2016

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ABSTRACT

Renewable energy sources may be applied to drive refrigeration and air-conditioning systems, e.g. solar radiation, geothermal resources, heat derived from biomass or waste heat rejected form various thermal processes. In this case absorption refrigeration systems and ejection systems may be used for cooling applications. In both these systems the crucial problem is the development of a suitable liquid pump. Paper deals with experimental investigation of two-phase vapour-liquid injector as a liquid pump in refrigeration systems. The selected experiment results for the injector are presented for the case of isobutane as working fluid. Investigations covers the operation characteristics of the injector as well as heat transfer coefficient.

1. INTRODUCTION

Renewable energy sources may be applied to drive refrigeration and air-conditioning systems, e.g. solar radiation, geothermal resources, heat derived from biomass or waste heat rejected form various thermal processes. In this case absorption refrigeration systems and ejection systems may be used for cooling applications. In both these systems the crucial problem is the development of a suitable liquid pump.

It is worth to note that energy consumption to drive these systems is not a major problem here as in most cases amount of energy required to drive a mechanical pump is a contribution of at most a few percent of the overall energy balance of the system and in most cases is of the order of magnitude 1%. Therefore the most important problem is a special difficulty to select of a commercially available liquid pump for the thermal driven cycle (absorption or ejection one). Such pumps should be low cost, small size and should produce sufficiently high compression rate for the discussed applications. They should also be adapted to work in particularly hard operating conditions resulting from the application in the discussed systems low boiling fluids: these are high penetrating substances, most of them are chemically aggressive and high susceptible to erosion caused by cavitation destruction. The above reasons causes that there is lack of commercially available mechanical liquid pumps of small and medium capacity which could be applied in refrigeration systems at reasonable cost and without other particular problems concerning the operation. Under present conditions most of the pumps that could be applied have inadequate overall dimensions as well as are very expensive so the contribution of the cost of the relevant pump is dominant in the total cost of the system. As an effect refrigeration systems driven by renewable thermal sources are commercially unprofitable and unattractive.

Mechanical liquid pump is the only element of the refrigeration system that consume electric power. Therefore it is also desirable that the whole system would be fully thermal driven one, without consumption of electric power to drive. The above is another important condition to search for the alternative solution for that would be competitive to mechanical classic liquid pumps. Two-phase vapour-liquid ejectors may be thought as required alternative. The general motivations of application of the two-phase injector instead of a mechanical pump in the discussed systems are:

- two-phase injector is driven by part of vapour that is generated already in the system; in this case the refrigeration system is fully thermal driven, without any consumption of electric power to drive the system;
- two-phase injector operates also as pre-heater of liquid phase supplied to the vapour generator that additionally improve the system efficiency;
- this type of injector is more simple and reliable than mechanical liquid pump since it has no moving parts as well as is not influenced to possible cavitation that may occur in mechanical pumps.

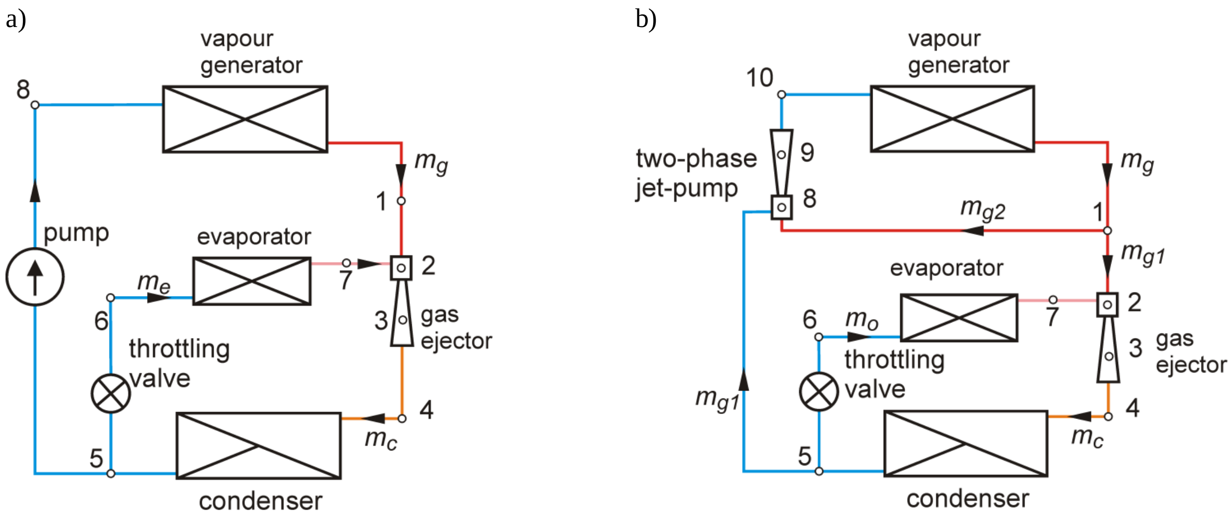

Figure 1. Diagrams of ejection refrigeration systems:
a) classic system with mechanical pump; b) modified system with two-phase injector.

