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Design Optimization of R744 Ejector for Compressor Oil Pumping

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ABSTRACT

Standard two-phase refrigerant ejectors have been widely considered for the recovery of expansion work in vapor compression refrigeration systems so far. However, one innovative way through which ejectors could be utilized in HVAC&R systems is to pump the compressor lubricant logged in the evaporator as well as other system components and ensure improved oil return to the compressor with time. In this regard, the design of an ejector dedicated to oil pumping is crucial and the design guidelines could be very different from that of the standard two-phase ejector. For design procedures of an oil ejector, no literature study has ever been reported. This paper presents an approach for gaining insights about the design of ejectors dedicated to oil pumping considering a suitable ejector cycle. This unconventional ejector cycle might be very important for some immiscible refrigerant-lubricant combinations concerning HVAC&R systems. Initially, numerical modeling has been performed regarding this ejector model for compressor lubricant pumping. Preliminary modeling results reveal insights about the bypass flow required to pump the compressor oil and ensure oil return to the compressor. An experimental layout of the oil entrainment test setup is designed and integrated with the existing test facility for the standard two-phase ejector cycle. For initial test experiments, standard two-phase R744 ejectors can be used to get insights about the oil entrainment capability for fixed motive flow rates. In the next step, different motive flow rates and pressures are considered in a wide range of operating conditions to assess the oil entrainment capability through the ejector of fixed geometry conditions. Depending upon the results from these sets of experiments, the ejector design for oil pumping can be optimized.

1. INTRODUCTION

The throttling process occurs in the vapor compression refrigeration systems for the expansion of the high-pressure refrigerant from the condenser or gas cooler to the low-pressure evaporator. Some of the most widely used expansion devices in a vapor compression refrigeration system are capillary tube, thermostatic expansion valve (TXV), electronic expansion valve (EEV), etc. However, the energy during this throttling process is lost unless it is recovered by some other means. As this energy is lost during the throttling process, it impacts the overall system performance for a vapor compression system. Consequently, the cooling capacity and Coefficient of Performance (COP) become lower than that of the Carnot vapor compression refrigeration cycle. Over the last few decades, some approaches have been considered to improve the system performance in HVAC&R systems. However, ways for the expansion work recovery seem to be promising for achieving better system performance. In this regard, one way to recover some of the expansion work and utilize this energy for the system performance improvement is to use an ejector. So far two-phase refrigerant ejectors have been widely considered in the case of different refrigerants, for example, R744, R134a, R410A, R1234yf, etc. Ejectors are a simple piece of equipment, and it requires no additional energy consumption. An ejector works on the principle of momentum transfer between two streams of fluid: motive fluid and suction fluid. Figure 1 illustrates the working principle of a standard ejector. Typically, a high-pressure motive fluid expands in a converging-diverging nozzle or sometimes, converging nozzle in the motive nozzle of the ejector. Due to this expansion of the motive fluid, a very high velocity is achieved at the exit of the motive nozzle. The low-pressure state at the motive nozzle exit helps to entrain the suction fluid through the suction chamber of the ejector. Eventually, motive and suction fluids get mixed in the mixing section and their velocities tend to approach each other. Finally, the two-phase mixed-flow flows through the diffuser, and the pressure rise is achieved at the diffuser exit compared to the pressure at the suction chamber inlet.

So far, the improvement potential through an ejector in HVAC&R systems comes from either compressor suction pressure increase (Lucas and Koehler, 2012) or evaporator liquid overfeed (Lawrence and Elbel, 2015). Contemporary research on ejector systems focused on achieving reduced compressor work or better evaporator performance so far. Therefore, most of the design guidelines stated in the contemporary research are suitable for vapor as well as two-phase ejectors. Huang *et al.* (1999) proposed design procedures for predicting the ejector performance in the case of the vapor phase ejector with the help of the ideal gas model. A review study concerning the ejector design guidelines as well as the effect of the ejector geometry in the case of two-phase R744 ejectors was conducted by Lawrence and Elbel (2014). However, there could be some other utilization of ejectors through which it might be beneficial for HVAC&R systems. In this regard, one novel way of ejector utilization could be to pump the compressor lubricant logged in the evaporator and ensure intermittent oil return to the compressor. In HVAC&R systems, most often it becomes very crucial to ensure sufficient oil return to the compressor. The compressor might face reliability issues, and even if it could be damaged if sufficient compressor oil is not returned to it. No contemporary literature has ever reported some design guidelines concerning the oil pumping through an ejector. This paper presents some approaches for design guidelines in the case of ejectors dedicated to oil pumping through a pertinent ejector cycle. Regarding the design of the suction nozzle, numerical modeling is considered to predict the required motive flow rate for the oil ejector operation. Whereas, the motive nozzle of the ejector is designed with the help of an appropriate numerical modeling scheme. Besides, an experimental layout of the oil entrainment test experiment is designed and proposed for gaining insights about the design optimization in the case of oil ejector geometry.

