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Investigations Of Heat Transfer And Components Efficiencies In Two-Phase Isobutane Injector

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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 development of a suitable liquid pump. 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). Two-phase injector can be applied as a liquid pump in the ejection refrigeration systems. Paper deals with experimental investigations of the two-phase vapor-liquid injector as a liquid pump in refrigeration systems. The selected experimental results of the injector are presented for the case of isobutane as a working fluid. Investigations covered heat transfer coefficient as well as evaluation of the efficiencies of the components of the injector. It was demonstrated that these efficiencies depend on operation parameters and may not be treated as contact quantities.

1. INTRODUCTION

Low or medium temperature heat sources can be used as motive heat in thermally driven refrigeration systems: ejection systems and sorption systems (absorption and adsorption ones). Depending on their availability solar, geothermal and other renewable or waste heat sources may be applied for this purpose. However, in all of these systems liquid pump is the most problematic component.

Two-phase vapor-liquid injectors are driven by heat and this feature makes these devices as potential alternative to mechanical classic liquid pumps. The possible application of a two-phase injector instead of a mechanical pump for the ejection refrigeration system is presented in Fig. 1a. One of the advantages of the presented systems is lack of electric power needed to drive the system because the injector is driven by the same vapor as the vapor ejector. However, the injector requires only a small part of the vapor produced in the vapor generator. This feature makes that this system becomes “a green system” driven by heat only. Another important feature of the injector is its simple design in comparison with a mechanical pump. Injector does not require lubricants and has no moving parts.

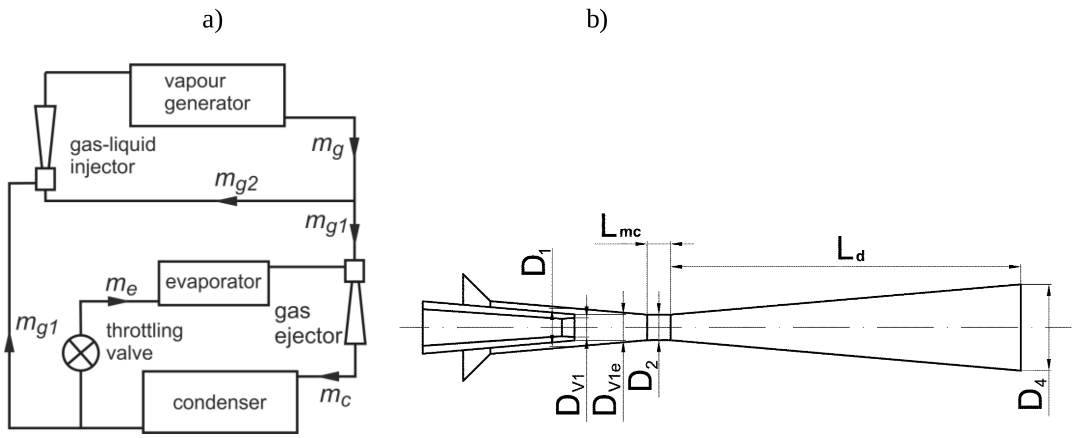

Figure 1: a) Diagram of modified ejection refrigeration system with two-phase injector; b) Schematic of the tested two-phase injector, dimensions are given in Table 1

The paper is strictly related with previous paper presented by authors (Smierciew et al. 2016). It can be stated that results presented here are the continuation of that investigations. Therefore, due to space limit of the present paper, some description dealing with literature reviews, experimental details and description of the test stand are not provided in this paper, however it can be found in Smierciew et al. (2015), (2016). The efficiency of the two-phase ejector components were investigated for the case of isobutane as working fluid (Buecker and Wagner, 2006). The dimensionless correlations were proposed for the component efficiencies. The pressure ratio was selected as the quantity that influences the component efficiencies of the two-phase ejector. However, no such investigations were reported before for the case of the two-phase injector. Moreover, because of complicated flow structure inside the injector it may be expected that the component efficiencies should depend on various quantities that describe two-phase flow inside the injector.

