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Analysis of application of feed-water injector heaters to steam power plants

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Abstract

Steam-water injectors are devices in which exchange of mass, momentum and energy between two fluids being in direct contact, occurs. They can operate as pumps, mixers or direct contact heat exchangers. In the last aspect their use as feed-water heaters in Rankine thermal cycle of steam power plants both in land and sea applications (to merchant and naval ships) is very interesting. This paper presents selected results of heat-and-flow investigations of a supercritical steam-water injector, obtained in Institute of Fluid Flow Machinery, Polish Academy of Sciences (IMP PAN). On their basis value of average heat transfer coefficient for mixing chamber was determined; the obtained values were even a few dozens greater than those for classical shell-and-tube heaters. In the theoretical part of this work is presented an original injector model based on balances of mass, momentum and energy, written for control volumes containing separately particular elements of injector. On the basis of the model flow parameters in characteristic cross-sections of injector were determined. The calulations were performed for two different injectors tested in IMP PAN (Gdańsk) and SIET (Piacenza, Italy), and their good compliance with experimental data was achieved.

Keywords: *power plants, feed-water heaters, supercritical steam-water injector, injector model based on balances of mass.*

Introduction

Steam-water injectors are devices in which exchange of mass, momentum and energy occurs due to direct contact between two fluids. In 1901 an injector was used by Charles Parsons to air extraction from condensers, and in 1910 Maurice Leblanc applied an injector to a cooling device. Generally, injectors can operate efficiently as pumps, mixers or direct contact heat exchagers. In engineering a special role is given to double phase steam-water injectors. In recent years interest to such injectors has increased because of new proposals of their application to Rankine cycle of thermal power plants both land-based and marine ones (on merchant and naval ships). The most important advantages of such injectors consist in their possible operation without electric supply, rather small gabarites and lack of movable parts, resulting in their high operational reliability.

Two-phase steam-water injectors can be generally split into two kinds: (a) -subcritical ones (Fig. 1a) and (b) -supercritical ones (Fig. 1b). In both the cases water steam is their driving medium. In subcritical injector steam expands in the tapered steam nozzle SN and its discharge velocity w_{V1} is smaller than or at most equal to the sonic velocity a, ($w_{V1} \leq a$), hence Mach number reaches value: $M \leq 1$. In the second case (Fig. 1b) water steam expands down to the pressure smaller than critical one, as a result the discharge velocity from the nozzle is greater than the sonic one, hence Mach number reaches value: $M \geq 1$. Discharge steam from the nozzle flows into the mixing chamber MC and contacts there with cold water. Due to difference of temperatures and velocities between both the phases heat exchange (steam condensation) and momentum exchange takes place. The processes terminate in vicinity of mixing chamber throat, that is manifested by a sudden rise of pressure, characteristic for shock wave. Behind the wave, only the liquid flows through the diffuser. From the cognitive point of view the steam-water injector exemplifies a device which, though of simple design, is characteristic of a variety of flow phenomena. In the two-phase flow zone (covering the mixing chamber and condensation shock wave) drastic change in flow structure takes place – from stratified (annular) flow to homogeneous (bubbly one). Also flow velocity undergoes large changes – from a high subcritical value (smaller than about 450 m/s for the version a) or supercritical one (of the order of 500÷1000 m/s for the version b) at inlet to the mixing chamber, down to the value of the order of 20 m/s in the diffuser.

In this paper the results are presented of the authors' experimental investigations which covered measurements of pressure and temperature distribution along injector at different flow parameters, between inlet to and outlet from the injector. The investigations were carried out for the supercritical injector, (Fig. 1b), [1], i.e. the case considered more general. The critical injector (M = 1) is finally obtained by shortening the divergent part of the steam nozzle at maintaining the same inlet steam parameters. If steam pressure at inlet to the nozzle SN is now decreased then subcritical flows in the nozzle are obtained, and at the inlet: M < 1.

On the basis of the investigations were determined the maximum values of the injector discharge pressure (p_{4max}) at which the device can operate stably. Also, value of the average heat transfer coefficient for mixing chamber as well as injector efficiency, were determined. The injector's theoretical model based on balances of mass, momentum and energy, written for control volumes containing flow through particular elements of the injector, i.e. steam nozzle, water nozzle, mixing chamber, condensation shock wave zone and diffuser, shown in Fig. 1, has been also presented. By solving the model's equations for given parameters of driving steam and sucked-in water, flow parameters were determined in the injector characteristic cross-sections located on the boundaries of the above mentioned control volumes.