2. MODELING PROCEDURE OF EJECTOR CYCLE FOR OIL PUMPING

2.1 Ejector Cycle Dedicated to Oil Pumping

Kim *et al.* (2016) patented an ejector cycle that could be utilized for oil pumping. The schematic diagram of this ejector cycle for oil pumping is illustrated in Figure 2. In this ejector cycle, a certain portion of the compressed vapor is bypassed from the compressor exit (point 2) and fed as the motive flow to the ejector motive nozzle (point 5). An oil separator is used downstream of the compressor (point 2s) to separate the oil from the refrigerant vapor. During the normal operation of the system, some oil escapes the oil separator and flows through the condenser/gas cooler and other system components. Eventually, some oil gets deposited in the evaporator as well as other system components during the normal system operation. The high-pressure motive fluid helps to entrain accumulated oil from the evaporator through the suction chamber (point 6) of the ejector. Refrigerant vapor and oil get mixed with each other and finally discharged through the ejector discharge channel (point 7). In this ejector cycle, an internal heat exchanger (IHX) is utilized to allow the heat exchange between the high-temperature oil separated from the oil separator return channel (point 2c, 2d) and the low-temperature two-phase refrigerant/lubricant mixture from the ejector discharge channel (point 7, 8). Through this heat exchange, the two-phase refrigerant gets vaporized and only vapor refrigerant enters into the compressor. Meanwhile, oil can return to the compressor inlet (point 1) from both oil separator return channel as well as by being pumped out of the evaporator thus ensuring sufficient oil return to the compressor.

2.2 Numerical Modeling of the Ejector Cycle for Oil Pumping

Numerical modeling of the ejector cycle dedicated to oil pumping is performed considering suitable assumptions. Most often, the commonly used compressor lubricant with CO_2 in HVAC&R applications is PAG or POE oil. This paper represents a steady-state operation of the transcritical CO_2 ejector refrigeration cycle considering a fixed oil circulation rate (OCR) of 2% (by mass) at the compressor outlet. The evaporator and gas cooler refrigerant temperature are assumed fixed at 10 °C (283 K) and 40 °C (313 K), respectively for the considered modeling scheme. However, in the considered numerical model, engine oil properties are used to obtain the specific entropy for simplifications as it is assumed that these engine oil properties are very close to that of the PAG or POE oil properties. The evaporator cooling capacity is considered as fixed at 5 kW whereas, an oil separator of 95% efficiency is assumed for the preliminary numerical study. Besides, the gas cooler pressure is considered as 10 MPa in the numerical model for this transcritical R744 ejector cycle modeling. Equations for the ejector components as well as the system equations are iteratively solved through Engineering Equation Solver (EES).

