MICRO-SCALE COMPATIBLE REFRIGERANT

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Abstract: The advancement of circuit design and manufacturing has led to faster and more compact microchips. This in-turn has increased the need for steady and fast heat dissipation. This paper looks at refrigerants compatible with micro technology to pave ways for a next generation cooling system that can be integrated with circuits and power optics (lasers). With the effect of the Kyoto and Montreal protocols, the use of flouro-carbon based refrigerants have to be reduced and subsequently phased out. Therefore, only natural refrigerants are considered in agreement with the protocols. The three natural refrigerants under study are ammonia (R717), water (R718) and carbon dioxide (R744). The criteria for the selection process for a micro-scale refrigerant, which includes: volume flow rate, power consumption, coefficient of performance and safety issues are discussed in some detail.

1. INTRODUCTION

Carbon Dioxide (CO_2) was a well known and widely accepted refrigerant in the early 1900's, but its popularity reduced with the introduction of fluorocarbons. The revival of CO_2 as a refrigerant started over a decade ago in Europe with the work of Dr. Gusav Lorentzen and Dr. Jostein Petterson [1]. This sudden rediscovery was invoked by growing environmental concerns of global warming and ozone depletion.

CO_2 has some very attractive properties, which makes it destined to be used as a working fluid. It is non-toxic, non-flammable, non-ozone depleting, has good heat transfer properties, a high volumetric capacity, it is easily available and economic. However its critical temperature is 31.1^{o}C, which is generally lower than the heat rejection temperature of a typical refrigeration and air conditioning system. Thus, wherever the heat rejection temperature is greater than the critical temperature, CO_2 must operate in a transcritical cycle, i.e, with a sub critical low-side pressure and a supercritical high side pressure.

2. TRANSCRITICAL CYCLE

The critical temperature is the temperature above which there is no clear distinction between liquid and gaseous phase. As the critical temperature is approached, the properties of the gas and liquid phases become the same. Above the critical temperature, there is only one phase (supercritical fluid) that is characterized by density and no latent heat effects. The critical pressure is the vapor pressure at the critical temperature. CO_2 has a critical pressure of 7.38 MPa at the critical temperature of 31.1^{o}C.

In a normal refrigeration cycle, the gas from the compressor outlet is condensed in the condenser, by removing latent heat of condensation. In a transcritical CO_2 cycle, the discharge pressure of the compressor is above the critical point, where heat transfer cannot take place by phase change (condensation). In such a cycle, the gas from the compressor is cooled in a gas cooler, causing the density of the gas to increases, while temperature decreases. The supercritical temperature and pressure are not coupled, so they can be optimized, giving an additional degree of freedom.

