Design analysis of a hybrid jet-pump CO_2 compression system

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ABSTRACT: Transport refrigeration contributes to anthropogenic global warming directly because of leakage of refrigerant, usually using high global warming potential (GWP) refrigerants, and indirectly because of the greenhouse gases emitted in driving the vehicle and the refrigeration system. A hybrid jet-pump CO_2 compression system is being designed for transport refrigeration so that the GWP of the system is reduced and its performance improved. The jet-pump utilises waste heat from the exhaust gases of the engine to subcool the refrigerant and so enhance performance, reduce energy required from the engine and minimise GWP of the system. The hybrid jet-pump CO_2 compression system has been simulated and its performance determined for different operating conditions and optimised using entropy generation minimisation. At an evaporator temperature of -18°C, an ambient temperature of 35°C and a generator temperature of 120°C, the COP increases from about 1.0 to 2.27 as the degree of subcooling increases from 0K to 20K. Similarly, compressor work is reduced by 24% at 20K subcooling. The optimum degree of subcooling was approximately 10K for the operating conditions described above. An improved COP is achieved whilst the size of heat exchangers required to operate the jet-pump are minimised with respect to the overall weight of the system and thus its impact on indirect emissions.

Keywords: CO_2, ejector, transport refrigeration, subcooling, GWP

NOMENCLATURE

COP	coefficient of performance	
h	specific enthalpy	(J/ kg)

$\dot{m}$	mass flow	(kg/s)
p	pressure	(Pa)
$\dot{Q}$	heat transfer	(W)
s	specific entropy	(J/kg/K)
T	temperature	(°C)
ΔT	Temperature difference	(K)
$\dot{W}$	work transfer	(W)

Subscripts and superscripts

1,2,….18	state points in cycles
0	Environmental condition
c	condenser
com	compressor
e	evaporator
e-ej	jet-pump evaporator
ej	jet-pump system
g	generator
gc	gas cooler
p	primary
pc	precooler
pump	pump
sc	subcooler
vc	vapour compression system

1. INTRODUCTION

Mankind has developed the cold chain to store and preserve food and to get it from the farm or the ocean to the plate in as high a quality as possible. As living standards have improved and expectations of quality have increased, the requirement for safe, secure and fresh food has grown. However, energy is required to remove the heat to maintain a given temperature and the majority of the refrigerants used have proved to be damaging to the environment when they have been released. The indirect impact of the energy consumed and the direct impact due to leakage contributes to environmental degradation including an anthropogenic warming effect.

Transport refrigeration is one of the most crucial parts of the cold chain, one of the most vulnerable parts through leakage and energy consumption and one with the most potential for innovation. Between 10 and 25% of the energy used in refrigerated road vehicles is by operating the refrigeration system, according to the DEFRA report [1] on greenhouse gas impacts of food retailing. Refrigerant leakage ranges from 10% to 37% per annum, according to the IPPC/TEAP report [2] on safeguarding the ozone layer and the global climate system. Transport systems are vulnerable because of the extreme temperature and weather conditions over which the system may operate and the vibrations, shocks, collisions, and maintenance and charging regimes associated with the systems as described by Tassos *et al* [3].

This paper concentrates on road transport which can range from small refrigerated vans delivering products locally to medium trucks and large container lorries. The small vans and trucks tend to drive their refrigeration systems from a compressor driven from the engine through a vee-belt. The larger trucks and lorries will often have an independent diesel engine separate from the traction unit of the vehicle that drives a compressor and is housed in front of the compartment or underneath the trailer.

The most common refrigerant in transport systems is R404a, however the global warming potentials compared with a reference 1 kg of CO_2 time-integrated over 100 year time period (GWP_{100}) are 3260.

The two characteristic aspects of transport refrigeration that make it a problem, namely high energy consumption and leakage of high GWP refrigerants, also presents engineers with an opportunity to convert these problems into assets and to exploit them. This paper describes a refrigeration system that uses a natural refrigerant and exploits the waste heat from the engine to enhance performance both of the prime mover and the refrigeration system.