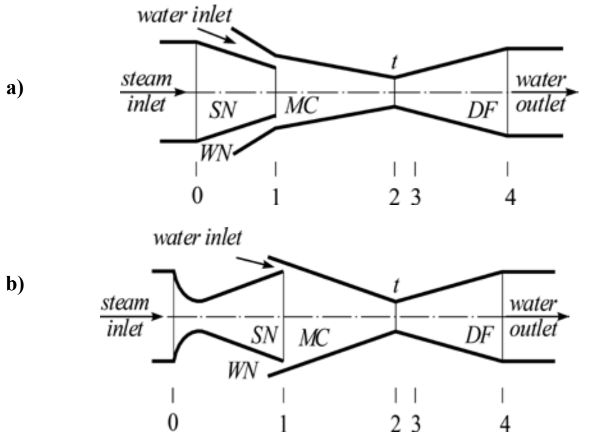

Fig. 1. Schematic diagram of steam – water injector: a) subcritical injector, b) supercritical injector. Main parts of injector : SN – steam nozzle, WN – water nozzle, MC – mixing chamber, DF – diffuser, t – throat

Experimental investigations of steam-water injector

In the Institute of Fluid Flow Machinery of Polish Academy of Sciences, Gdańsk, experimental investigations of the supercritical two-phase steam – water injector driven by water steam (Fig. 1), were performed with the use of the laboratory model [1]. They were carried out by using the so called „short" injector of the mixing chamber pressure $p_{MC} \cong 20\,\mathrm{kPa}$, with the aim to determine two quantities characteristic for injectors, namely: the maximum pressure p_{4max} and value of the average heat transfer coefficient $\overline{\alpha}$ for the mixing chamber MC. The tests were a continuation of the work carried out previously with the use of the so called „long" injector (of the pressure $p_{MC} \cong 8\,\mathrm{kPa}$) [2, 3] with the aim to investigate their applicability to feeding nuclear reactor in failure states. Inlet parameters of the tested laboratory injector could be controlled within the following ranges: steam flow rate between 85÷130 kg/h, inlet steam pressure down from 0.5 MPa, steam superheating temperature between 0÷40 ºC, water flow rate between 1÷6 m^3/h, water temperature between 15÷20 ºC. It was also possible to change conditions of two-phase flow through mixing chamber by controlling the water nozzle gap size within the range of 0.5÷1.5 mm.

During the steam-water injector investigations, apart from the above mentioned inlet parameters, also temperature and pressure distributions along mixing chamber and diffuser, were recorded. Moreover, value of counterpressure at outlet from the injector, possible to be set by means of control valve, was measured. In Fig. 2 typical distributions of pressure and temperature along mixing chamber and diffuser, measured at different values of discharge pressure (Fig. 2a) and water flow rate (Fig. 2b), are presented. During investigations of injector performance characteristics the maximum discharge pressure at different values of the injection coefficient $U = m_{L0}/m_{V0}$ for three selected values of water nozzle gap, was determined. Results of the measurements are presented in Fig. 3. On the basis of the obtained characteristics it can be concluded that an increase of injector discharge pressure can be achieved both by increasing the coefficient U and mass flow of steam and or water at inlet to mixing chamber.

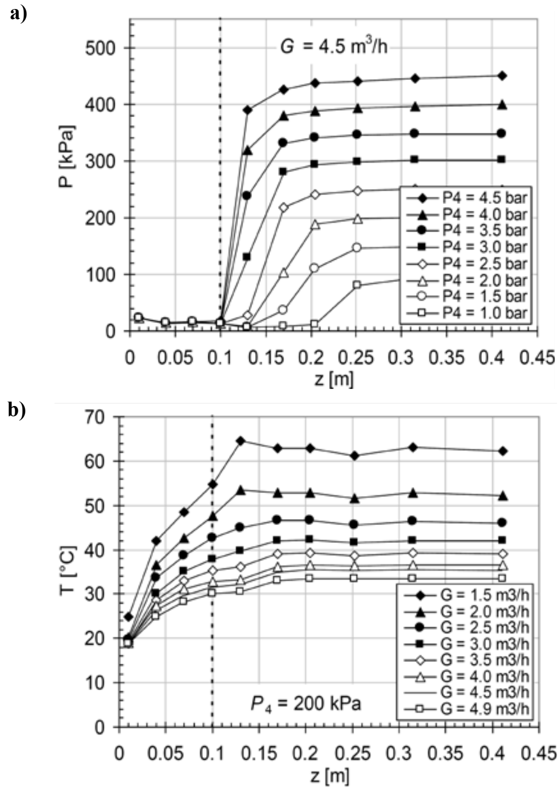

Fig. 2. Example profiles of pressure, (a), and temperature, (b), measured in mixing chamber and diffuser of a steam-water injector operating with selected values of the outlet pressure p4 and water flow rate G. The inlet steam mass flux.

