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EXPERIMENTAL INVESTIGATIONS OF FULLY PASSIVE HEAT DRIVEN TWO-PHASE INJECTOR

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ABSTRACT

Energy required to drive a mechanical pump in heat driven systems in most of the cases is a few percent of the overall energy balance of the system and in most cases is of the order of magnitude 1%. The most important problem is a special difficulty to select of a commercially available liquid pump for the heat driven cycle. Mechanical liquid pump liquid is the only element of the refrigeration system that consumes electric power. It is desirable that the whole system would be fully heat driven one, without consumption of electric power to drive. Two-phase vapour-liquid injectors may be thought as required alternative. They are using vapour that is generated already in the system; operates as pre-heater of liquid phase supplied to the vapour generator that additionally improve the system efficiency; and have no moving parts. Paper deals with experimental investigation of operation of the fully passive system with two-phase vapour-liquid injector as a liquid pump in refrigeration systems. Presented results are the first ones for the system operating with low-GWP refrigerant under conditions of fully passive operation.

Keywords: injector, compression efficiency, heat driven system, ejector refrigeration system, refrigerants, two-phase ejector

1. INTRODUCTION

Renewable energy sources find more and more application in the power industry, they can also be used to drive refrigeration systems. In this case, sorption and jet ejector systems can be used in refrigeration engineering. In all of these systems the liquid pump should be applied which is one of the crucial limitations of the investment costs as well as operation reliability and flexibility of these systems. A special attention is paid to the application of the liquid pump in the ejector refrigeration systems due to great variety of the working fluids that may be selected for these systems and unfavourable their properties in terms of the liquid pump requirements.

Figure 1a shows the ejector refrigeration system in which the mechanical pump is applied. This pump is the only electrically driven element. Therefore, it is desirable to have the entire system fully driven by heat, without the use of electricity. The ejector refrigeration cycle with a two-phase injector used as a liquid pump is shown in Fig. 1b. The theoretical analysis showed that the two-phase injector operates also as a pre-heater of the liquid phase supplied to the vapour generator which improves the efficiency of the system, Śmierciew et al. (2015). Moreover, this type of liquid pressurization device is simpler and more reliable than a mechanical pump as it has no moving parts. The operation conditions under which the injector is driven by heat only, without any mechanical support, is named as passive operation.

The potential application of various working fluids for operation with was investigated by Chen et al. (2014), the reasonably good performance of isobutane was demonstrated which was compared to the synthetic fluids R141b and R245fa. Regulation of the European Parliament and the EU Council No. 517/2014, enacted on April 16th, 2014 forbids the use of commercial and domestic air conditioning units with more than 3 kg of refrigerant with GWP higher than 150 in EU countries. Therefore, it is necessary to take into consideration new alternative fluids that may be applied in the heat driven systems, especially for the ejector refrigeration systems. The fluids selection was analysed by Smierciew et al. (2017) and Smierciew et al. (2019). These analysis demonstrated a good feasibilities for application of the injector as a liquid pumping device for she ejector systems operating with HFO refrigerants. However, no experimental data were available that demonstrate the operation of the injector under passive conditions for these fluids which is motivation for the present paper.