2. A HYBRID JET-PUMP AND CO_2 VAPOUR COMPRESSION SYSTEM

CO_2 is a natural refrigerant with a GWP_{100} of 1 and has become increasingly popular as an alternative to HFCs since the 1980s when Lorentzen and Pettersen [4] demonstrated the advantages in mobile air conditioning (MAC) systems. As the critical temperature of CO_2 is 31.1°C, heat rejection usually takes place above this temperature. In MAC and heat pump systems, where weight and space are important considerations, CO_2 systems discharge in the supercritical region and are said to operate in a transcritical cycle because evaporation is in the sub-critical region and heat rejection is in the supercritical region. At high ambient conditions, discharge pressures and temperature are high, efficiency is reduced and problems with heat transfer at temperatures close to the critical point can be experienced.

A jet-pump refrigeration system is proposed to extract heat from the discharge gases and vehicle exhaust and subcool the CO_2 system. Subcooling increases the refrigeration effect and reduces the gas cooler outlet temperature below the critical point and so improves heat transfer. This report describes simulation results carried out to determine the performance of the hybrid system over a range of operating conditions to provide theoretical benchmarks to assess the system and to design a system for laboratory testing.

2. 1 Operation of the hybrid jet-pump and CO_2 compression system

This section describes some of the features of the hybrid jet-pump CO2 compression system, but for detailed analysis of CO2 systems the reader is directed to Sung *et al* [5] and Kim *et al* [6], and for those interested in jet-pump systems the reader is directed to Eames *et al* [7].

Figure 1 shows a simplified circuit diagram of the proposed hybrid jet-pump and CO_2 vapour compression refrigeration system.

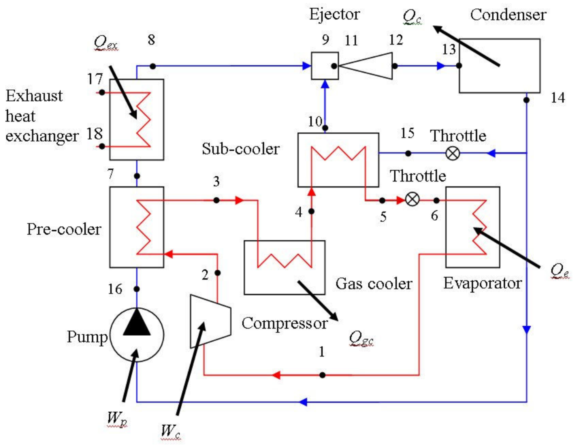

Figure 1. The CO_2 vapour compression-jet-pump refrigeration system

Some typical operating conditions and corresponding *T-s* diagrams of the two systems are shown in Figures 2 and 3. The CO_2 circuit is shown by the red and the jet-pump circuit is shown in blue. A thermodynamic analysis of the cycle has been carried out to determine the operating conditions. Conservation of energy, momentum and continuity were applied to the processes and the equations were solved using the software package Engineering Equation solver (EES) developed by Klein [8].

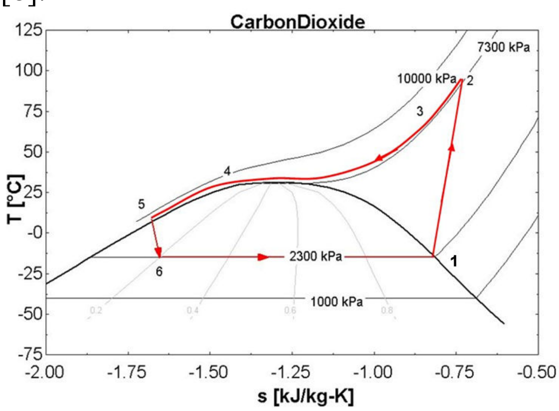

Figure 2. *T-s* diagram of the vapour-compression CO_2 cycle.

The jet-pump circuit working fluid in this example is methanol, which is one of the candidates being considered as the working fluid in the ejector system.

In the CO_2 sub-cycle, the refrigerant leaves the evaporator at state 1 and its pressure is increased by a compressor to state 2.