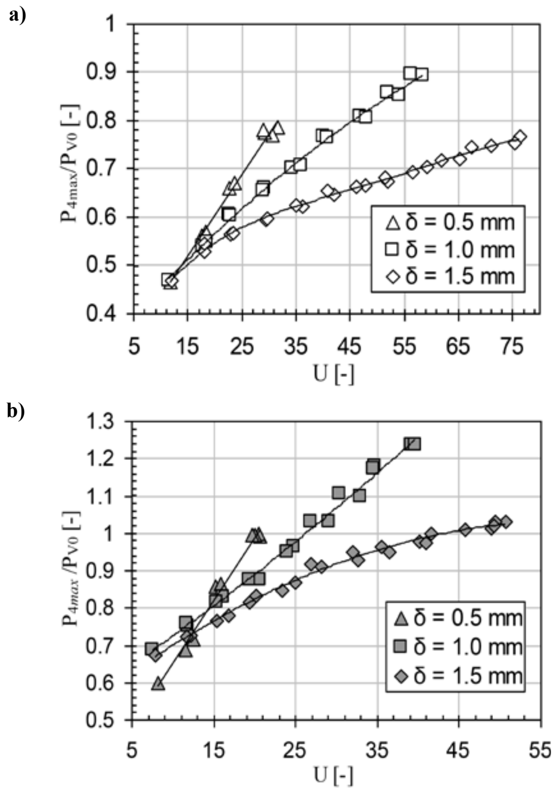

Fig. 3. Values of the dimensionless maximum outlet pressure p_{4max}/p_{V0} in function of the injection coefficient U, measured for three selected values of the water nozzle gap δ at the inlet steam flux equal to:
a) 85 kg/h, b) 125 kg/h.

Heat exchange in mixing chamber

The average heat transfer coefficient for the injector mixing chamber was determined on the basis of experimental data by using the method proposed in [3, 4]. By assuming that the flow of both phases through injector mixing chamber is one-dimensional and balanced it is sufficient, in order to determine thermodynamical parameters of water, to measure temperature only as steam parameters are depending only on pressure. Taking the above into account one can write the following energy balance equation for the mixing chamber:

$$m_{V1} h_{V1} + m_{L1} h_{L1} = (m_{L1} + m_c) h_L(T_3) + (m_{V1} - m_c) h_V(p_3) \quad (1)$$

from which it results that the condensed steam mass flow is equal to:

$$m_c = \frac{m_{V1}\left[h_{V1} - h_V(p_3)\right] + m_{L1}\left[h_{L1} - h_L(T_3)\right]}{h_L(T_3) - h_V(p_3)} \quad (2)$$

Now the average heat transfer coefficient for mixing chamber of the investigated steam–water injector can be calculated as follows:

$$\overline{\alpha} = \frac{m_c h_{fg}}{A_{MC} \Delta \overline{T}_{MC}} \quad (3)$$

The so determined $\overline{\alpha}$ values were generalized and presented in dimensionless form with the use of similarity numbers resulting from dimensional analysis of the equations of mass, momentum and energy conservation, written for flow through injector [1,6]. The dimensional analysis resulted in a correlation function expressed in the form of Nusselt number dependent on dimensionless similarity parameters, for the average heat transfer coefficient.