The possible application of two-phase injector instead of mechanical pump for the ejection refrigeration system is presented in Fig. 1. A part of the motive vapour of working fluid is delivered to the motive nozzle of two-phase injector. However, in this case the system is fully thermally driven.

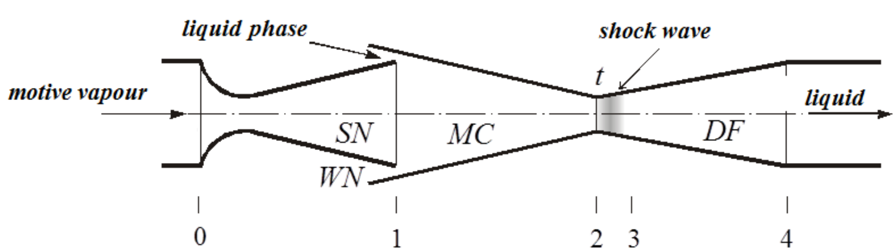

Figure 2. Schematic of vapour-liquid injector: SN – motive vapour nozzle,
WN – liquid nozzle, MC – mixing chamber, DF – diffuser, t – mixing chamber throat.

Two-phase injector (Fig. 2) is driven by vapour of high pressure and subcooled liquid phase is a secondary fluid. Motive vapour is expanded and accelerated to supersonic velocity in convergent –divergent motive nozzle. Subsequently it enters the mixing chamber (MC) at low static pressure sufficient for liquid to be drawn into the MC through an annulus surrounding the motive nozzle exit. In the mixing chamber due to difference in temperature and velocity, transfer of mass, momentum and energy during direct contact condensation of vapour occurs. Resulting two-phase flow is compressed in a condensation wave developing in the diffuser (DF) downstream of the throat of the mixing chamber. Inside the wave region the flow velocity decreases substantially as the vapour phase completely condenses. Thus only liquid phase of higher temperature is leaving the injector at a pressure that in some conditions could exceed the pressure of the motive vapour.

The possibility of application of two-phase injector as a liquid pump in refrigeration systems was previously theoretically studied by Shen *et al*. (2005), Zhang and Shen (2006), and Abdulateef *et al*. (2009). The physical basis of the operation of the two-phase injector as well as theoretical models were formulated. The modified ejection cycle operation was analysed for R718, R 717, and R134a for various operating conditions. Recently, the authors of this paper, Smierciew *et al*., (2015) also theoretically studied the modified ejection cycle for the case of isobutane. It was demonstrated that moderate decrease of the system efficiency may be expected in this case. Przybylinski *et al*. (2013) demonstrated on the basis of simple balance model that isobutane may be thought as preferable working fluid for modified system in terms of the efficient operation of the injector. However, there is no previous experimental studies of operation of the two-phase injector for refrigerants. The only previous experimental investigations covered steam-water injectors only, Trela and Butrymowicz (2004), Trela *et al*. (2005), Trela *et al*. (2010). It may be concluded on the basis of analysis of operations of such injector that condensation heat transfer play substantial role and drives momentum transfer in the injector. A clear need for the experimental data showing operation of such injector was motivation for the present experimental studies. On the basis of the previous studies the authors decided to carry out experimental investigations for the case of isobutane as working fluid.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The schematic of the experimental testing rig dedicated for the investigations of the two-phase injectors is presented in Fig. 3. The photo of the testing stand and the tested injector are presented in Fig. 4.