It is a clear need for the experimental data showing operation of two-phase vapor-liquid injector in refrigeration applications. In this paper the components efficiencies were investigated along with condensation heat transfer inside the mixing chamber. Because of complicated nature of the two-phase flow that is formed in the injector the components efficiencies should not be treated as constant quantities since they should depend on the two-phase flow features, i.e. two-phase flow pattern inside the injector components. This approach was proposed in the present paper. On the basis of the previous studies (Smierciew et al., 2015a,b) isobutane was selected as a promising and perspective low GWP working fluid.

2. INJECTOR DETAILS AND PERATING CONDITIONS

The schematic of the experimental test stand dedicated to the investigations of the two-phase injectors is presented in Smierciew, et al. (2016). Main dimensions of the tested injector are given in Table 1. The motive nozzle position can be changed within the range 0 – 0.50 mm, where 0 mm means no liquid flow since nozzle touches the mixing chamber surface. Therefore, the cross-section area for the entrained liquid is variable, and the thickness of the liquid gap (annulus) is defined as $\delta = D_1 - D_{V1s}$. For the liquid gap δ = 0.21 mm, the mixing chamber length is L_{mc} = = 2.73 mm, while for the liquid gap δ = 0.30 mm the mixing chamber length is L_{mc} = 3.75 mm.

Table 1. Dimension of the tested two-phase injector

description	symbol	value
inlet diameter of the mixing chamber inlet	D_1	3.29 mm
inner diameter of the motive nozzle outlet	D_{V1}	1.65 mm
outer diameter of the motive nozzle outlet	D_{V1e}	2.25 mm
diameter of the injector throat	D_2	2.20 mm
diameter of the diffuser outlet	D_4	7.45 mm
diffuser length	L_d	30.0 mm
constant area mixing chamber length	L_{mc}	2.0 mm
converging part of the mixing chamber angle		10°

Table 2. Variation of the operation parameters of the two-phase injector

motive pressure p_V	liquid gap $\Delta\delta$	entrainment ratio U	No. of measurement points
MPa	mm	-	-
1.1	0.30	variable in range 2.5-5.5	29
1.0	0.13	4.5	85
	0.21	2.5	247
		3.5	
		4.5	
		5.5	
	0.30	3.5	504
		4.5	
0.8	0.21	4.5	91
	0.30	4.5	61

The design of the tested injector enabled the change of the motive nozzle geometry as well as control of the distance of the nozzle outlet from the throat of the mixing chamber. This enables to investigate not only the effect produced by the change of the operation parameters but also to take into consideration the effect of the injector geometry.

The investigations of the performance of the injector were carried out under the steady-state operation conditions. For the case of the reported investigations, the motive pressure as well as vapor superheating were kept constant. Also, concerning the secondary fluid the parameters were kept constant, i.e. liquid subcooling as well as liquid pressure were fixed at constant levels. The discharge pressure was varied by means of the control valves applied at the discharged line of the injector. The measurements covered the operation conditions for the case of the lowest possible discharge pressures that correspond to totally opened control valve 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. The operation conditions of the tested injector are presented in Table 2.

3. EXPERIMENTAL RESULTS OF INJECTOR OPERATION

3.1 Motive nozzle

Motive nozzle in the vapor-liquid injector is the converging-diverging de Laval nozzle since the injector operates under conditions of suction to motive pressure ratio that is lower than critical pressure ratio. It is very common that this type of well performed nozzle for single phase vapor flow has efficiency higher than 0.90. The relationship between nozzle efficiency and velocity coefficient is $\eta = \varphi^2$, therefore, the velocity coefficient has almost constant value $\varphi_n > 0.95$. Because of this motive nozzle is not analyzed in this paper.