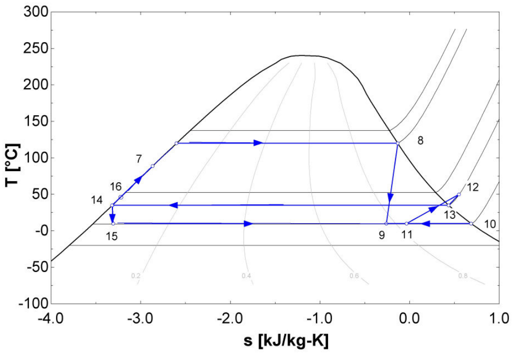

Figure 3. *T-s* diagram of subcooler jet-pump cycle.

In the supercritical region temperature and pressure are independent, Inokuty [9] reported that there is an optimum COP dependent on discharge pressure for a constant gas cooler outlet temperature. In this case, at a gas cooler outlet temperature of 35°C, the compressor discharge pressure and temperature that optimise COP are 7330 kPa and 92 °C, respectively. In Figure 2, the refrigerant has been subcooled by 25 K to 10°C. The refrigerant will not condense in the supercritical region and so it is assumed that cooling takes place at constant pressure, with a temperature glide from state 2 through to state 5 in Figure 2. The refrigerant leaves the pre-cooler at state 3 at a reduced temperature and then enters the gas cooler. Further heat is rejected to the surroundings through the gas cooler. The refrigerant leaves the gas cooler at state 4. The refrigerant enters the subcooler where heat is rejected to the jet-pump circuit from state 14 to state 10 in Figure 3, and the CO_2 is cooled below the ambient temperature and the critical temperature of the refrigerant. The refrigerant leaves the subcooler at state 5 and flows through a throttle where it is assumed to expand isenthalpically to the pressure and temperature of the evaporator. The evaporator absorbs heat from the cooling load causing the refrigerant to vaporise, leaving the evaporator as a dry saturated vapour at state 1.

In the jet-pump sub-cycle shown in Figure 3, liquid refrigerant at state 14 is pumped at constant temperature to a pressure corresponding to the pre-cooler pressure at state 16. The refrigerant enters the pre-cooler where heat is transferred from the CO_2 circuit gaining sensible heat leaving the pre-cooler at state 7. The refrigerant enters the exhaust heat exchanger where heat from the exhaust gases of an engine is used to vaporise the refrigerant. The refrigerant leaves the exhaust heat exchanger as a dry saturated vapour at state 8. The refrigerant enters the ejector where it expands to the pressure in the subcooler and accelerates to supersonic velocity at state 9. Vapour from the subcooler is entrained by the supersonic jet, causing a reduction in pressure and temperature. Heat is absorbed from the CO_2 sub-system from states 4 to 5 to balance the cooling effect produced by the ejector. The two streams mix and leave the ejector as a superheated state at 12. The mixed stream is cooled to a dry saturated vapour at 13 and then condensed to a saturated liquid at state 14. A proportion of the liquid refrigerant is returned to the subcooler through a throttle where it expands to the pressure and temperature of the subcooler at state 15. The remainder is re-circulated to the pre-cooler and the exhaust heat exchanger by the pump.

2.2 Exploitation of exhaust gas temperature

A special case of refrigerated transport presents an opportunity to exploit the exhaust gases from an independent diesel engine that is the system used in the majority of medium to large refrigerated vehicles. The temperature of the exhaust gases from these engines range between 300°C and 500°C and the heat available can vary from 2kW upwards depending on the cooling capacity and hence the engine power output. The optimum jet-pump performance depends on the generator temperature available and the degree of subcooling required. A parametric study of the performance of a jet-pump at varying generator and subcooler conditions was carried out.

3. PERFORMANCE OF HYBRID SYSTEM

Traditionally, thermodynamic systems have been analysed using the first law of thermodynamics and it has been successful in providing quantitative results. However, it does not inform us of the qualitative performance of the system. Since the early 1980s, analysis using the second law of thermodynamics has been developed in such fields as cryogenics and solar energy utilisation where the minimisation of losses and the optimisation of systems are crucial to effective use of energy. Various methods have been developed using the second law according to Bejan [10]. These include exergy analysis, thermodynamic optimisation, entropy generation minimisation, and available work.