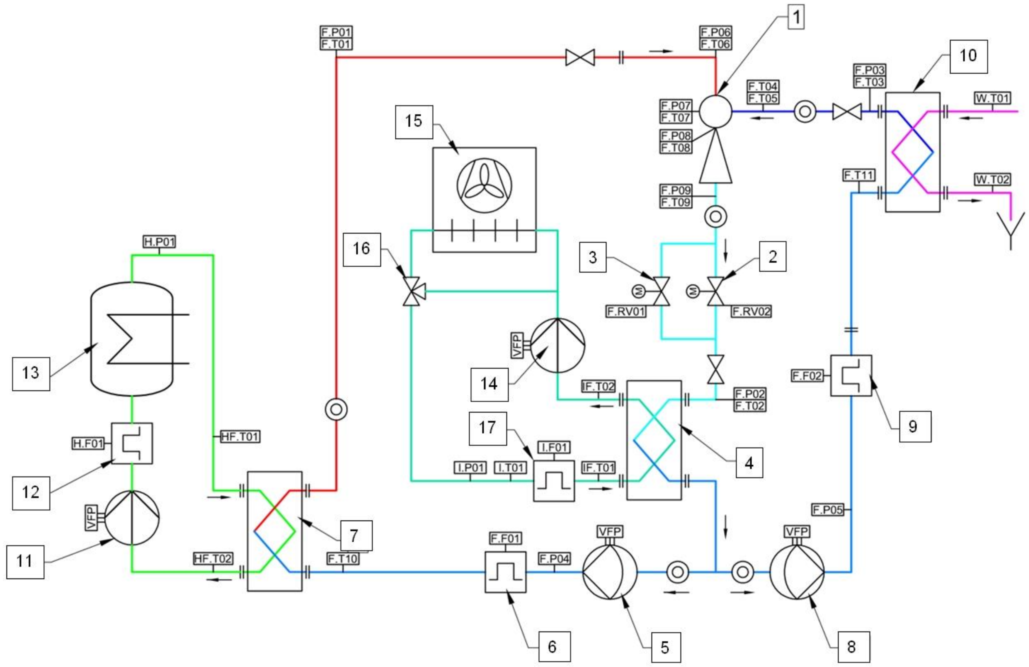

Figure 3. Schematic of the testing rig: 1 – tested injector; 2,3 – control valve; 4 – condenser; 5,8 – circulating pump of working fluid; 6,9 – mass flow meter; 7 – vapour generator; 10 – liquid subcooler; 11 – circulating pump of glycol; 12 – mass flow meter; 13 – electric heater; 14 – circulating pump of glycol, 15 – fan cooler; 16 – control valve; 17 – mass flow meter.

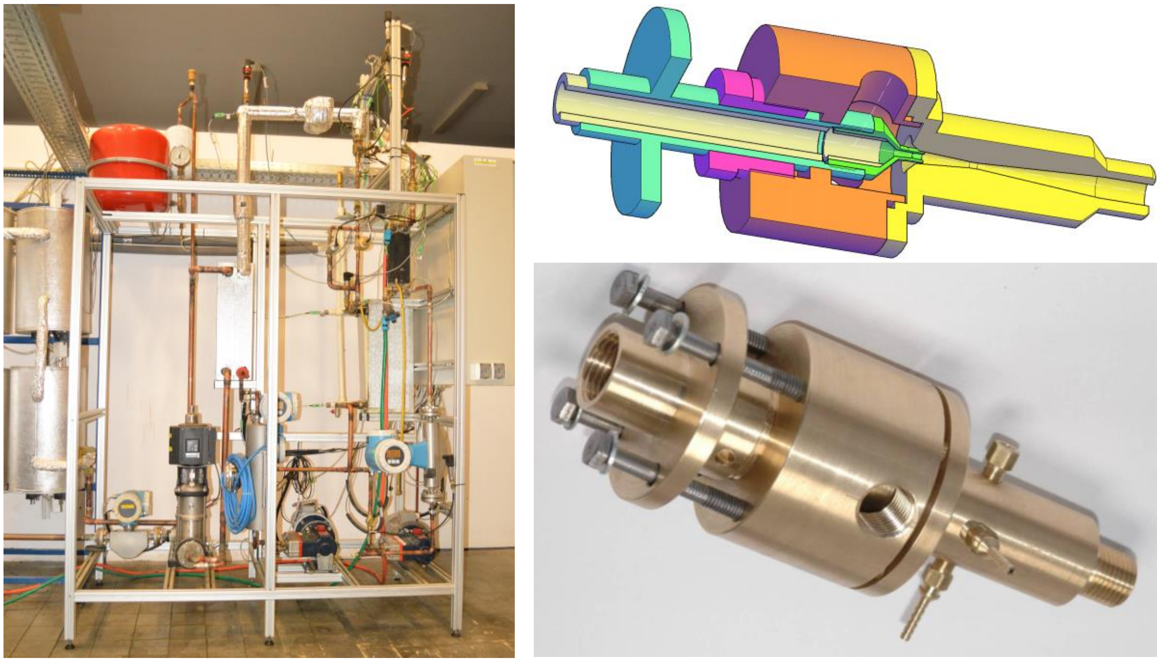

Figure 4. Photo of testing stand (left) and tested two-phase injector (right) equipped with the motive nozzle of the throat diameter 1.5 mm; the diameter of the mixing chamber 2.2 mm.