3.2 Liquid nozzle

In modeling of single or two-phase injector it is assumed that static pressure of liquid at cross-section 1 (see Fig. 2) is equal to vapor pressure at the motive nozzle outlet:

$$p_{V1} = p_{L1} = p_1 \tag{1}$$

Liquid pressure p_{l1} is lower than pressure at the injector inlet p_{l0}, and relation between velocity and pressure change is given by Bernoulli equation. Therefore, velocity of liquid at cross-section 1 can be calculated as follows:

$$w_{L1s} = \varphi_s \sqrt{\frac{2}{\rho_{L1}}\left(p_{L0} - p_{L1}\right) + \frac{\rho_{L0}}{\rho_{L1}} w_{L0}^2} \tag{2}$$

Eq. (2) requires that velocity coefficient has to be known. On the basis of the experimental results it can be seen that there is relation between velocity coefficient and mass entrainment ratio. This relation is shown in Fig. 2. The following relationship for velocity coefficient of the liquid nozzle is proposed:

$$\varphi_s = 0.0179U^3 - 0.2401U^2 + 1.0577U - 0.5266 \tag{3}$$

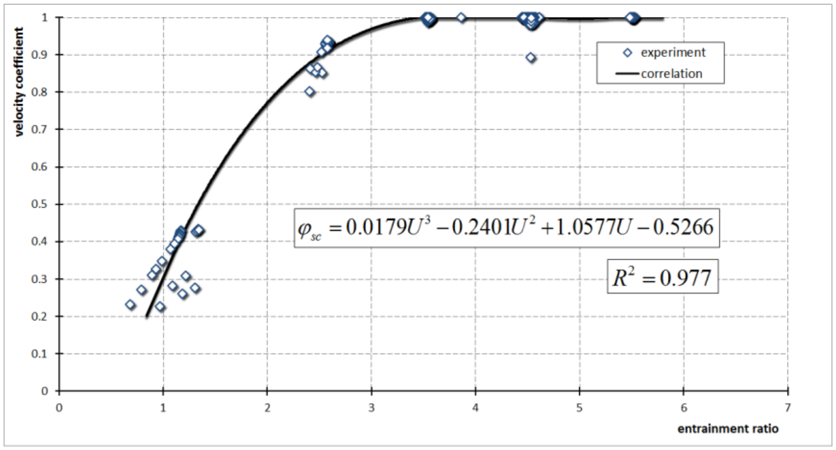

Figure 2: Velocity coefficient of the liquid nozzle with relation to mass entrainment ratio *U*.

The coefficient of determination R^2 = 0.977 was achieved. The proposed relationship was validated for entrainment ratios $U < 6.0$.

3.3 Mixing chamber

For the mixing chamber the momentum balance equation has the following form:

$$\frac{1}{\varphi_m}\dot{m}_2 w_{2s} + A_2 p_2 + (A_1 - A_2)p_{mw} = \dot{m}_{V1}w_{V1} + \dot{m}_{L1}w_{L1} + A_1 p_1 \tag{4}$$

and it can be applied for determination of the velocity coefficient φ_m. Loss of momentum in the mixing chamber is determined by several parameters which occur during condensation of motive vapor within very complex two phase flow. This includes the following criterial numbers:

- Jakob number:

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

- Reynolds number of vapor phase and liquid phase, respectively, at the inlet to the mixing chamber:

$$\mathrm{Re}_V = \frac{w_{V1}D_{V1}}{\nu_{V1}}, \quad \mathrm{Re}_L = \frac{w_{L1}\delta_1}{\nu_{L1}} \qquad \text{(6a), (6b)}$$

-Martinelli parameter:

$$X = \left(\frac{\rho_{V1}}{\rho_{L1}}\right)^{0.50}\left(\frac{\mu_{L1}}{\mu_{V1}}\right)^{0.10} \tag{7}$$

- dimensionless width of the liquid nozzle:

$$\Delta_1 = \frac{\delta_1}{D_2} \tag{8}$$

Following dimensionless relationship for the mixing chamber velocity coefficient is proposed:

$$\varphi_m = 7454 Ja^{-0.200}\,\mathrm{Re}_V^{-0.136}\,\mathrm{Re}_L^{-0.382}\,X^{1.890}\Delta_1^{0.362} \tag{9}$$

The coefficient of determination R^2 = 0.963 was achieved. Comparison of the velocity coefficient calculated on the basis of the experimental results with calculated on the basis of the proposed eq. (9) is shown Fig. 3. In proposed correlation the mass entrainment ratio is not included since in the correlation Reynolds numbers for both phases were taken into account. The proposed eq. (9) is valid for the following range of the dimensionless parameters: 420 000 < Re_V < 550 000; 13 380 < Re_L < 30 000; 0.180 < X < 0.22; 0.09 < Δ_1 < 0.14; 0.075 < Ja < 0.215.