With the use of the authors' experimental data for the two injectors which have been so far examined in IMP PAN [1,2,3] (comprising altogether 2051 measurement points from testing the „short" injector as well as the „long" one) the following correlation function was obtained for the average Nusselt number:

$$Nu = 0.00284\ \mathrm{Re}_v^{0.974}\ S^{0.186}\ U^{0.112}\ \xi^{0.605} \quad (4)$$

where:

$$\mathrm{Nu} = \frac{\overline{\alpha}\ D_{SN}}{\lambda_{L1}}$$

$$Re_V = \frac{\rho_V\ w_V\ D_{SN}}{\mu_V}$$

$$U = \frac{m_{L1}}{m_{V1}}$$

The correlation (4) is valid for values of the dimensionless parameters contained within the following intervals, respectively:

$$88240 \le \mathrm{Re}^v \le 234950\ ,\ 24 \le S^i \le 211\ ,\ 11 \le U \le 76\ ,\ 50 \le \xi \le 111$$

together with the correlation coefficient R = 0.91. Physical parameters of water and steam as well as flow velocities in both the phases are calculated at the inlet cross-section of mixing chamber.

In order to make use of the correlation (4) to know geometry of steam and water nozzles is necessary because pressure and velocities at inlet to mixing chamber are associated with it. In Fig. 4 are presented example calculation results of heat transfer coefficient in compliance with the correlation (4), for two values of steam mass flow rate and one value of water flow rate, which correspond with the conditions of the experiments carried out in IMP PAN. The points representing two values of the mixing chamber pressure p_{MC}, equal to 8 and 20 kPa, are depicted on the diagram. The steam flow velocities w_{V1} as well as the steam nozzle outlet diameters D_{SN} corresponding with the pressures were determined on the basis of thermodynamical calculations of expansion in steam nozzle (from the initial pressure p_{V0} to the final pressure p_{MC}). As the water nozzle outlet area was known, the water flow velocity w_{L1} and the slip ratio S could be determined. The term expressing the physical properties ξ was determined for the given pressure p_{MC} from the water vapour tables. From the calculations two important observations result:

- In mixing chamber the heat transfer coefficient $\overline{\alpha}$ obtains very large values.
- Values of the coefficient $\overline{\alpha}$ are increasing along with pressure increasing in mixing chamber. For the pressure p_{MC} = 8 kPa its average value amounted to 350 kW/m^{2}K, and for the pressure p_{MC} = 20 kPa - as much as 725 kW/m^{2}K.

Such influence of the pressure on intensity of steam condensation on water film was confirmed by results of the injector investigations performed in Japan [12].

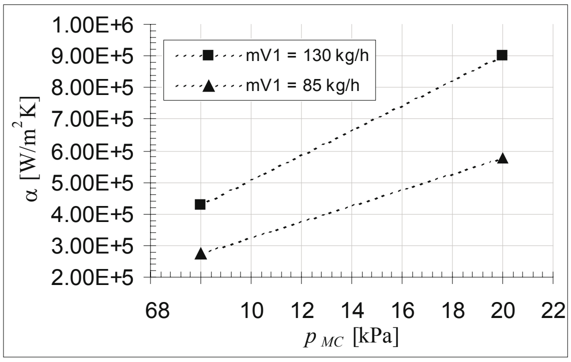

Fig. 4. Run of mean values of mean heat transfer coefficient, $\overline{\alpha}$, for mixing chamber, calculated acc. (4) in function of the pressure p_{MC} for two selected values of the steam mass flux $\dot{m}_{V1}$, at a set value of the outlet nozzle diamater D_{SN}, and the water mass flux $\dot{m}_{L1}$= 3000 kg/h..

Calculation of maximum discharge pressure

Modeling the flow through steam-water injector faces great difficulties especially in the zone of two-phase flow in mixing chamber and condensation wave zone. The difficulties result first of all from the complexity of flow structures occurring successively in various parts of mixing chamber (annular, drop-like and bubble-like), presence of phase transformations (condensation on water film and in shock wave) and large changes in flow velocity (interphase slip at inlet to mixing chamber, transition from supercritical to subcritical shock wave). However it is possible to determine the maximum injector discharge pressure by using a simple model based on balances of mass, momentum and energy, written for control volumes containing mixing chamber and condensation shock wave zone, separately.

In the proposed model equations which describe flow through steam nozzle, and water nozzle as well, can be solved for parameters given at inlet to injector, and consequently yield the flow parameters at inlet to mixing chamber (marked '1'). Next, the solution of the balance equations of mass, momentum and energy for two-phase mixture in mixing chamber provides data to solve the set of equations for the successive control volume (between the cross-sections '2' and '3'), i.e. the shock wave zone. Now, on the basis of Bernoulli equation for liquid (water) flow through diffuser the pressure and temperature at outlet from injector, (marked '4'), can be determined.