The whole experimental testing rig may be divided in four sub-systems, namely: system of circulation of working fluid; system of heating of vapour generator; system of condenser cooling, and system of cooling of subcooler. The motive vapour was delivered to the motive nozzle of the tested injector from the vapour generator. Motive pressure was thermally controlled then. Because the injector was tested for various operating conditions then secondary fluid parameters were controlled by means of the additional liquid pump 8 as it is seen in Fig. 3. This pump is gravitationally fed with liquid from the condenser 4. The liquid pump 5 feeds the vapour generator.

Positive displacement liquid pump Hydra-Cell WANNER of the series G 03 was applied in the testing system to feed the vapour generator with liquid. In order to protect the pump against excessive pressure the pressure transducer F.P04 was applied as a pressure controller. The capacity of this pump is controlled by means of frequency converter of the type ABB ACS 50-01E-04A3. The mass flow rate of the motive fluid is measured by mass flow mater 6. The motive nozzle of the tested injector is fed up with superheated vapour. The vapour superheating is controlled by the capacity of the heating system of the vapour generator.

The secondary fluid in the discussed case is subcooled liquid that is delivered by liquid pump 8 through the subcooler 10. Therefore both pressure and temperature of the secondary fluid was controlled in the system. In order to control the discharge pressure the control valves were applied between the diffuser outlet and the condenser. Both control valves were driven by stepping motors that enabled the precise control of discharge of the tested injector.

Four electric heaters 13 were applied in the heating system of the vapour generator. The total heating capacity of the electric heating system was 20 kW. The heating system enabled precise control of the operation of the vapour generator. The circulating liquid pump of variable capacity 11 was applied in the heating system. Propylene glycol solution ANTIFROGEN SOL (VP 1981) was applied as heat transfer fluid in heating system. Vapour superheating was controlled by reading of the pressure transducer F.P01 and thermocouple F.T01. The level of the vapour superheating was controlled by simultaneous control of the electric heaters capacity and glycol pump capacity.

The appropriate heat rejection from the condenser 4 was achieved by means of fan cooler 15. The cooling system was equipped with circulating pump 14 that is equipped with frequency converter so that precise cooling conditions may be achieved. Additionally, three-way control valve 16 was applied for the precise control of cooling medium flow rate. Glycol solution Ergolid A was applied as a cooling medium.

The design of the tested injector enabled the change of the motive nozzle as well as control of the distance of the outlet of the nozzle from the throat of the mixing chamber. This enables to investigate not only the effect produced by the change of the operating parameters but also to take into consideration the effect of the injector geometry. Four measurement gauges were applied in the tested injector to measure temperature and static pressure.

The investigations of the performance of the injector were carried out in the steady-state conditions. For the case of the reported investigations, the motive pressure as well as vapour superheating were kept constant . Also, concerning the secondary fluid the parameters were kept constant, i.e. liquid subcooling as well as liquid pressure were kept constant. The discharge pressure was varied by means of the control valves applied at the discharged line of the injector. The measurements covered the operating conditions for the case of the lowest possible discharge pressures that corresponds to totally opened control valves up to the maximum possible discharge pressure when the injector stops. Under conditions of the maximum possible discharge pressure the so-called stalling occurs which unable further operation of the injector.

3. EXPERIMENTAL RESULTS

Not only static pressure increase but also temperature increase occurs in the injector due to intensive condensation heat transfer. Therefore two concepts of the injector efficiency may be applied, i.e. the efficiency of compression as well as total efficiency of the injector that include also heat transfer. Compression efficiency η_c may be defined as a ratio of power consumption for isochoric compression of liquid phase (since injector operates as a liquid pump) to motive power consumption defined as thermal capacity of the vapour generator (since the injector is a thermally driven device):

$$\eta_c = U \frac{p_d - p_{sL}}{\rho_{dL}\left(h_V - h_{dL}\right)} \tag{1}$$

where: U –entrainment ratio, i.e. ratio of mass flow rate of liquid to motive vapour; 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.

However, 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:

$$\eta_e = \frac{U}{h_V - h_{dL}}\left[\frac{p_d - p_{sL}}{\rho_{dL}} + h_{dL} - h_{sL}\right] \tag{2}$$

where h_{sL} is specific enthalpy of liquid phase at injector suction chamber.