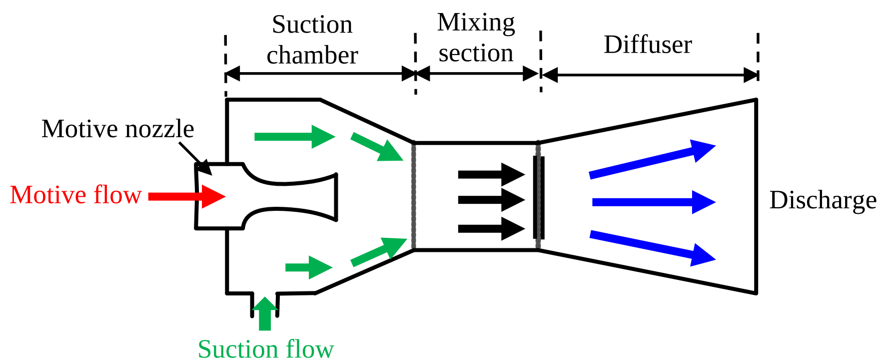

Figure 1: Schematic diagram of a two-phase ejector

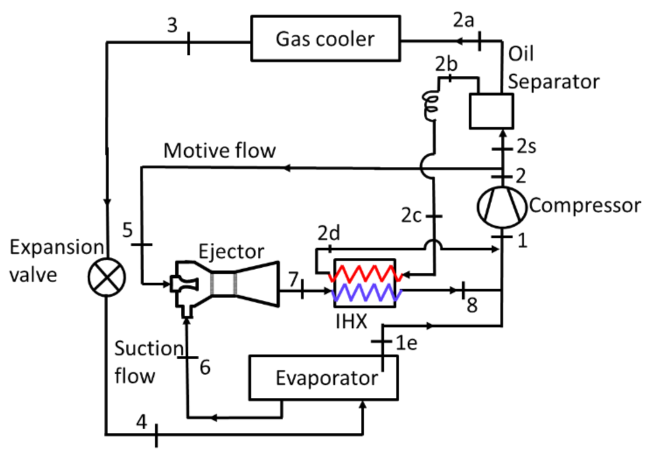

Figure 2: Schematic diagram of the ejector cycle patented by Kim *et al.* (2016)

Oil circulation rate (OCR) is calculated at different points in the ejector cycle through mass conservation equations in the considered numerical modeling scheme. From this numerical model, it is possible to predict the oil accumulation rate at the evaporator for a fixed evaporator cooling capacity.

$$OCR = \frac{\dot{m}_{oil}}{\dot{m}_{oil}+\dot{m}_{ref}} = \frac{\dot{m}_{oil}}{\dot{m}_{total}} \tag{1}$$

It is assumed that the OCR values at the oil separator inlet (point 2s), as well as the motive nozzle inlet (point 5), are the same as the considered fixed 2% OCR at the compressor exit. According to the ejector cycle by Kim *et al.* (2016), a certain portion of the compressor outlet flow is bypassed and fed as the motive flow thus effectively sacrificing some of the cooling capacity for oil pumping. Through the considered numerical modeling scheme, some insights can be gained regarding the amount of bypass flow that is defined as the bypass ratio (r). In the ejector model, for simplicity of calculations, only the refrigerant mass flow rate is considered as the motive mass flow rate.

$$r = \frac{\dot{m}_5}{\dot{m}_2} = \frac{Mass\ flow\ rate\ at\ motive\ nozzle\ inlet}{Total\ mass\ flow\ rate\ at\ compressor\ outlet} \tag{2}$$

The oil separator efficiency (η_{os}) is defined as the similar way that had been defined as the overall separation efficiency by Xu and Hrnjak (2019). The oil separator efficiency is termed as the ratio of the oil flow rate separated by the oil separator to the total incoming oil flow rate at the oil separator.

$$\eta_{os} = \frac{\dot{m}_{oil,2b}}{\dot{m}_{oil,2s}} = \frac{Oil\ flow\ rate\ separated\ by\ the\ oil\ separator}{Total\ incoming\ oil\ flow\ rate\ at\ the\ oil\ separator} \tag{3}$$

The iterative simulation procedure proposed by Kornhauser (1990) is modified for the considered ejector cycle of oil pumping. This modified simulation procedure is applied with the knowledge of reasonable ejector components' efficiency values based upon the pertinent experimental studies. In addition, a compressor having fixed compression efficiency of 90% is considered for the system model. No pressure drop is assumed from the compressor outlet to the motive nozzle inlet as well as from the evaporator to the suction line of the ejector. In the case of the ejector, the motive nozzle inlet properties are considered to be the same as the properties at the compressor exit. Regarding the oil properties, the correlation proposed by Thome (1995) is used for the calculation of the oil's specific heat capacity (c_p) whereas, the specific enthalpy (h_{oil}) and specific entropy (s_{oil}) are obtained with the knowledge of the corresponding temperature.

$$c_p = 4.186\left[\frac{0.388+0.00045(1.8T+32)}{S^{0.5}}\right] \tag{4}$$

$$h_{oil} = \int_{T_{reference}}^{T} c_p(T)dT + h_{reference} \tag{5}$$

Here, temperature and enthalpy at the reference condition are considered for the appropriate standard in the case of saturated refrigerant/lubricant mixture. In the case of modeling of the ejector's mixing section, the mixing pressure is given as an input corresponding to the saturation pressure for the 1K saturation temperature drop of the refrigerant from the suction chamber inlet. Besides, the motive flow rate ratio (r_{KH}) defined as the ratio of motive mass flow rate to the total mass flow rate is also an input parameter for applying the iterative simulation procedure. As this ejector model considers both refrigerant and lubricant, separate species balance of both refrigerant and oil are taken into consideration in the analysis of ejector components.