The equations which describe driving steam expansion in steam nozzle were obtained under the assumption that the flow is adiabatic and irreversible. The steam parameters at outlet from the nozzle (cros-section '1') result from the following mass and energy balances:

where s -index stands for isentropic expansion value (i.e. when $s_{V1} = s_{V0}$). In real flow irreversibility losses contribute to lowering the discharge velocity w_{V1}, that can be taken into account by applying the velocity coefficient

$$A_{V0}\rho_{V0}w_{V0} = A_{V1}\rho_{V1}w_{V1} \tag{5}$$

$$h_{V0} + \frac{w_{V0}^2}{2} = h_{V1s} + \frac{w_{V1s}^2}{2} = h_{V1} + \frac{w_{V1}^2}{2} \tag{6}$$

c_v defined as follows:

$$c_v = \frac{w_{V1}}{w_{V1s}} \tag{7}$$

The coefficient takes into account not only losses resulting from friction but also due to condensation (when expanding steam parameters exceed Wilson's line) and formation of shock wave in the vicinity of the steam nozzle outlet (which takes place when at inlet to the mixing chamber a difference between steam and water pressure appears). As results from the investigations presented in [2], the above mentioned irreversible phenomena cause that value of the coefficient c_v for injector steam nozzle is relatively low, i.e. equal to about 0.9.

Under the relation (7) the double equation (6) can be transformed to the following form:

$$h_{V1} = h_{V1s} + \left(1 - c_v^2\right)\frac{w_{V1s}^2}{2} = c_v^2 h_{V1s} + \left(1 - c_v^2\right)\left(h_{V0} + \frac{w_{V0}^2}{2}\right) \tag{6a}$$

In the typical steam-water injector the driving steam is superheated at inlet to the nozzle but during expansion it enters wet-steam zone, hence the balance equations make the below given constitutive relationships to be complete:

$$\rho_{V0} = \rho\left(p_{V0}, h_{V0}\right)\ ,\ s_{V0} = s\left(p_{V0}, h_{V0}\right) \tag{8a}$$

$$h_{V1s} = x_{1s}h_{V,sat}\left(p_{V1}\right) + \left(1 - x_{1s}\right)h_{L,sat}\left(p_{V1}\right) \tag{8b}$$

$$x_{1s} = \frac{s_{V0} - s_{L,sat}\left(p_{V1}\right)}{s_{V,sat}\left(p_{V1}\right) - s_{L,sat}\left(p_{V1}\right)} \tag{8c}$$

$$\rho_{V1} = \frac{x_1}{\rho_{V,sat}\left(p_{V1}\right)} + \frac{1 - x_1}{\rho_{L,sat}\left(p_{V1}\right)} \tag{8d}$$

$$x_1 = \frac{h_{V1} - h_{L,sat}\left(p_{V1}\right)}{h_{V,sat}\left(p_{V1}\right) - h_{L,sat}\left(p_{V1}\right)} \tag{8e}$$

where x stands for steam quality and the index sat concerns parameters in state of saturation.

Water nozzle discharge parameters can be determined under the assumption that pressure of water is the same as that of steam: $p_{V1} = p_{L1} = p_1$, and its temperature is the same as that at inlet to injector. Then $\rho_{L1} = \rho(p_{V1}, T_{L0})$ and the mass conservation equation:

$$\dot{m}_{L0} = A_{L1}\rho_{L1}w_{L1} \tag{9}$$

is sufficient to determine the water nozzle discharge velocity w_{L1}.

Flow parameters on the mixing chamber boundaries satisfy the balances of mass, momentum and energy, counted between the inlet cross-section 1 and the throat 2 (see Fig. 1), which are expressed by the following equations:

$$A_2\rho_2 w_2 = A_{V1}\rho_{V1}w_{V1} + A_{L1}\rho_{L1}w_{L1} \quad \text{or} \quad m_2 = m_{V1} + m_{L1}, \tag{10}$$

$$m_2 w_2 + A_2 p_2 + (A_1 - A_2)p_{MC} = m_{V1}w_{V1} + m_{L1}w_{L1} + A_1 p_1 \tag{11}$$

$$m_2\left(h_2 + \frac{w_2^2}{2}\right) = m_{V1}\left(h_{V1} + \frac{w_{V1}^2}{2}\right) + m_{L1}\left(h_{L1} + \frac{w_{L1}^2}{2}\right) \tag{12}$$

It was assumed here that steam phase is always in state of saturation and in the cross-section 2 both the phases (steam and water) move with the same velocity w_2. Therefore to determine the enthalpy h_2 the relationship valid for homogeneous mixture can be used:

$$h_2 = h\left(p_2, \rho_2, T_{L2}\right) = x_2 h_{V2}(p_2) + (1 - x_2)h_{L2}(\rho_{L2}, T_{L2}) \tag{13}$$

Moreover, it was assumed that the average pressure which acts on the mixing chamber conical surface and appears in the momentum balance equation (11), is equal to $p_{MC} = (p_1+p_2)/2$.