3.4 Diffuser

The velocity coefficient for the diffuser may be calculated on the basis of Bernoulli equation as follows:

$$p_4 + \varphi_d^2\frac{\rho_4 w_{4s}^2}{2} = p_3 + \frac{\rho_3 w_3^2}{2} \tag{10}$$

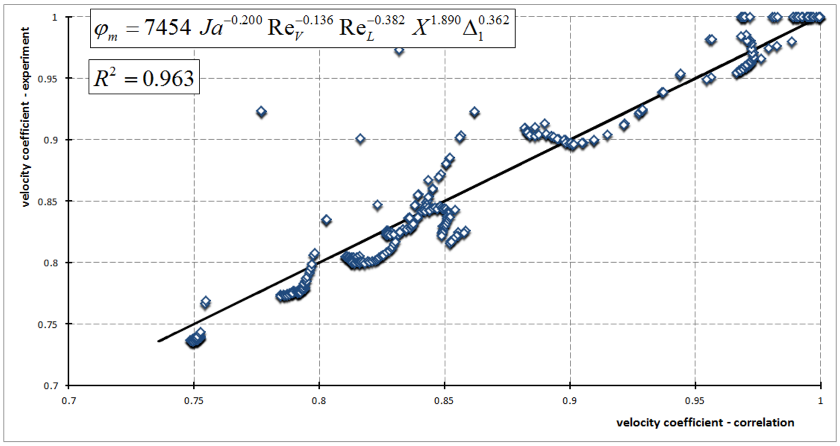

Figure 3: Comparison of calculated and experimentally determined velocity coefficients for the mixing chamber.

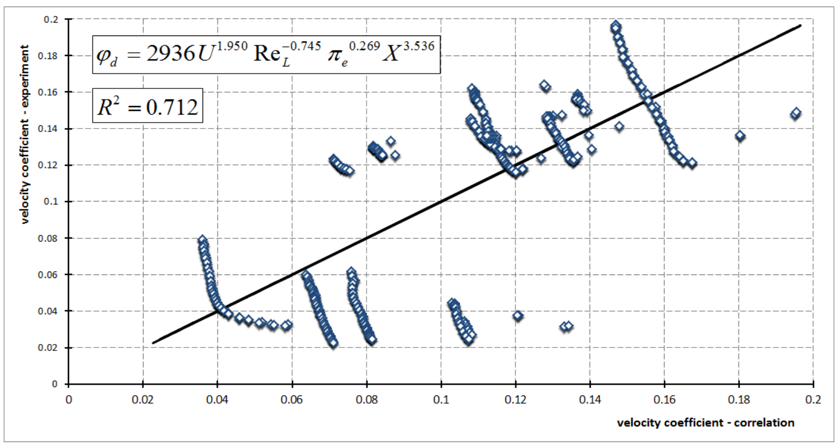

Figure 4: Comparison of calculated and experimentally determined velocity coefficients for the diffuser.

However, it should be taken into account that the diffuser may be thought as a specific injector component because of the nature of the flow pattern. In general the flow is very turbulent, unpredictable and unsteady, the flow can be single-phase or two-phase of various wet vapor quality. By the nature of the flow inside the diffuser, determination of the velocity coefficient for this component of the two-phase injector for the entire range of the operation conditions is thought to be very complex and problematic task due to large discrepancy and randomness of the results. Part of the experimental results indicates operation of the diffuser with liquid phase only, part of the results indicates that condensation shock wave is located inside the diffuser, and much larger pressure difference occurs. Such performance reveals the fact that operation of the diffuser does not correlates, or correlation is minor with any physical parameter. Nevertheless, the dimensionless relationship in the following form is proposed:

$$\varphi_d = 2936 U^{1.950}\ \mathrm{Re}_L^{-0.745}\ \pi^{0.269} X^{3.536} \tag{11}$$

The velocity coefficients calculated with use of the proposed eq. (11) and obtained from experimental results are shown in Fig 4. As it seen the coefficient of determination is $R^2 = 0.712$. The proposed correlation is valid for the conditions: $13\,380 < Re_L < 30\,000$; $0.180 < X < 0.22$; $0.8 < \pi < 1.6$; $U < 6$.