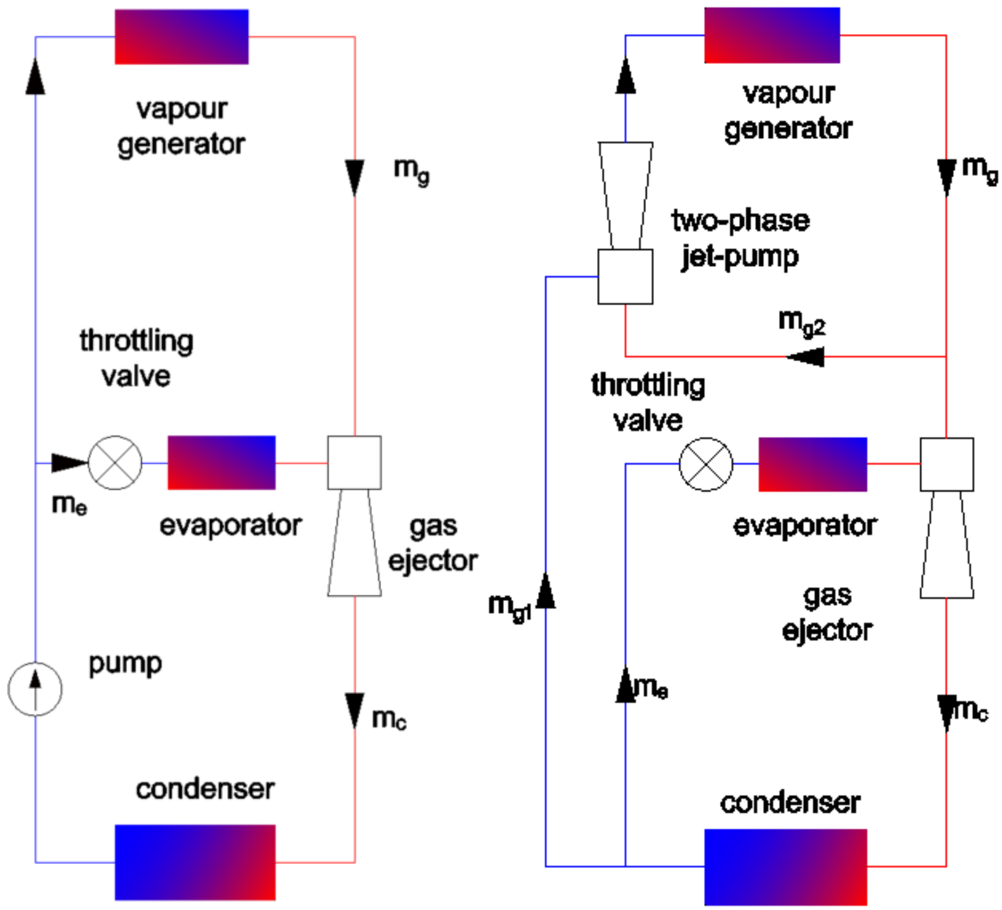

Fig. 1. Refrigeration ejection system:
a) with a mechanical pump, b) with two-phase injector

The theoretical analysis of the operation of a two-phase vapour-liquid injector for use in ejector refrigeration systems was developed by Przybylinski et al. (2013). This model was validated with own experimental data for isobutane as a working fluid with good results. However, in this case the injector operated with artificially produced entrainment ratio and compression produced by the injector was analysed.

2. EXPERIMENTAL STAND

The experimental research was carried out on a proprietary stand built at the Bialystok University of Technology for testing of the two-phase vapour-liquid injectors. Figure 2 presents the cross-section of the tested injector. The tested ejector enables change of the location of the motive nozzle of the injector which is one of the most important parameter of the geometry of the injector.