When the mixing chamber inlet parameters (cross-section 1) are known to solve the set of balance equations (10)÷(12) is still necessary in order to know the liquid phase temperature in the throat, T_{L2}, which is lower than the steam temperature T_{V2}. The temperature T_{L2} can be determine on the basis of heat flow which rises water temperature during condensation of driving steam in mixing chamber. The heat flow is connected with the average heat transfer coefficient $\overline{\alpha}$ by the relation (4).

The balance equations for the shock wave zone (between the cross-sections 2 and 3) are similar to those for mixing chamber and take the following form:

$$A_3\rho_3 w_3 = A_2\rho_2 w_2 \quad \text{or} \quad m_3 = m_2 \tag{14}$$

$$m_3 w_3 + A_3 p_3 + (A_2 - A_3)\, p_{SW} = m_2 w_2 + A_2 p_2 \tag{15}$$

$$m_3\left(h_3 + \frac{w_3^2}{2}\right) = m_2\left(h_2 + \frac{w_2^2}{2}\right) \tag{16}$$

In the shock wave zone complete steam phase condensation takes place, hence in the cross-section 3 only liquid (water) appears, whose enthalpy amounts to

$$h_3 = h(p_3,\rho_3).$$

The average pressure which acts, within the shock wave zone, on the diffuser conical surface, was determined from the approximate relation: $p_{SW} = (p_2+p_3)/2$. Shock wave width and location of the cross-section 3, resulting from it, was determined on the basis of pressure profile measurements.

On the basis of the balance equations (14)÷(16) e.g. water pressure in the cross-section 3 can be determined. By taking into account the pressure increase determined form the Bernoulli equation:

$$p_4 + \frac{\rho_4 w_4^2}{2} = p_3 + (1-\zeta)\frac{\rho_3 w_3^2}{2}$$

$$\text{where} \quad \rho_4 = \rho_3,\; w_4 = \frac{A_3}{A_4} w_3 \tag{17}$$

value of the injector discharge pressure p_4 can be achieved. Value of the pressure loss coeficient ζ in (17) can be determined acc. [5]. As in the equations (14)÷(16) it was assumed that shock wave is located in direct vicinity of mixing chamber throat the determined pressure p_4 obtains its maximum value to keep injector's operation stable.

The presented model was used to determine the maximum discharge pressure for the flow conditions complying with the measurements performed in IMP PAN, [1], and presented in Fig. 3b, as well as with the investigations carried out in SIET (Piacenza, Italy), [7,8]. Relatively good calculation conformity was obtained between the above mentioned experiments (Fig. 5 and 6), namely with the error of about 15%.

Injector efficiency

In the subject-matter literature different definitions of injector efficiency can be met [9,10]. It is usually the ratio of compression work done within injector and steam energy delivered to the injector. As steam-liquid injector

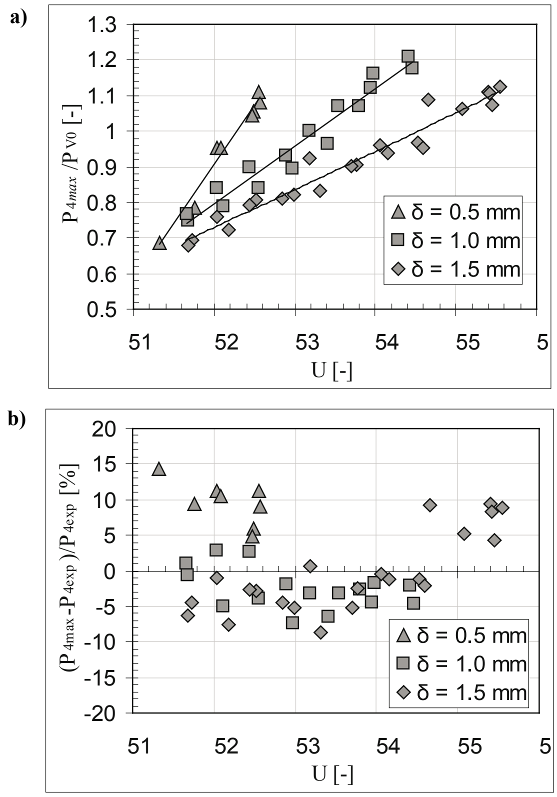

Fig. 5. *Results of calculations of the maximum outlet pressure p_{4max} for the flow conditions given in Fig. 3b: a) calculated dimensionless maximum outlet pressure, b) relative difference between the calculated value, p_{4max}, and measured one, p_{4exp}.*