3.5 Mean heat transfer coefficient in mixing chamber

Liquid subcooling is a very important parameter, since it affects the flow parameters in the throat of the injector. In the mixing chamber, during condensation process, a significant part of the momentum transfer between phases proceeds as an effect of condensation heat transfer, where mass transfer is a parallel process. Note that saturation pressure and subcooled liquid temperature change during flow inside the mixing chamber. Heat transfer in this process can be described by balance equation:

$$\dot{m}_c \Delta h_m = \alpha_m A_{mw} \Delta T_m \tag{12}$$

where mean specific enthalpy difference is calculated as follows:

$$\Delta h_m = \left(\Delta h_1 - \Delta h_2\right) / \ln\left(\Delta h_1 / \Delta h_2\right) \tag{13a}$$

and

$$\Delta h_1 = h_{V1}\left(p_1, T_{V1}\right) - h_{L1}\left(p_1, T_{L1}\right) \tag{13b}$$

$$\Delta h_2 = h''\left(p_2\right) - h_{L2}\left(p_2, T_{L2}\right) \tag{13c}$$

Analogically, mean temperature difference is calculated as follows:

$$\Delta T_m = \left(\Delta T_1 - \Delta T_2\right) / \ln\left(\Delta T_1 / \Delta T_2\right) \tag{14a}$$

where

$$\Delta T_1 = T_{V1} - T_{L1}\ ;\ \Delta T_2 = T''\left(p_2\right) - T_{L2} \tag{15b}$$

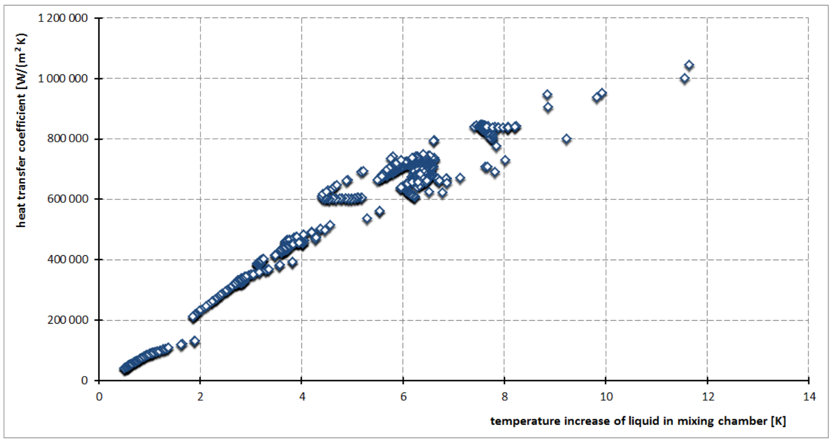

Figure 5: Condensation film heat transfer coefficient versus temperature increase of liquid phase.

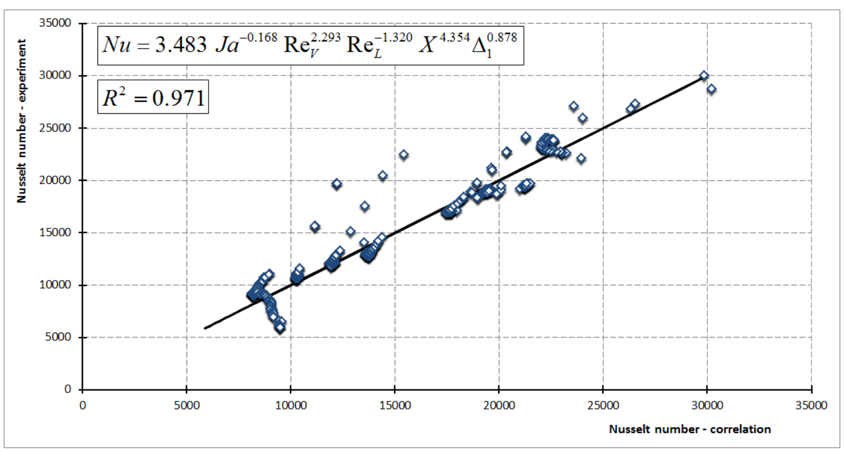

Figure 6: Comparison of calculated and experimentally determined Nusselt number.