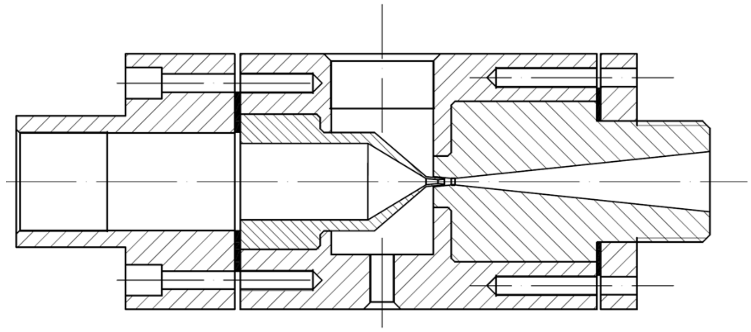

Fig. 2. Sectional view of the tested injector

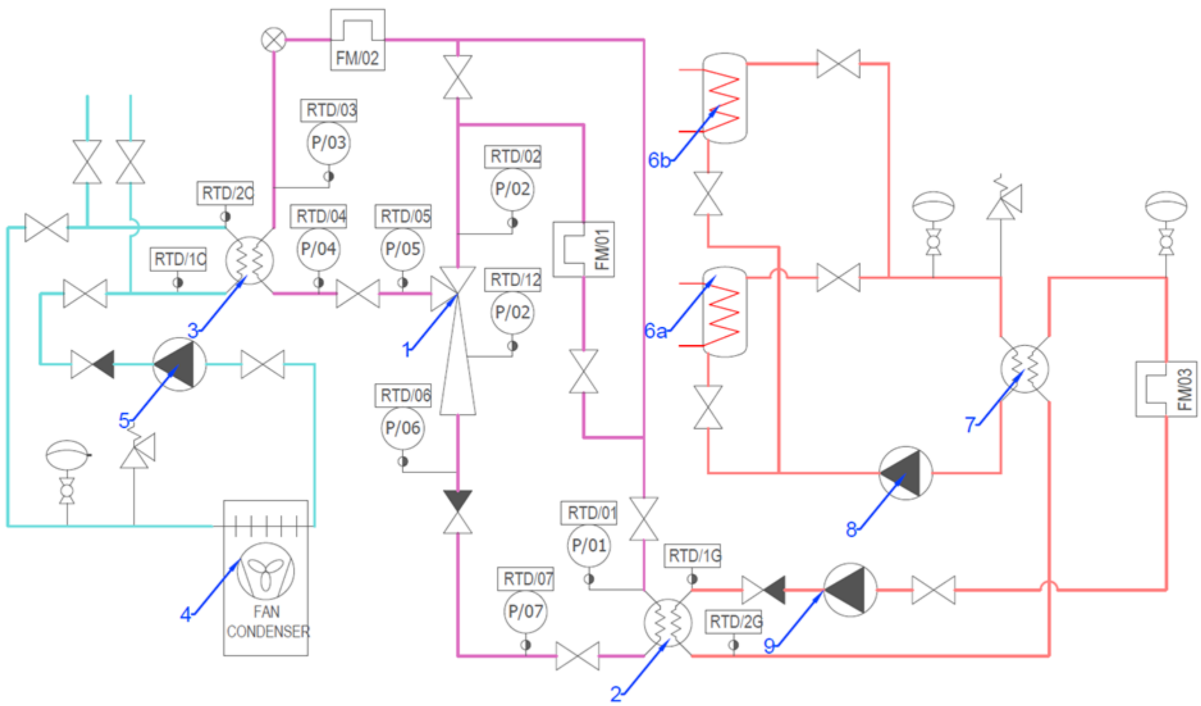

Fig. 3. Experimental stand for testing of the two-phase injectors: 1 – tested injector; 2 – vapour generator; 3 – condenser; 4 - fan cooler; 5.8,9 - circulation pump; 6a, 6b - electric heater; 7 - accumulation tank with a coil; RTD - temperature sensor, P - pressure sensor, FM – flow meter

Tab. 1. List of the elements of the experimental system

Nr	Name of the element	Description
1	Two-phase injector pump	own design
2	Vapour generator	plate heat exchanger
3	Condenser	plate heat exchanger
4	Fan cooler	own design
5	Circulation pump	L.FP PML2 40/130
6a,b	Electric heater	KOSPEL ECOC.L1
7	Heat accumulator	Galmet SGW200
8	Circulation pump	Pump integrated with an electric heater
9	Circulation pump	GRUNDFOS CRNE15-02
FM/01	Mass flow meter	PROMASS 80
FM/02	Mass flow meter	PROMASS 80
FM/03	Mass flow meter	PROMASS 40
P	Pressure sensor	WIKA S20
RTD	Resistance temperature sensor	CZAKI

Figure 3 shows the schematic of the test stand. The condenser is cooled by water through the heat exchanger or by means of a dedicated fan cooler. The refrigeration pressure control valve is responsible for control of the subcooling of liquid refrigerant at the inlet to the injector. The vapour generator heating loop was implemented with the help of two EKCO.L1-24z boilers from Kospel, which heat glycol to the required temperature in the heat accumulator. Water from the heat accumulator to the vapour generator is pumped using a Grundfoss CRNE pump.

Fig. 3. Photo of the experimental stand

The quantities measured during the measurements on the experimental stand are: temperature, pressure and mass flow of the refrigerant. Temperature measurement was carried out using the CZAKI PT100 sensors, the static pressure measurement was carried out by WIKA pressure transducers of the type of S20. The mass flow rate of both phases was measured with use of Endress Hauser Promass mass flow meters. The sensor data will be collected by the National Instruments CompactRIO system and sent to a computer equipped with LabView software (National Instruments) for recording and processing.