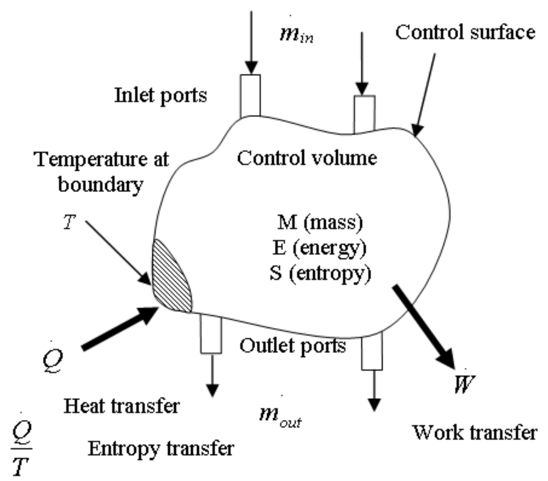

Figure 4. Diagram of open system

3.1 First and second law analysis

Figure 4 shows a diagram of an open thermodynamic system with energy and mass flow interactions across the system boundary.

For an open system neglecting changes in elevation and velocity, the first law of thermodynamics gives;

$$\frac{\partial E}{\partial t} = \sum_{in} \dot{m}h - \sum_{out} \dot{m}h + \dot{Q} - \dot{W} \qquad [1]$$

And the second law gives;

$$\frac{\partial S}{\partial t} \geq \sum_{in} \dot{m}s - \sum_{out} \dot{m}s + \frac{\dot{Q}}{T_0} \qquad [2]$$

The rate of entropy generation is;

$$\dot{S}_{gen} = \frac{\partial S}{\partial t} - \frac{\dot{Q}}{T_0} - \sum_{in} \dot{m}s + \sum_{out} \dot{m}s \geq 0 \qquad [3]$$

Combining equations 1 and 3 gives the maximum shaft power available;

$$\dot{W}_{\max} = \sum_{in} \dot{m}(h - T_0 s) - \sum_{out} \dot{m}(h - T_0 s) \qquad [4]$$

The quantity (h-$T_0 s$) is a thermodynamic property of the system when the environmental temperature T_0 is specified and is referred to as the availability function for steady flow after Keenan [11]. Maximum work available is equal to the actual work plus the loss in the available work due to irreversibilities in the system. Therefore the loss in the available work is defined as

$$\dot{W}_{lost} = \dot{W}_{\max} - \dot{W} \qquad [5]$$

Substituting equations 4 and 1 into 5, assuming steady state and neglecting changes in velocity and elevation, gives

$$\dot{W}_{lost} = T_0 \left(\sum \dot{m}s_{out} - \sum \dot{m}s_{in} - \frac{\dot{Q}}{T_0} \right) \qquad [6]$$

This can be easily recognised as

$$\dot{W}_{lost} = T_0 \, \dot{S}_{gen} \qquad [7]$$

This equation is known as the Gouy-Stodola theorem. It shows that the lost available work is directly proportional to the rate of entropy generation of an open system in contact with an environment at T_0.

3.2 Coefficient of performance

The coefficient of performance of the sub-cycles and the system are determined from 1st law analysis of the systems.

The system COP is the ratio of the cooling capacity to the compressor work input, the pump work input and the heat input to the generator.

$$COP_{sys} = \frac{\dot{Q}_e}{\dot{W}_{com} + \dot{W}_{pump} + \dot{Q}_g} \quad [8]$$

The COP of the mechanical system is the ratio of the cooling capacity to the compressor work input and the pump work input.

$$COP_{mech} = \frac{\dot{Q}_e}{\dot{W}_{com} + \dot{W}_{pump}} \quad [9]$$

It neglects the generator heat input because this is recovered heat from the engine and therefore demonstrates its performance compared with standard vapour-compression systems.