The additional parameters describing operation of the injector is compression ratio defined as ratio of discharge pressure to suction pressure:

$$\pi_e = \frac{p_d}{p_{sL}} \tag{3}$$

and temperature increase of liquid phase:

$$\Delta T_L = T_{dL} - T_{sL} \tag{4}$$

which could be presented in dimensionless form as Jakob number:

$$Ja = \frac{c_{pL} \Delta T_L}{h_{fg}} \tag{5}$$

where c_{pL} is specific heat of liquid chase at constant pressure, T_{dL} and T_{sL} are temperature of liquid phase at suction chamber and diffuser outlet, respectively; and h_{fg} is specific enthalpy of vaporisation.

Experimental investigations covered measurements for the motive nozzle of the throat diameter 1.50 mm with position of the nozzle changed so that three various thickness of the liquid gap were produced, namely: 0.13 mm, 0.21 mm, and 0.30 mm. The mass entrainment ratio was kept constant from U = 3.50 up to U = 5.50, motive pressure was kept also constant 10 bar. The results in terms of the total injector efficiency and characteristic of the injector operation for various liquid gap thickness as well as measured film heat transfer coefficient are presented in Fig. 5, Fig. 6, and Fig. 7, respectively.

The relationship between total injector efficiency η_e as a function of Jakob number Ja is presented in Fig. 5. As it is seen the total injector efficiency changes in the range of 9 to 30% for the thickness of the liquid nozzle 0.21 mm and motive vapour pressure 10 bar.

The effect produced by the change of the liquid nozzle gap thickness on the injector performance is presented in Fig. 6. It is seen that for two gap thickness: 0.21 mm and 0.30 mm the performance characteristics covered two zones. For the lowest compression ratio the non-complete condensation zone occurs, and after achieving a threshold compression level complete condensation zone appears. Very moderate temperature increase occurs in the last zone due to formation of the condensation shock wave. The shape of the performance characteristics of the injector operation for the gap thickness of 0.13 mm is different, however.

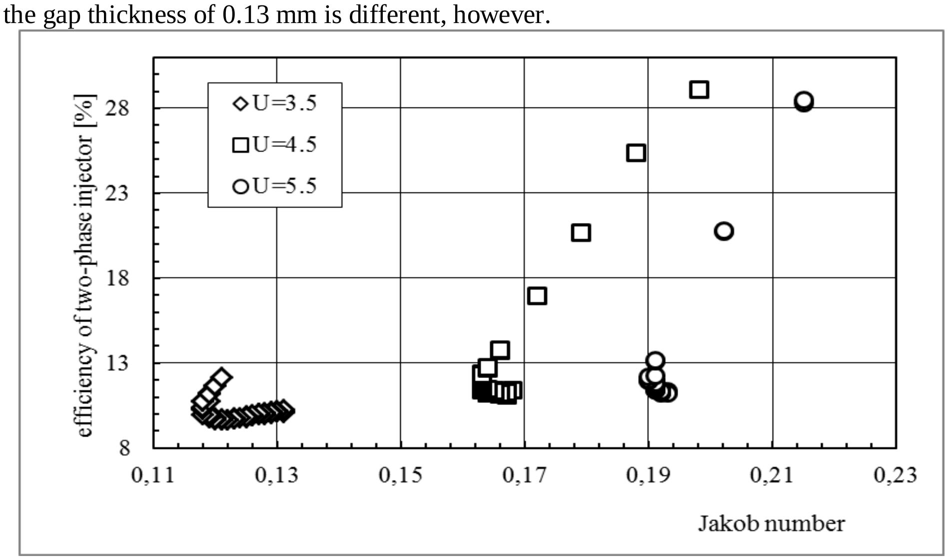

Figure 5. Relationship between total injector efficiency η_e versus Jakob number Ja.