3. VALIDATION OF THE TWO-PHASE INJECTOR MODEL

The criterion numbers adopted during the validation of the model are the pressure ratio π_d , eq. (1), the mass entrainment ratio U which is ratio of mass flow rate of liquid to motive vapour, eq. (2), these quantities are defined as follows:

$$\pi_d = \frac{p_d}{p_{sL}}, \tag{1}$$

$$U = \frac{\dot{m}_L}{\dot{m}_V}. \tag{2}$$

The assessment of the efficiency of the injector operation may be thought as an open issue. The investigated two-phase injector is also an effective heat exchanger so that liquid temperature increase that occurs in the injector may be also thought as an useful effect of the injector operation. Therefore the total efficiency of the injector may be defined as a ratio of sum of power consumption for isochoric compression of liquid phase and heat delivered to the liquid phase to motive power consumption defined as thermal capacity of the vapour generator:

$$\eta_e = \frac{U}{h_V - h_{dL}} \left[\frac{p_d - p_{sL}}{\rho_{dL}} + h_{dL} - h_{sL} \right], \tag{3}$$

where h_{sL} is specific enthalpy of liquid phase at injector suction chamber; p_d – discharge pressure; p_{sL} – pressure of liquid phase at the injector suction chamber; h_V – specific enthalpy of the motive vapour; h_{dL} – specific enthalpy of liquid at the vapour generator inlet that is theoretically equal to the specific enthalpy of liquid at the discharge of the injector); ρ_{dL} – density of liquid delivered to the vapour generator.

The saturation temperature of the drive was $t_{g,\,sat}$ = 28.5 ° C ± 0.3 °C, at average superheat ΔT_g = 23 K ± 0.4 ° C. The average value of the drive pressure was p_g = 0.56 MPa. The condensation temperature was $t_{c,\,sat}$ = 22.5 ° C ± 0.6 ° C, which corresponds to p_c = 0.45 MPa. The main geometrical parameters of the tested injector are: d_{nt} = 1.0 mm, d_n = 1.3 mm, d_t = 1.8 mm, d_d = 33 mm. The location of the exit position of the motive nozzle can be changed in the range of 0–3.0 mm, where 0 mm means that the nozzle touches the surface of the mixing chamber. In the case of the performed measurements, the thickness of the liquid gap was δ = 0.28 mm.