3. SIMULATION RESULTS OF HYBRID SYSTEM

A parametric study of the performance of a jet-pump at varying generator and subcooler conditions was carried out for constant gas cooler outlet temperature, evaporator temperature and cooling capacity of 35°C, -15°C and 3kW, respectively. The following analysis describes the changes Figure 5 shows how entrainment ratio varies with the degree of subcooling. As the degree of subcooling increases, entrainment ratio decreases, with a consequent reduction in COP. At a generator temperature of 90 °C, entrainment ratio is 0.856 at a degree of subcooling of 5 K. Entrainment ratio decreases to 0.09 as subcooling increases to 20 K. As the generator temperature increases, entrainment ratio increases for the same degree of subcooling.

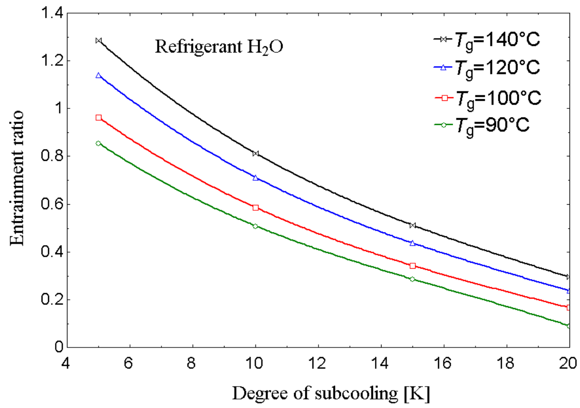

Figure 5. Variation in entrainment ratio with the degree of subcooling, at constant generator temperatures

At 5 K, entrainment ratio increases from 0.856 to 1.286 when the generator temperature increases from 90 °C to 140 °C. Similarly, at 20 K, entrainment ratio increases from 0.09 to 0.291 with the same increase in generator temperature. The consequence of increasing the degree of subcooling is to increase the generator heat transfer area required because of the increase in primary flow at constant secondary flow and also increasing the required condenser heat transfer area because of the increase in generator heat input that requires rejection to atmosphere. The diagram demonstrates that for low degrees of subcooling moderate generator temperatures are adequate but at large degrees of subcooling high values of generator temperature would be required to achieve adequate performance.

Figure 6 shows the variation in COP_{sys} and COP_{mech} with generator, gas cooler outlet and evaporator temperatures and entrainment ratio held constant.

At 0 K subcooling, representing the COP of the vapour compression system alone, the value is approximately 1.30.

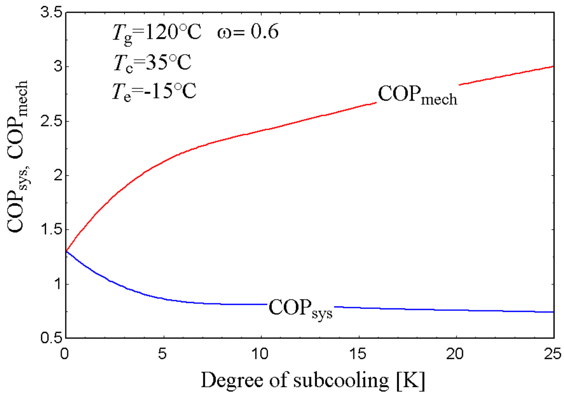

Figure 6. Variation in COP_{sys} and COP_{mech} with degree of subcooling.

As the degree of subcooling increases, COP_{sys} decreases whilst COP_{mech} increases. At 10 K subcooling COP_{sys} has decreased from 1.30 to 0.81 compared to an increase in COP_{mech} from 1.30 to 2.41. From Figure 6, COP_{sys} decreases rapidly with subcooling to about 10 K subcooling then flattens off, whilst COP_{mech} continues to rise beyond this point. At subcooling greater than about 10 K COP_{sys} only decreases marginally whereas COP_{mech} continues to increase. At subcooling greater than this value, the heat transfer area would need to be increased to handle the extra heat input but the overall benefit would be small.