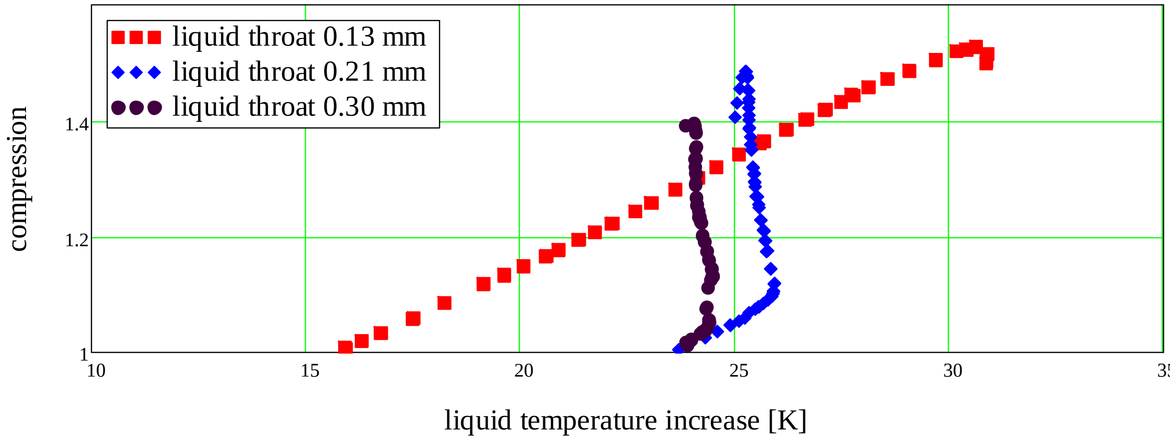

Figure 6. Performance characteristics of the injector operations for various liquid nozzle gaps thickness.

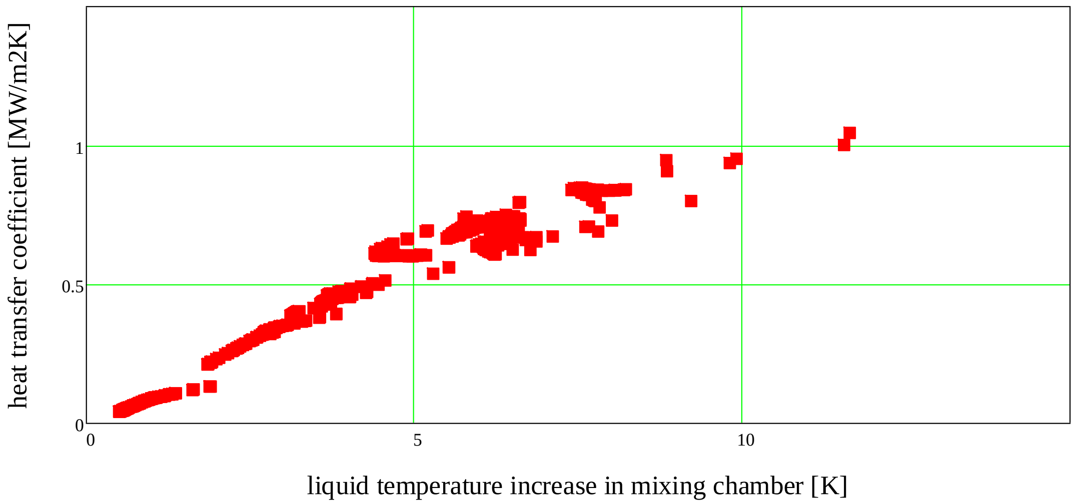

Figure 7. Condensation film heat transfer coefficient versus temperature increase of liquid phase.

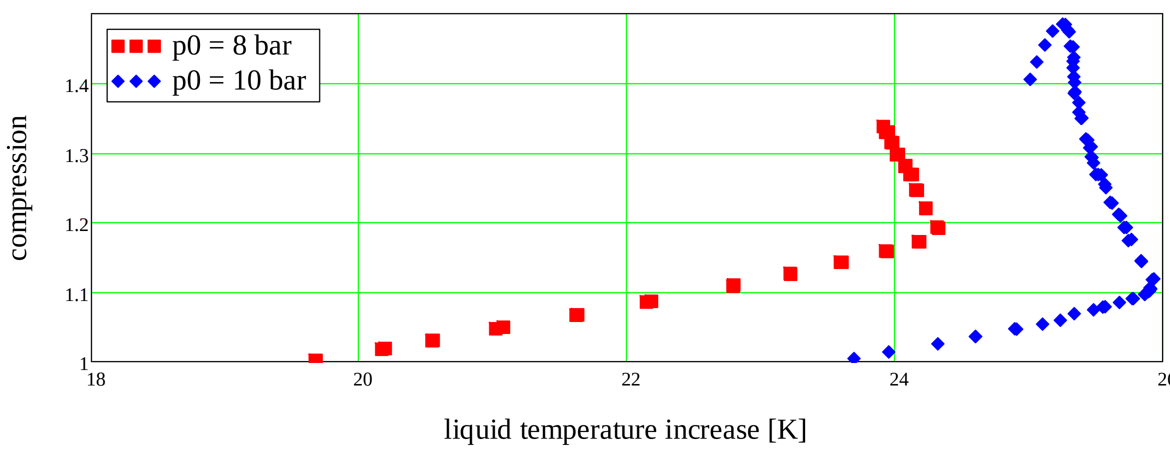

Figure 8. Compression ratio versus liquid phase temperature increase for various motive pressures.