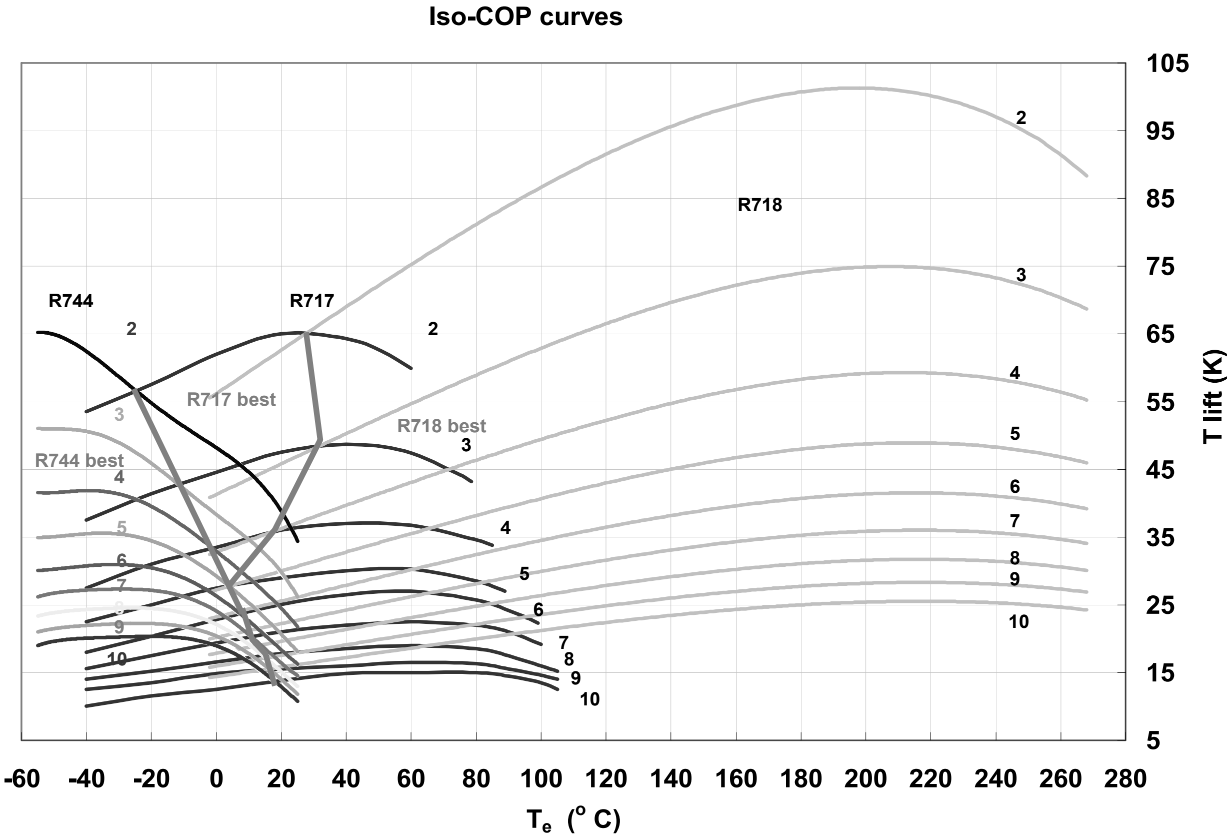

Figure 1: Iso-COP curves of R744, R717 and R718 in an ideal cycle

3. COMPARISON OF NATURAL REFRIGERANTS

In this section we compare the COP and volume flow rates of ammonia (R717), carbon dioxide (R744) and water (R718). It has already been derived that the thermodynamic and the transport properties of CO_2 are comparable with other refrigerants. [1]

Compared to R717 and R718, R744 shows always the best COP in the temperature range -55°C to -35°C, for any temperature lift of up to around 55 K. With degrading temperature lift down to 15K, this range extends to an evaporator temperature of about 15°C.

The volume flow rate of CO2 is considerably lower than that of other refrigerants, which renders it ideal for miniaturization. For an evaporator temperature of 20 °C, CO_2 has a volume flow rate approximately 10 times lower than the nearest competitor (R-22).

Studies comparing CO_2 with other refrigerants have shown that the risk of the high pressure in a CO_2 system is negated by the fact that the energy contained in the system is relatively lower than that in a R-22 system owing to the lower volume and refrigerant charge.[1]

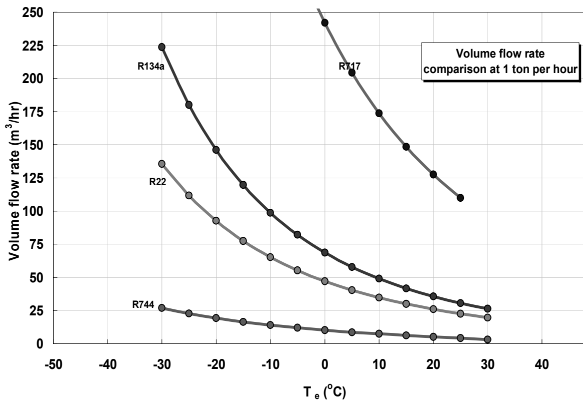

Figure 2: Volume flow rate comparison for 1 ton per hour refrigeration

The high pressure, low flow rate also allows for the design of small diameter tubing or even micro-channel cooling. This property can well be exploited for micro applications like cooling of electronics and lasers systems. While due to the high pressures involved, the use of conventional material like silicon for the microchannels can be challenging, diamond microchannels can be an alternative. Diamond has good thermal conductivity and is already being used as a heat sink in micro chips. Using diamond, it is possible to attain high fin efficiency for heat sinks.[2] However, pressure tests are yet to be done on diamond micro-channels for pressure ranges encountered in the CO_2 system. Research done on stainless steel and aluminum microchannels have shown that due to the reduced diameter they are able to withstand the high pressure.[2]

4. PRESSURE RATIO

In the reversed Rankine cycle the compressor outlet pressure is dictated by the condensation pressure corresponding to the condenser temperature. In the transcritical cycle, the compressor outlet pressure can be varied to any value above the critical pressure. However, there exists an optimum compressor discharge pressure corresponding to the gas cooler outlet temperature. Since the cycle has no subcooling, the evaporator temperature dictates the compressor suction pressure.

Combining the optimum pressures corresponding to the gas cooler temperature and the evaporator temperature, the compressor pressure ratio can be calculated. These results are consistent with those presented by Dr. Liao, Zhao and Kakobsen of the Hong Kong University of Science and Technology, which where limited to very small temperature range. [3]

It is seen that although R744 operating pressure is much higher than that of other refrigerants, the pressure ratio of the compressor is comparatively small. This is very beneficial if micro compressors are to be integrated. Also the low pressure ratio results in much higher isentropic efficiency if designed and produced with same quality (polytropic efficiency).

