in pumping power, a reduction in condenser COP also occurs (Figure 10). The evaporator COP and overall system COP are shown in Figures 6 and 7 and also depend on the choice of secondary coolant. The evaporator COP is calculated with reference to the capacity at the primary cycle, whereas the overall system COP is relative to the available refrigeration capacity. The COP is calculated with reference to the compressor power, pump power and fan power. Capacity is reduced for all set point temperatures due to the additional power requirements of the secondary pumps and the additional temperature difference across the heat exchanger.

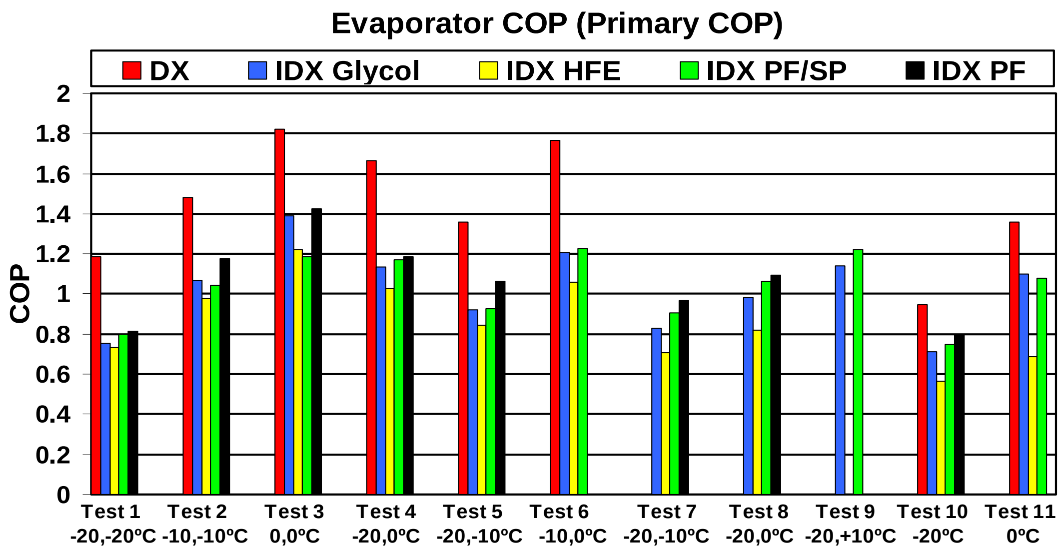

Figure 8. Evaporator COP.

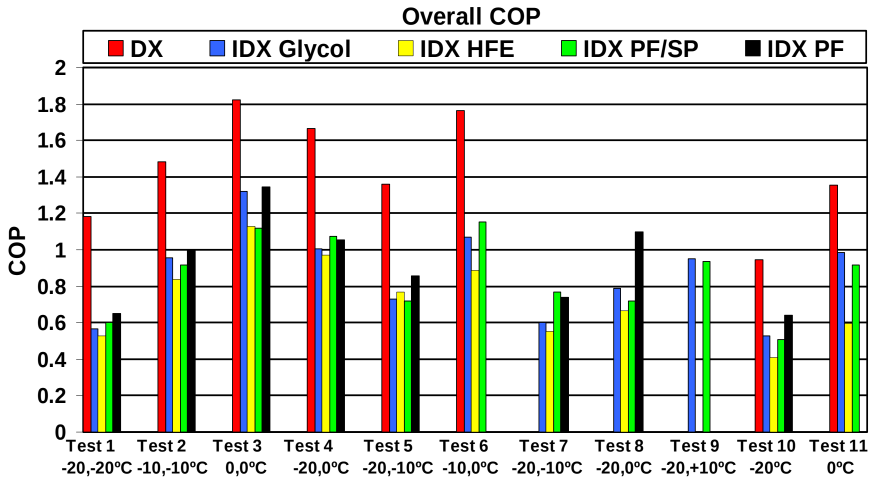

Figure 9. Overall COP.