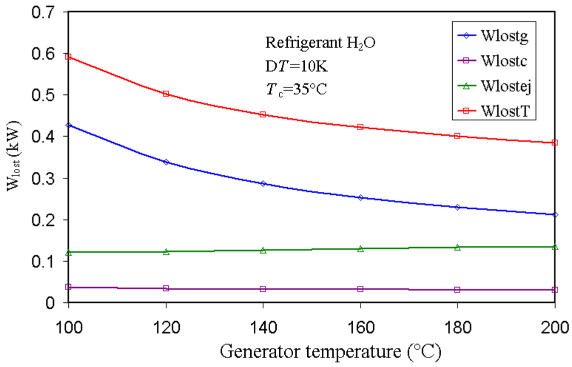

Figure 7. Variation in lost available work for individual components and overall

Figure 7 shows how the lost available work from individual components and the overall loss varies with generator temperature. The total lost available work decreases from 0.6 kW to 0.4 kW as the generator temperature increases from 100 °C to 200°C. The ejector loss in available work increases slightly but the generator loss in available work decreases significantly with increasing temperature. It can also be seen that the lost available work of the generator is a substantial proportion of the total, at approximately 80% of the total loss in available work at 100 °C. The proportion drops to just over 45% at 200°C. In contrast, the lost available work from the condenser remains almost constant with increase in generator temperature. The proportion of the total lost available work is almost constant at about 5%. The lost available work from the ejector increases from about 23% at 100 °C to about 44% at 200 °C, showing that the lost work is being more evenly shared between the generator and the ejector.

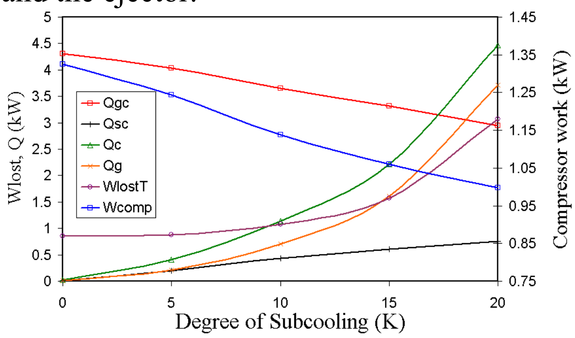

Figure 8. Variation in heat transfer across gas cooler, sub-cooler, condenser and generator, lost work and compressor work.

Figure 8 shows the variation in loss in available work and component heat transfer with degree of subcooling. At 0K of subcooling compressor work is 1.325 kW, the same as a vapour-compression system alone. As the degree of subcooling increases, the work required to drive the CO_2 system decreases. At 10 K of subcooling, the compressor work is approximately 1.138 kW, a reduction of approximately 14%. At 20 K of subcooling, compressor work is approximately 0.998 kW, a reduction of approximately 25%. Cooling duty of the gas cooler is reduced from 4.308 kW to 2.941 kW at 20K subcooling, a reduction of 32%. The total loss in available work increases slowly from 0 K where it is 0.8 kW. At 10 K subcooling, the lost available work rises to 1.1 kW, an increase of 37.5%. Total lost available work increases more rapidly beyond 10 K of subcooling. At 20 K, the total lost available work has risen to 1.6 kW.

4. CONCLUSIONS

The main source of heat available for the vapour generator is from the exhaust gases. For a given generator and evaporator temperature, the COP of the system decreases until at subcooling in excess of 10K, the COP only varies a small amount. The COP of the system neglecting the heat input to the generator increases. This shows that for a given cooling load, the work input is decreasing. For a given condenser and evaporator temperature, COP increases with increase in generator temperature. A compromise between enhanced performance of the vapour-compression system due to an enhanced refrigeration effect and reduced compressor work, and reduced overall performance due to the addition of heat exchange equipment for heat rejection to ambient shows that for a hybrid system operating at a gas cooler outlet temperature of 35 °C, and an evaporator temperature of -15 °C, the jet-pump system should be designed to give a degree of subcooling of 10 K at a generator temperature of 120 °C. This would increase the COP of the system by a factor of 1.85. Second law analysis showed that at subcooling beyond 10 K, the loss in available work increased rapidly, which was reflected in the changes in system COP with subcooling.

ACKNOWLEGEMENTS

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