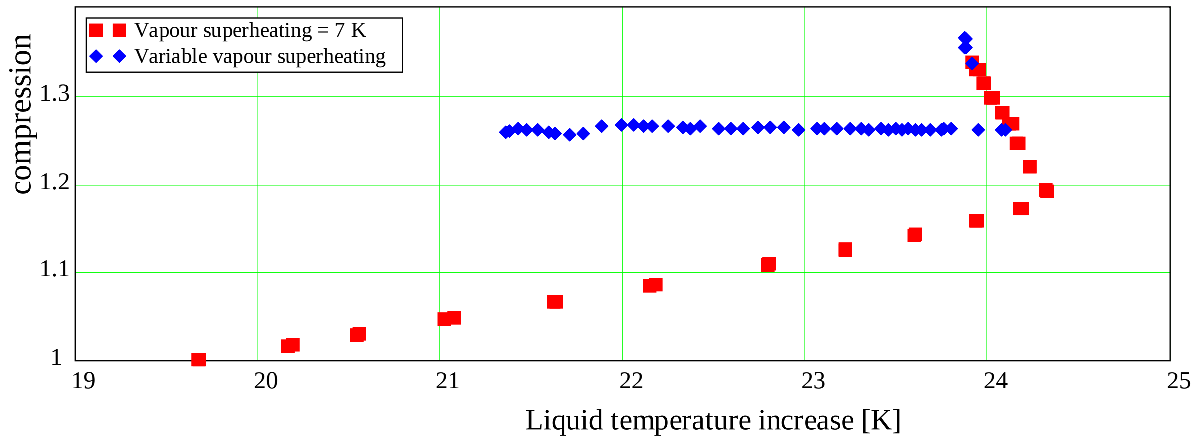

Figure 9. Compression ratio versus liquid phase temperature increase for various vapour superheating: $U = 4.5$, $p_V = 8$ bar, $\delta_l = 0.21$ mm

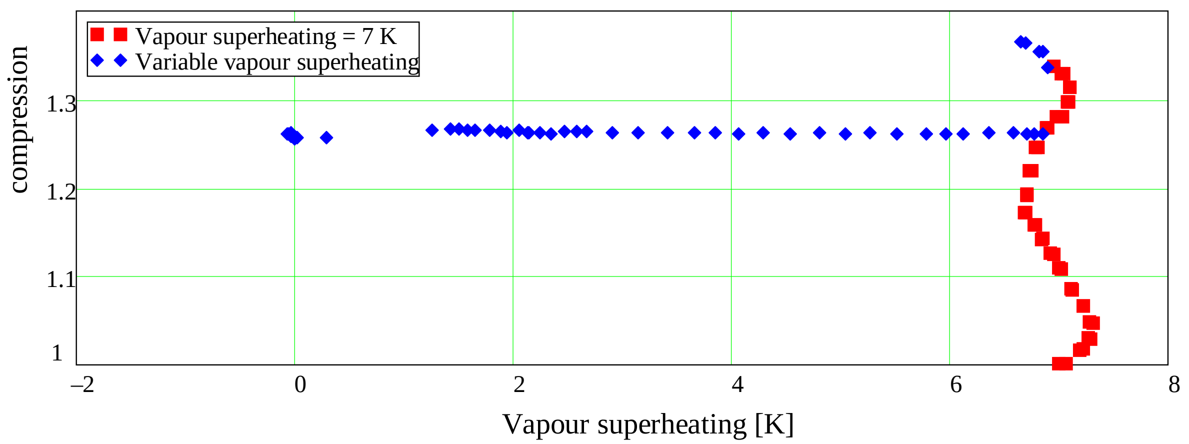

Figure 10. Compression ratio versus vapour superheating: $U = 4.5$, $p_V = 8$ bar, $\delta_l = 0.21$ mm