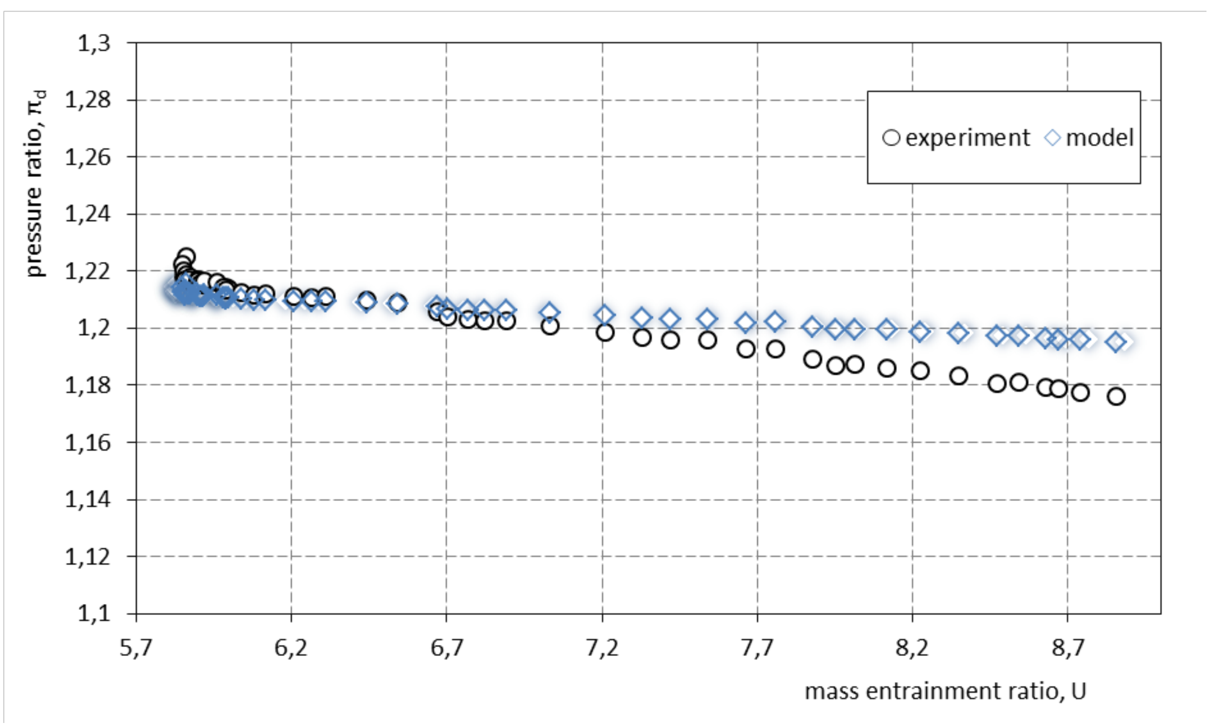

Fig. 4. Relation of pressure ratio π_d in function of mass entrainment ratio U

In Fig. 4 there was presented relationship between the pressure ratio π_d as a function of mass entrainment U. As it can be seen the entrainment ratios were obtained in the range from about 5.7 to 8.8, the pressure ratio oscillated in the range of 1.2. The experimental results were compared with theoretical predictions based on the model proposed by Przybylinski et al (2013) and Smierciew et al (2015, 2017). It was demonstrated therefore that the mathematical model is highly compatible with the experiment, the maximum calculation error for this case was 1.59%.