Inputs: Refrigerant states at motive and suction nozzle inlets, $r_{KH}, P_{ms}, \eta_{mn}, \eta_{diff}$

⇩

Motive nozzle: $h_{mn,out,is} = f\left(P_{ms}, s_{mn,in}\right)$ [equation of state]

$h_{mn,out} = h_{mn,in} + \eta_{mn}\left(h_{mn,out,is} - h_{mn,in}\right)$ [efficiency definition]

$v_{mn,out} = \sqrt{2\left(h_{mn,in} - h_{mn,out}\right)}$ [energy conservation equation]

⇩

Suction chamber: $c_{p,oil} = f(T_{ms})$ [equation of state]

$h_{sn,out} = c_{p,oil}\left(T_{ms} - T_{reference}\right) + h_{reference}$

$v_{sn,out} = \sqrt{2\left(h_{sn,in} - h_{sn,out}\right)}$ [energy conservation equation]

⇩

Mixing section: $r_{KH} = \frac{\dot{m}_{mn}}{\dot{m}_{mn}+\dot{m}_{sn}} = \frac{1}{1+ER}$ [input parameter]

$v_{diff,in} = r_{KH}v_{mn,out} + (1 - r_{KH})v_{sn,out}$ [linear momentum conservation equation]

$h_{diff,in} = r_{KH}h_{mn,in} + (1 - r_{KH})h_{sn,in} - 0.5v_{diff,in}^2$ [energy conservation equation]

$s_{diff,in,ref} = f\left(P_{ms}, h_{diff,in,ref}\right)$ [equation of state]

⇩

Diffuser: $h_{diff,out,is,ref} = h_{diff,in,ref} + 0.5\eta_{diff}v_{diff,in}^2$ [efficiency definition]

$h_{diff,out,ref} = h_{diff,in,ref} + 0.5v_{diff,in}^2$ [energy conservation equation]

$P_{diff,out,ref} = f\left(s_{diff,in,ref}, h_{diff,out,is,ref}\right)$ [equation of state]

⇩

Outputs: P_{lift}, r

Figure 3: Ejector modeling procedure for the ejector cycle of oil pumping

2.3 Ejector Performance Metrics

The performance improvement of the ejector refrigeration cycle depends largely on the performance of the ejector itself. The mass entrainment ratio (*ER*) and pressure lift (P_{lift}) characterize mostly the ejector performance. The entrainment ratio (*ER*) is defined as the ratio of the suction fluid mass flow rate to the motive fluid mass flow rate, and this parameter governs the entrainment capability of an ejector. Besides, the pressure lift which is the difference between diffuser outlet pressure and suction chamber inlet pressure also signifies the ejector performance. For a good ejector operation, it is desirable to have a higher entrainment ratio as well as a reasonable amount of pressure lift. In addition to the pressure lift and entrainment ratio, another important performance indicator of an ejector is the ejector efficiency (η_{ej}). Contemporary research defined ejector efficiency in different ways. However, in this considered ejector model, the ejector efficiency definition is applied according to the definition proposed by Elbel and Hrnjak (2008). According to this definition, ejector efficiency is termed as the ratio of work recovered by the ejector to the maximum amount of recoverable work through an ejector. Elbel and Hrnjak (2008) considered isentropic compression of the suction fluid stream from the inlet of the suction nozzle to the diffuser outlet. However, in the present study, the suction fluid stream is the compressor oil that has been considered as the incompressible fluid for simplifications of obtaining properties. For this reason, the ejector efficiency definition is modified in terms of the calculation of the work recovered by the ejector.