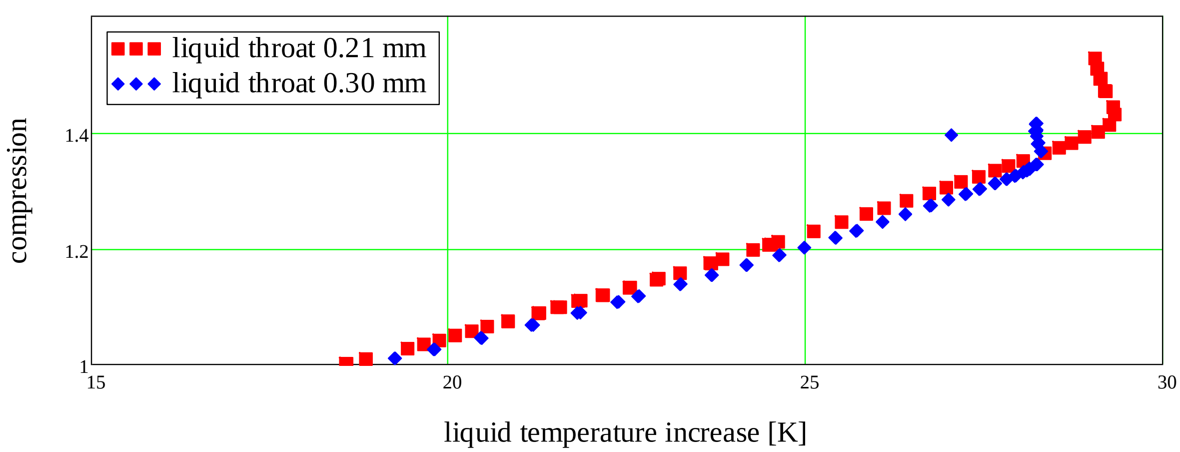

Figure 11. Compression ratio versus liquid phase temperature increase for: $U = 3.5$, $p_V = 10$ bar.

The results presented in Fig. 6 reveals very interesting features of the heat transfer and momentum process. It occurred that in the case of non-complete condensation there is possible to achieve relatively very high maximum compression ratio under stalling conditions as well as temperature increase of the liquid subcooled phase in comparison with operation under complete condensation conditions. This phenomenon may be intuitively explained as an effect produced by different physical conditions of heat and momentum transfer since in the case of the smallest liquid gap thickness the liquid film thickness is also the smallest which leads to formation of different two-phase flow patterns in comparison with significantly thicker liquid film cases. Nevertheless, these phenomena requires further investigations.

The measured film heat transfer coefficient related to the surface area of mixing chamber are presented in Fig. 7. The values of heat transfer coefficient were calculated from energy balance of the mixing chamber on the basis of the measured mass flow rates of motive and secondary fluids, temperature of liquid phase at the inlet and outlet, as well as static pressure in the mixing chamber.

The comparison of the performance characteristics of the operation of the injector for various motive pressures for the thickness of the liquid gap 0.21 mm and fixed mass flow rate U = 4.5 are presented in Fig. 8. As it is seen higher liquid temperature increase levels are achieved under conditions of higher motive vapour pressure and fixed vapour superheating. It is also possible to conclude that under these conditions stalling appears as more mild process.

The comparison of performance line for different vapour superheating and constant entrainment ratio U = 4.5 is given in Fig. 9 and 10. The motive pressure was p_V = 0.8 MPa. Liquid nozzle gap thickness was0.21 mm. Vapour superheating was lowering during experiments from nominal value of 7.0 K up to zero (saturation conditions). At saturation wet vapour was observed at nozzle inlet. Results shows that vapour superheating has a crucial influence on the mass flow rate, which increases. At this conditions higher compression ratio and higher temperature increase of liquid phase are achieved. Note, that refrigeration control systems operating with typical superheating at lever 4 – 5 K, and lower superheating of motive vapour might be difficult for control systems in order to provide proper and stable operation.

The comparison of performance characteristics of the operation of the injector for the fixed motive vapour pressure and vapour superheating as well as fixed mass entrainment ratio U = 3.5 for various liquid nozzle gap thickness is presented in Fig. 11. As it is seen within the zone of non-complete condensation process the effect of the gap thickness may be thought as negligible. However, for the smallest liquid gap thickness the performance characteristics may be thought as much more favourable since it is possible to achieved higher compression ratio so that the stalling appears under higher discharged pressures as well as higher temperature increase of liquid subcooled phase.

4. CONCLUSIONS

On the basis of the presented results the following conclusions may be drawn:

- The experimental investigations covered the measurement of the performance of the two-phase injector for isobutane as working fluid in terms of the compression ratio, heat transfer, and mass entrainment ratio. It was shown that intensive condensation heat transfer occurs inside the mixing chamber of the injector in spite of the unfavourable thermokinetic properties of isobutene as working fluid.
- The effect of the liquid gap thickness lays crucial role on momentum and heat transfer in the mixing chamber of the injector.
- The effect of incomplete condensation process strongly affects heat and momentum transfer inside the injector.
- Exceptionally high level of heat transfer coefficient was achieved in the injector so that the tested device may be thought not only as an alternative thermally driven liquid pump but also an effective direct contact condenser.

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ACKNOWLEDGEMENT

The work was completed within the statutory activities S/WM/2/2016