From Figures 8, 9 & 10, it can be seen that the choice of secondary coolant can influence the system COP. Relatively good performance is evident for pure potassium formate, however its moderately high COP is due in part to a lower condensing temperature. The low viscosity and pumping power of HFE contributes to good COP at low temperatures, with limitations of specific heat capacity considerably reducing COP at higher temperatures. Good performance is also noted for ethylene glycol, especially at higher temperatures such as 0,0°C, where its relatively low viscosity and high specific heat capacity contributes to good heat transfer performance at these operating conditions.

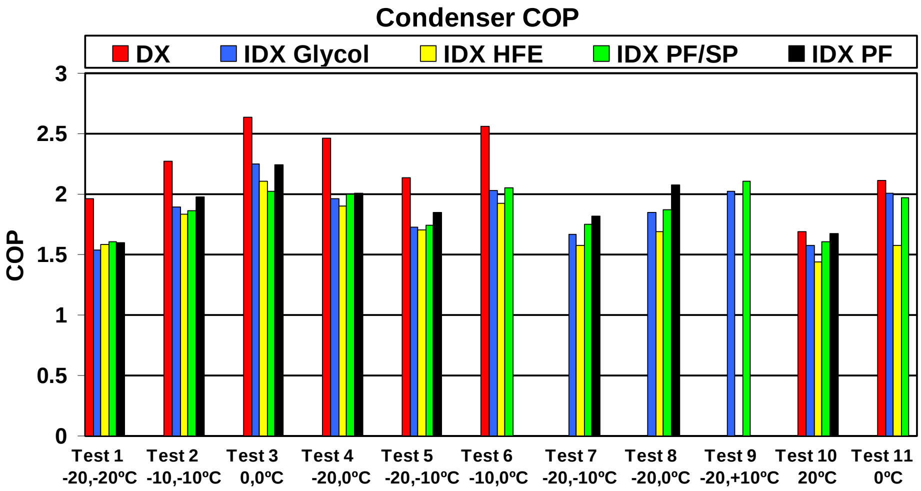

Figure 10. Condenser COP.

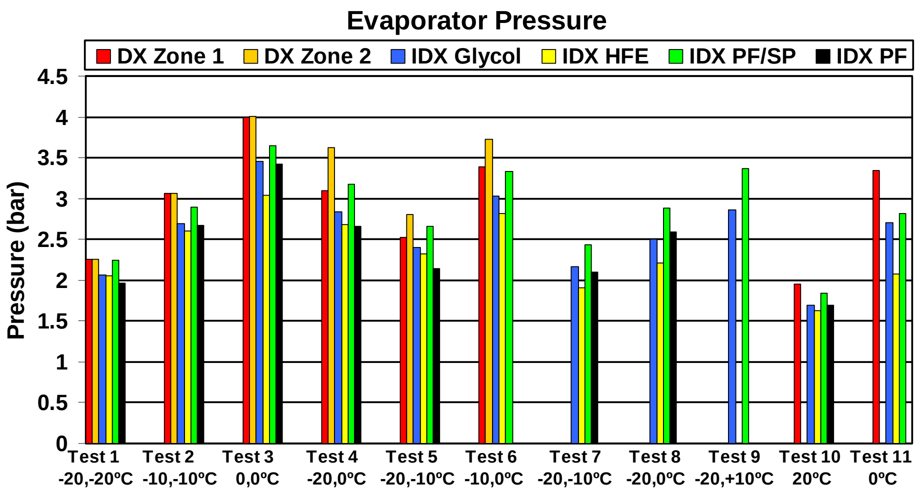

Figure 11. Evaporator Pressure.

The evaporator pressure is depicted in Figure 11. It is evident from these results that the evaporator pressure varies with the choice of secondary working fluid. The highest evaporator pressure is noted for potassium formate/sodium propionate. This is due to the low viscosity and high specific heat capacity at low temperatures which minimises temperature glide and results in a relatively high evaporator saturation temperature. Comparable evaporator pressure is noted relative to the DX baseline at low temperatures. A slight reduction in compressor discharge temperature (Figure 12) is also evident.