$$\eta_{ej} = \frac{\dot{m}_{sn}}{\dot{m}_{mn}} \left[\frac{h(P_{diff,out}, T_{sn,in}) - h_{sn,in}}{h_{mn,in} - h(P_{diff,out}, s_{mn,in})} \right] \tag{6}$$

3. EJECTOR DESIGN GUIDELINES

3.1 Numerical Modeling Results for the Ejector Cycle of Oil Pumping

The ejector cycle illustrated in Figure 2 is simulated for the steady-state operation of a transcritical R744 system. Numerical modeling results reveal the insights regarding the bypass flow or motive flow, pressure lift, ejector efficiency, etc. for optimization of the ejector design. Figure 4 illustrates the variation of bypass ratio with the entrainment ratio for the considered ejector cycle and other stated assumptions necessary for modeling the ejector cycle. Another important assumption regarding the present study is that the same amount of oil is pumped out of the evaporator as the oil deposition rate at the evaporator. As the suction oil mass flow rate is constant for a particular condition, the bypass flow changes with the change in entrainment ratio, and from Figure 4, it is evident that the bypass ratio (r) decreases with the increase in entrainment ratio (ER). For the considered fixed parameters, the required bypass flow is a maximum of around 0.9% of the total flow at the compressor outlet in the case of a low entrainment ratio. As the entrainment ratio increases, the oil entrainment capability of the ejector increases. Eventually, less driving flow is required to entrain the same amount of oil leading to the exponential decrease in bypass ratio. The relation between the ejector efficiency (η_{ej}) and entrainment ratio is plotted in Figure 5 whereas, it is obvious that the ejector efficiency increases with the increase of entrainment ratio. A maximum of around 30% ejector efficiency can be achieved in the case of the oil ejector operation for high entrainment ratio whereas, lower ejector efficiency is observed in the case of less oil entrainment through the ejector. However, the maximum range of the ejector efficiency matches with the contemporary literature (Elbel and Hrnjak, 2008), (Lucas and Koehler, 2012). Theoretically, a higher ejector efficiency can be achieved in the case of a higher entrainment ratio because in this case the entrainment capability of the ejector also increases thus requiring less driving flow for the ejector operation.

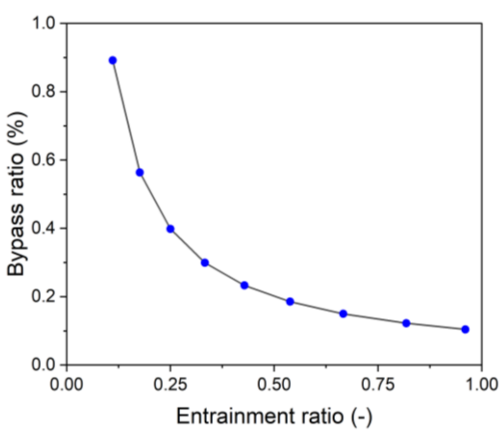

Figure 4: Variation of bypass ratio with entrainment ratio

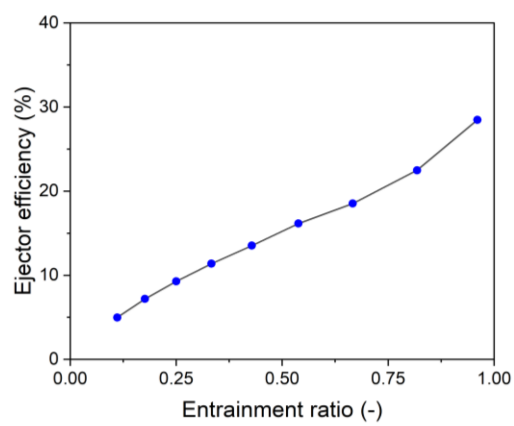

Figure 5: Variation of ejector efficiency with entrainment ratio

3.2 Design Procedure of the Motive Nozzle

For design optimization of the ejector dedicated to oil pumping, the first step is to design the motive nozzle. Regarding the design of the motive nozzle for the oil ejector, the nozzle throat diameter (D_t) and the nozzle exit position diameter (D_{NXP}) are key parameters. The motive nozzle throat diameter can directly influence the high-side system pressure whereas, the motive nozzle exit position diameter allows further motive flow expansion through the nozzle. The motive fluid mass flow rate is estimated from the iteratively solved ejector model for oil pumping, and a reasonable value for the isentropic flow coefficient through the motive nozzle is assumed based upon experimental studies. Regarding the design of the nozzle throat as well as the nozzle exit diameter, the real gas model in the case of vapor phase ejector proposed by Yoshida *et al.* (2021) is modified and implemented in the considered ejector model. Figure. 6 represents the motive nozzle design procedure for the ejector dedicated to oil pumping.