On the basis of eq. (12) and given experimental data mean heat transfer coefficient for the mixing chamber was determined, see Fig. 5. In general, mean heat transfer coefficient is representing by dimensionless Nusselt number:

$$Nu = \frac{\alpha D_2}{\lambda_{L1}} \tag{16}$$

Mean heat transfer coefficient calculated from eq. (5) allows for development of general form of correlation for Nusselt number:

$$Nu = 3.483\ Ja^{-0.168}\ \mathrm{Re}_V^{2.293}\ \mathrm{Re}_L^{-1.320}\ X^{4.354}\Delta_1^{0.878} \qquad (17)$$

For proposed correlation the coefficient of determination R^2 = 0.971 is achieved. Comparison of Nusselt number obtained on the basis of the experimental results versus Nusselt number predicted by eq. (17) is presented in Fig. 6. It is seen that correlation very well predicts the Nusselt number for the tested injector. Proposed correlation can be used as the closure equation in modelling of operation of the two-phase injectors for the case of refrigerants as working fluid. Proposed correlation is valid for: 420 000 < Re_V < 550 000; 13 380 < Re_L < 30 000; 0.180 < X < 0.22; 0.09 < Δ_1 < 0.14; 0.075 < Ja < 0.215.

6. CONCLUSIONS

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

- The operation of the two-phase vapor-liquid injector for isobutane as a working fluid was tested systematically. To the knowledge of the authors these results are the first ones in literature for the two-phase vapor-liquid injector operating with refrigerant as a working fluid.
- Although the compression efficiency of the tested injector is less than 1%, the total efficiency of the injector for was less than 28% for compression ratio up to $\pi \approx 1.6$ and for the entrainment ratio up to U = 5.5.
- The performance of the tested two-phase injector differs under complete and incomplete condensation conditions. Under conditions of the uncomplete condensation process the effect of the liquid gap thickness may be thought as negligible for performance of the two-phase ejector.
- High level of heat transfer coefficient up to 1 MW(m^2×K) was measured for the tested two-phase ejector. The proposed dimensionless relationship for heat transfer was proposed.
- On the basis of the obtained experimental results it may be concluded that the velocity coefficients for components of the injector may not be treated as constant quantities. The dimensionless relationships for the velocity coefficients of the injector components were proposed.
- The paper presents the first experimental results obtained for the case of the isobutane as a working fluid for the two-phase injector. Since nature of the physical phenomena that occur inside the tested injector are very complicated therefore it is a clear need for further analysis and modelling of the operation of the injector.

NOMENCLATURE

A	area	m^2
D	diameter	mm
c	specific heat at constant pressure	kJ/(kg×K)
h	specific enthalpy	kJ/kg
h_{fg}	latent heat of vaporization	kJ/kg
Ja	Jacob number	(-)
L	length	m
$\dot{m}$	mass flow rate	kg/s
Nu	Nusselt number	(-)
p	pressure	MPa
R	coefficient of determination	(-)
Re	Reynolds number	(-)
U	mass entrainment ratio	(-)
T	temperature	°C, K
w	velocity	m/s
X	Martinelli parameter	(-)
α	heat transfer coefficient	W/(m^2×K)
δ	liquid gap thickness	mm
Δ_1	dimensionless width of liquid nozzle	(-)
η	efficiency	(-)
λ	thermal conductivity	W/(m×K)
μ	dynamic viscosity	kg/(m×s)
ν	kinematic viscosity	m^2/s
π	compression ratio	(-)

ρ	density	kg/m^3
φ	velocity coefficient	(-)

Subscript

c	compression, condensate
d	discharge, diffuser
dL	liquid at the vapor generator inlet
e	ejector, total
L	liquid state
m	mixing flow
mc	mixing chamber
mw	mixing chamber walls
n	nozzle
s	suction chamber
sL	liquid phase at the injector suction chamber
V	motive vapor
″	saturation vapor state

0 – 4 cross-sections of the injector, respectively: vapor inlet, liquid inlet, ejector throat, shock wave, discharge

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