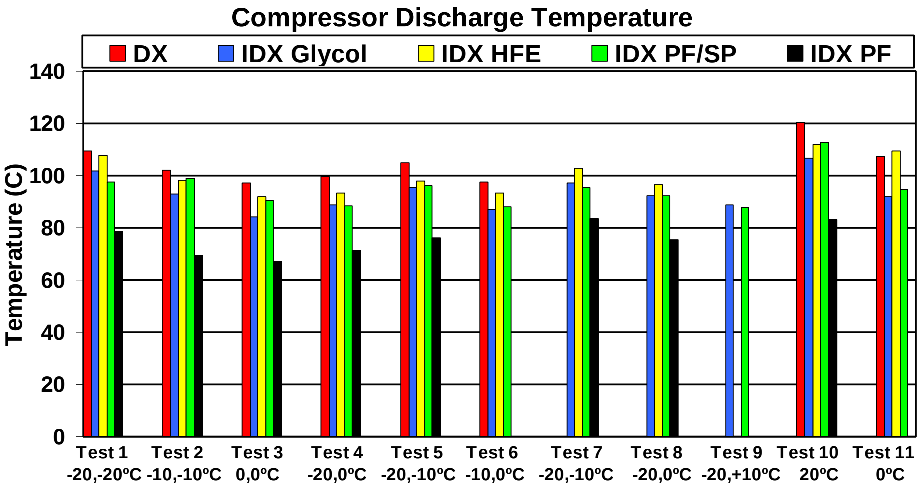

Figure 12. Compressor Discharge Temperature.

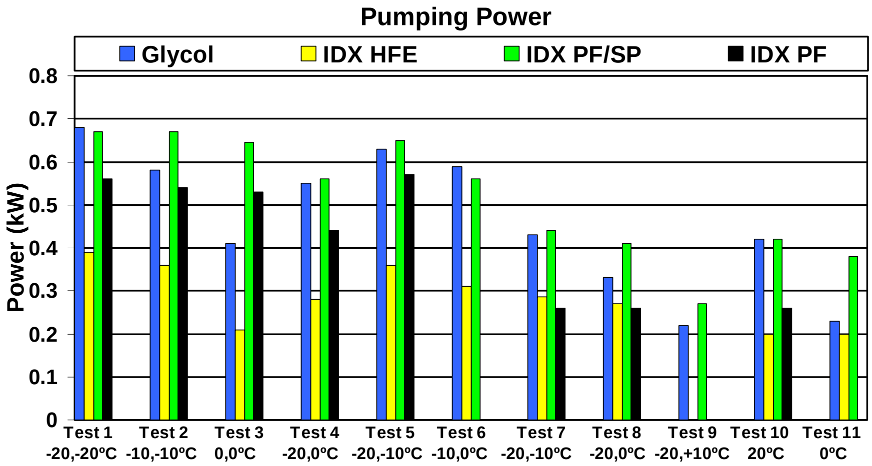

Figure 13. Pump Power.

Considering the pumping power in Figure 13, it can be seen that the lowest pumping power is evident for HFE, due to the low dynamic viscosity of this fluid. The highest pumping power is noted for ethylene glycol at -20°C. This would suggest that dynamic viscosity is an important parameter concerning pumping energy and COP. Across the operating range, the highest averaging pumping power is evident for the potassium formate/sodium propionate solution. This is due to the relatively high mass flow rates resulting in increased pumping power. However, the associated high heat transfer rate also contributes to its better COP.

8. CONCLUSIONS

A test facility was designed and constructed to meet the requirements of an experimental programme for analysis of IDX systems under ATP conditions. The rig was required to emulate an ATP Class C multi-temperature vehicle. Under the ATP test condition, the maximum cooling capacity uncertainty for the evaporator and condenser were ± 4.4% and ± 3.8% from the absolute values calculated. These values were in accordance with the energy balances standard for refrigerated transport (IIR 1995, ATP 2003). From the tests carried out, it was found that potassium formate/sodium propionate displays good refrigeration capacity relative to ethylene glycol and hydrofluoroether across the ATP range. However, the relatively high pumping power associated with high mass flow rates at elevated set-point temperatures can result in reduced COP. The low viscosity exhibited by hydrofluoroether facilitates low pumping power particularly at low set-point temperatures. At higher set-point temperatures, its performance is adversely affected, which can be attributed to its poor specific heat capacity. Overall, it would appear that system performance is highly dependent on the selection of a secondary coolant, however additional factors such as cost, material compatibility and corrosion, weight and quantity should be considered and this will influence any design decisions.

ACKNOWLEDGEMENTS

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