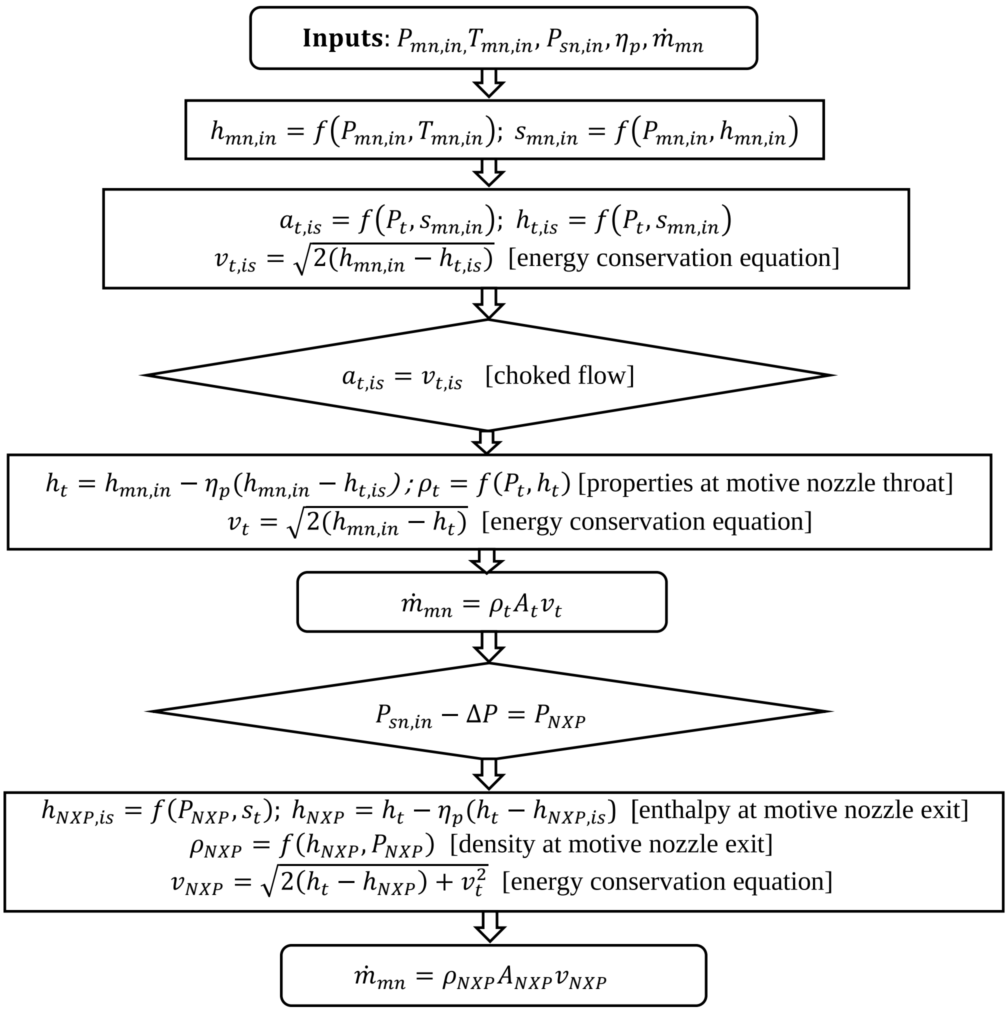

Figure 6: Design procedures of the motive nozzle for the ejector dedicated to oil pumping

The proposed design procedure of the motive nozzle assumes the choked flow at the motive nozzle throat position. As the high-pressure compressed vapor expands to a high velocity after expansion, it is a reasonable assumption to achieve choked flow at the motive nozzle throat. The motive nozzle throat pressure is iteratively solved through the developed design procedure. As the motive flow is assumed to be choked at the motive nozzle throat, the motive flow rate is considered to be constant through the entire motive nozzle. So, the motive nozzle throat diameter is calculated through the mass flow rate equation. To estimate the motive nozzle exit pressure, a fixed amount of pressure drop is assumed from the suction nozzle inlet similarly likewise the mixing pressure estimation. Finally, the motive nozzle exit diameter is calculated using the same mass flow rate through the motive nozzle.