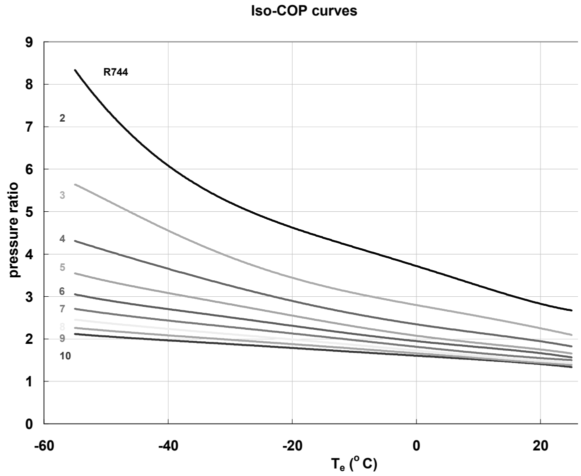

Figure 3: Iso-COP lines showing pressure ratio at various evaporator temperature

5. COMPRESSOR EFFICIENCY

In this section the effect of polytropic efficiency of the compressor on the cycle performance is investigated. However, the polytropic efficiency is not directly linked to the COP. To link the two, the isentropic efficiency of the compressor is calculated. For this, the polytropic exponent is related to the isentropic one.

The polytropic exponent n can be calculated from the isentropic exponent γ for various polytropic efficiencies:

$$\frac{n-1}{n} = \frac{\gamma - 1}{\gamma}\left(\frac{1}{\eta_{poly}}\right)$$

With the polytropic exponent, the polytropic work of the compressor is:

$$W_{isentropic} = R_{CO_2} T_1 \left(\frac{n}{n-1}\right)\left(\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right)$$

Hence the isentropic efficiency of the compressor corresponding to the polytropic efficiency can be calculated. as a function of the pressure ratio.

$$\frac{W_{is}}{\eta_{is}} = \frac{W_{poly}}{\eta_{poly}}$$

Figure 4 shows the iso-COP lines, showing variation of efficiency with evaporator temperature (which is a function of the pressure ratio).

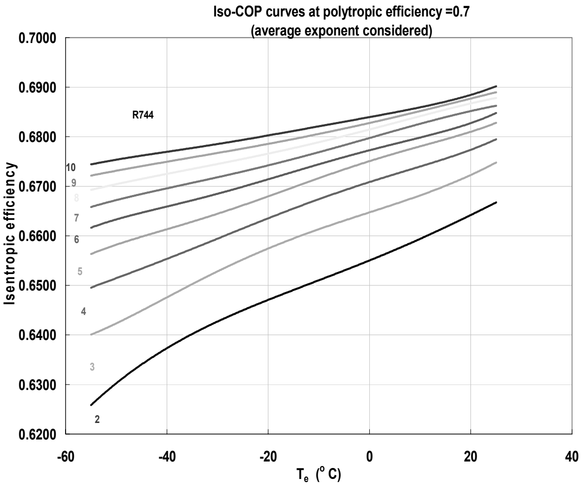

Figure 4: Isentropic efficiency vs. evaporator temperature for a ploytropic efficiency of 70 %

Using the definition of COP, we can now calculate the COP incorporating the isentropic efficiency of the compressor. Figure 5 shows the iso-COP lines, the iso-pressure ratio lines and the isentropic efficiency lines corresponding for evaporator temperature vs. temperature lift considering 70% polytropic efficiency of the compressor.

6. CONCLUSION

The study presents COP maps for conventional and transcritical R744 (CO_2) systems. It also provides the optimum pressure ratios for a wide range of gas cooler or condensation temperatures and evaporator temperatures as they have been used in this work. For certain compressor quality (polytropic efficiency) maps are generated that summarize the compressor requirements in terms of pressure ratio and isentropic efficiency, along with the obtainable COP for a wide range of combinations of evaporator temperature and temperature lift.

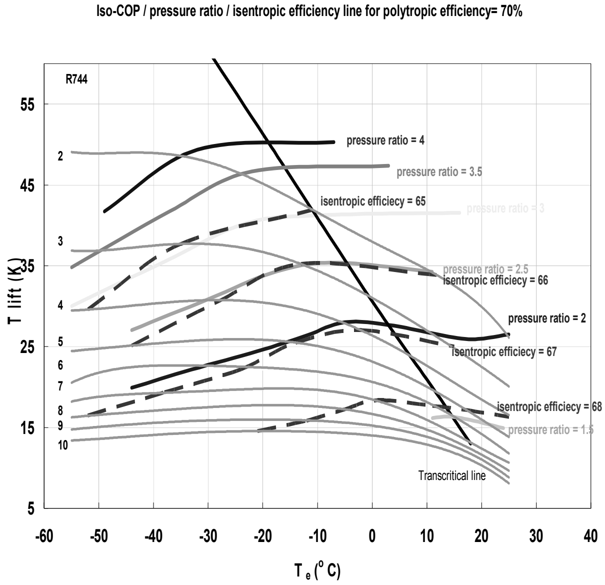

Figure 5; Iso-COP lines, iso-baric lines and isentropic efficiency for a ploytropic efficiency of 70 %

It can be concluded that R744 is a viable natural refrigerant, especially for lower temperatures or moderate temperature lifts. The low volume flow rate of R744 is very small as are also the pressure ratios. This combination renders R744 as an ideal refrigerant for micro-scale cooling systems.

7. REFERENCES

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