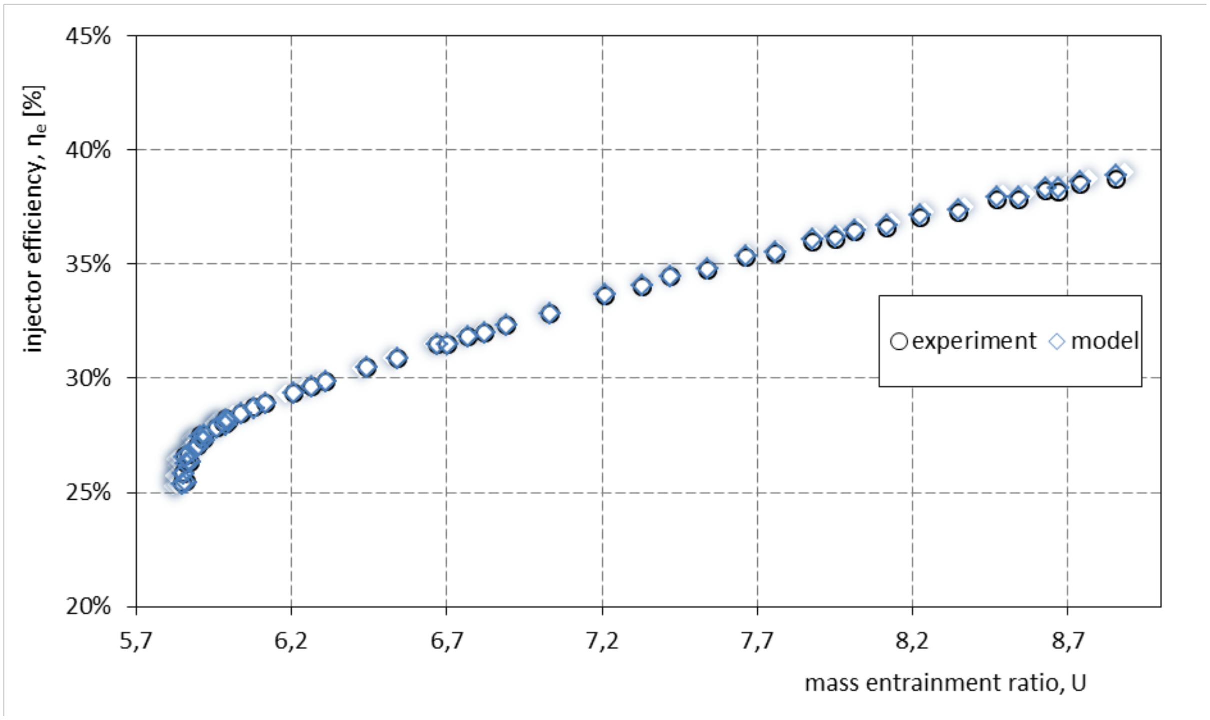

Fig. 5. Injector efficiency η_e vs mass entrainment ratio U

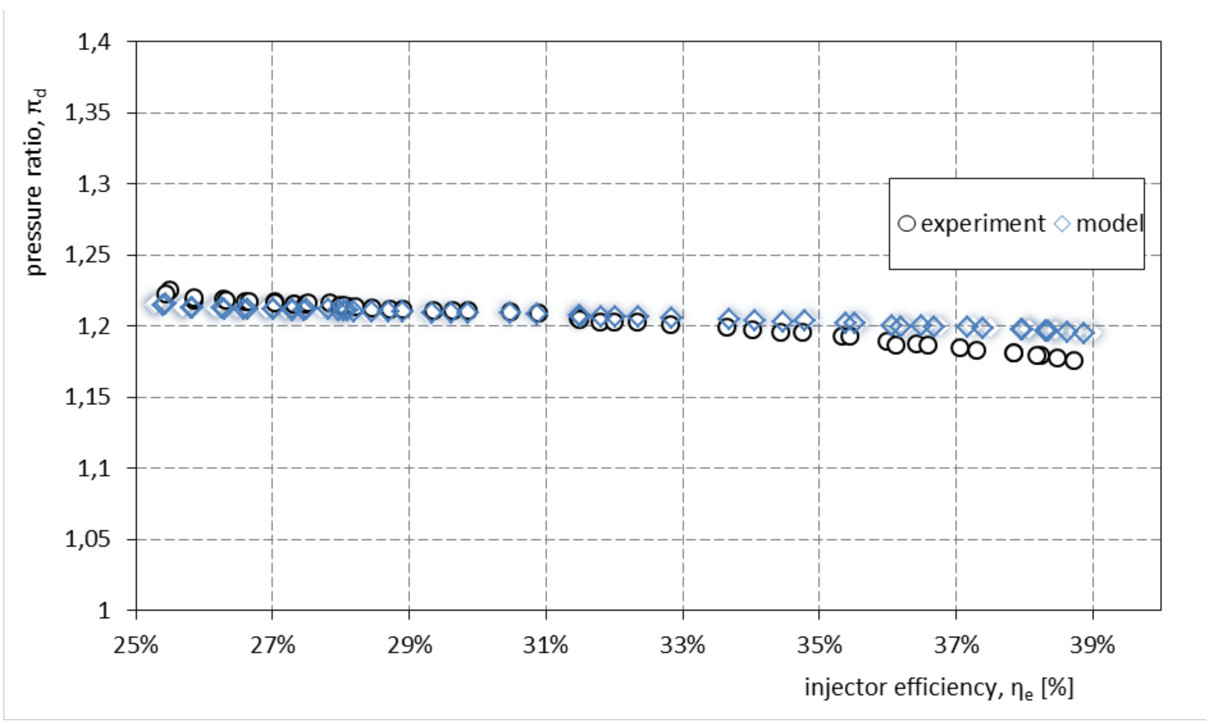

Fig. 6. Relation of pressure ratio π_d in function of injector efficiency η_e

Figure 5 shows the relationship between the efficiency of the ejector and entrainment ratio. The highest efficiencies were obtained for the highest entrainment ratios. The presented comparative analysis of the model with the experiment shows high compliance. The maximum error of calculation is 0.37%. In the tested scope of work, the injector achieved efficiency at the level of 25% to 40%.

The efficiency of the ejector was independent of the compression ratio. As it was shown in Fig. 6, the injector achieved a wide spectrum of efficiency with relatively constant compression ratio. This may result from the variable amount of the medium inside the injector, as shown in Fig. 5. The injector is a device that adapts itself to the operating conditions.

The series of experimental data on the basis of which the model was validated was characterized by the variability of the suction ratio, other parameters such as temperature and pressure of steam and liquid remained constant.

4. CONCLUSION

The paper provides with the experimental operation of the injector dedicated for thermally driven cycles. The results were obtained for refrigerant R1234ze which is dedicated for low temperature driven cycles. It was demonstrated operation of the injector during passive operation, i.e. under natural operating conditions. The presented results were applied for validation of the previously provided theoretical model with reasonable good agreement between experimental data and theoretical prediction. The total efficiency of the injector is assessed as sufficient for application of the thermal driven cycles, including the ejection cycles, as a liquid pumping device.

ACKNOWLEDGMENTS

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NOMENCLATURE

d	diameter, (m)	$\dot{m}$	mass flow rate, (kg/s)
h	specific enthalpy, (J/kg)	ρ	density, (kg/m^3)
U	entrainment ratio, (-)	η_e	injector efficiency
p	pressure, (Pa)	π_d	pressure ratio, (-)
T	temperature, (°C)	δ	liquid nozzle throat thickness, (m)

SUBSCRIPTS

d	diffuser	nt	vapour nozzle throat
e	injector,	V	vapour,
L	liquid	s	isentropic flow; suction chamber
n	nozzle	t	throat of the mixing chamber

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