3.3 Test Experimental Layout Design for Oil Entrainment Experiment

Numerical modeling of the considered ejector cycle reveals insights about the required bypass flow as well as the suction oil mass flow rate through the ejector in the case of the assumed test conditions. Besides, the ejector performance can be predicted in terms of ejector efficiency and pressure lift characteristics through this numerical model. However, for optimizing the suction chamber as well as the mixing section design, these numerical model results are not sufficient enough and experimental investigation is necessary. An oil separator is present in this ejector cycle, and the oil separator can separate the major portion of the oil from flowing through the condenser/gas cooler, evaporator, and other system components leading to the low rate of oil accumulation at the evaporator for steady conditions. Due to the low oil deposition rate at the evaporator, the suction oil mass flow rate through the suction chamber of the ejector becomes also lower. In this regard, an experimental study can better quantify the achievable oil suction mass flow rate through the ejector as well as the ejector performance. Contemporary literature reported no experimental study regarding this oil entrainment through the ejector, and the design guidelines of the oil ejector could be very different from that of the refrigerant ejectors due to the relatively high viscosity of compressor lubricant. For design optimization of the oil ejector, especially, the suction chamber, as well as the mixing section design, a prototype oil entrainment test experiment layout is designed to conduct oil entrainment experiments. Figure 7 illustrates the designed novel test experiment layout for oil entrainment experiments.

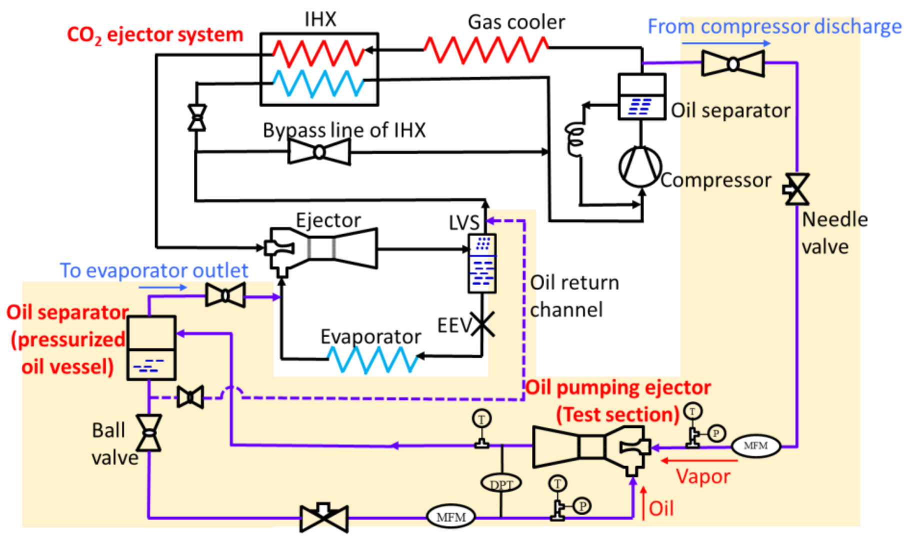

Figure 7: Test experimental layout design for oil entrainment experiments

A standard CO_2 experimental test facility was previously constructed for conducting the standard ejector cycle test experiments. In the standard CO_2 experimental test facility, an internal heat exchanger (IHX) is used downstream of the gas cooler. A separate bypass line is also installed to compare the system performance between two cases: with IHX and without IHX. This paper outlines a designed novel oil entrainment test facility (shaded color) that is integrated into the standard CO_2 ejector system with suitable piping and instrumentations.

In the oil entrainment test setup, a certain portion of the flow from the compressor outlet is bypassed through the bypass line and fed as the motive flow to the ejector. The motive flow rate of the ejector is controlled by installing necessary valves at the bypass line from the compressor exit. An oil separator is used as a pressurized oil vessel from which oil can be entrained through the suction chamber of the ejector. Pressure, temperature, and mass flow rate are monitored with the help of appropriate sensors for both motive and suction fluid; whereas, a differential pressure transducer measures the pressure lift from the ejector. From the ejector outlet, the vapor-oil mixture enters the oil separator where the oil gets separated allowing further entrainment through the ejector's suction chamber. The vapor is discharged from the oil separator outlet and joins the evaporator outlet flow before entraining through the suction chamber of the standard CO_2 ejector system. In this case, the evaporator outlet flow can be controlled by adjusting the evaporator load. In addition, a separate oil return channel is also incorporated from the bypass line of the oil separator's oil port to the compressor suction thus ensuring sufficient oil return to the compressor.

3.4 Ways for the Ejector Design Optimization

Oil entrainment test experiments will be conducted with the help of the designed oil entrainment experimental test facility to find out suitable correlations between the motive and suction flow rate of the ejector. Initially, the standard CO_2 refrigerant ejector can be used to conduct test experiments for gaining insights about the oil entrainment capability through the standard ejector. In the case of the diffuser design, the diffuser angle is an important parameter that should be chosen carefully. The optimum diffuser angle of 5° is reported to obtain good ejector performance in the contemporary literature (Elbel and Hrnjak, 2008), (Banasiak *et al.*, 2012). According to their study, it is also reported that the diffuser length as well as angle has relatively less impact on the ejector performance compared to the motive nozzle and mixing section geometries. In the first test sequence, the amount of oil entrainment can be investigated for a fixed motive flow rate and ejector geometry. Then the motive flow rate of the ejector needs to be varied to assess the oil entrainment capability for a wide range of operating conditions. In the final step, ejectors of varying geometries can be used to gain insights about the optimum ejector performance in different operating conditions for the ejector cycle dedicated to oil pumping. Thus, the design of an oil ejector can be optimized based upon the experimental as well as numerical modeling results.

4. CONCLUSIONS

This paper presents an approach to gain insights about the design guidelines as well as the design optimization for the R744 ejector in the case of pumping of compressor lubricant and returning it to the compressor intermittently. A suitable ejector cycle of oil pumping patented by Kim *et al.* (2016) is considered for a numerical investigation of the ejector cycle for predicting the required bypass flow as well as ejector performance characteristics. The numerical model considers a steady-state operation of a transcritical R744 ejector system with a fixed oil circulation rate (OCR) of 2% (by mass) at the compressor exit. Preliminary numerical modeling results reveal that a maximum of around 0.9% bypass ratio is required to perform the oil ejector operation; whereas, the maximum ejector efficiency can reach up to 30%. An appropriate design framework is proposed for designing the motive nozzle of the oil ejector. In addition, a novel oil entrainment test setup is designed for assessing the oil entrainment capability of the ejector for a wide range of operating conditions. The ejector design can be optimized based upon the oil entrainment experimental results. Future studies could focus on the experimentation with the built test facility to find out the achievable oil suction mass flow rate for different conditions. Furthermore, these studies could be extended for other refrigerants to assess the potential of compressor lubricant pumping and returning it to the compressor.

NOMENCLATURE

A	cross-sectional area	(m^2)
a	sound velocity	(m.s^{-1})
COP	Coefficient of Performance	(-)
c_p	specific heat capacity	(kJ.kg^{-1}.K^{-1})
D	diameter	(m)
ER	entrainment ratio	(-)
h	specific enthalpy	(kJ.kg^{-1})
$\dot{m}$	mass flow rate	(kg. s^{-1})
OCR	oil circulation rate	(-)
p	pressure	(kPa)
P_{lift}	pressure lift	(kPa)
ΔP	pressure drop	(kPa)
r	bypass ratio	(-)
r_{KH}	motive flow rate ratio	(-)
S	specific gravity	(-)
s	specific entropy	(kJ.kg^{-1}.K^{-1})
T	temperature	(°C)
v	velocity	(m.s^{-1})
η	efficiency, flow coefficient	(-)
ρ	density	(kg.m^{-3})

Subscript

diff	diffuser
ej	ejector
in	inlet
is	isentropic
mn	motive nozzle
ms	mixing section
NXP	motive nozzle exit position
oil	compressor lubricant
out	outlet
os	oil separator
p	primary flow or motive flow
ref	refrigerant
reference	reference properties
sn	suction chamber
total	refrigerant + compressor lubricant

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