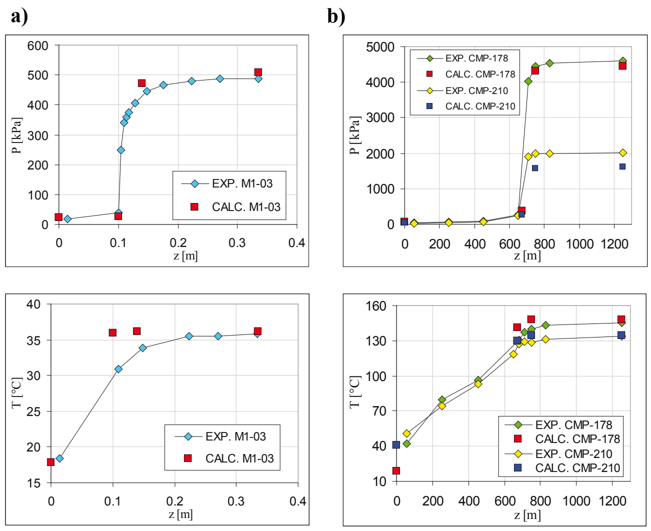

Fig. 6. *Measured and calculated values of the pressure p and water temperature T on wall of mixing chamber and diffuser : the left column (a) concerns IMP- PAN conditions, the right one (b) concerns SIET conditions (Piacenza, Italy) [7,8].*

can operate as a heat exchanger or jet pump therefore acceptance of two definitions of injector efficiency is justified.

In the case of injector operation as a heat exchanger it is proposed to use the notion of the exergetic efficiency η_b defined as the ratio of exergy of working fluid discharged from injector and exergy delivered to it:

$$\eta_b = \frac{(\dot m_{v0} + \dot m_{L0})\, b_4}{\dot m_{v0}\, b_{v0} + \dot m_{L0}\, b_{L0}} = (1+U)\frac{b_4}{b_{v0} + U\, b_{Lo}} \tag{18}$$

Calculation results of the efficiency for two versions („long" and „short") of the injectors examined in IMP PAN are presented in Fig. 7a.

In the case of injector operation as a jet pump, water exergy after compression is not so much important as its pressure part [11]. Hence the jet pump efficiency ηp can be defined as the ratio of compression mechanical work and delivered exergy:

$$\eta_p = \frac{(\dot{m}_{v0} + \dot{m}_{L0})}{\rho_L} \frac{p_4 - p_{MC}}{\dot{m}_{v0}\, b_{v0} + \dot{m}_{L0}\, b_{L0}} = \frac{1+U}{\rho_L} \frac{p_4 - p_{MC}}{b_{v0} + U\, b_{L0}} \quad (19)$$

Calculation results of the jet pump efficiency η_p are presented in Fig. 7b. From comparison of the efficiency η_b with the ηp it can be observed that the first is much greater.

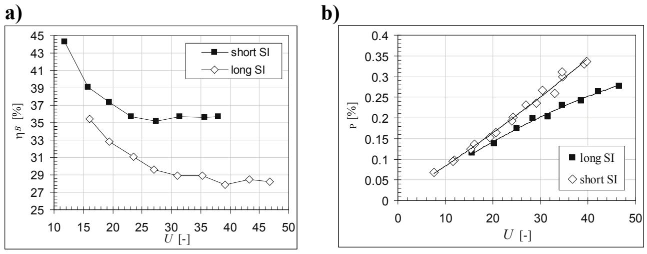

***Fig. 7**. Efficiency calculated for two versions of the injector experimentally tested at IMP PAN: a) exergetic efficiency, b) injector pump efficiency.*

Final remarks

In this paper the comprehensive results have been presented of model tests of two-phase steam-water injector, aimed at its application to steam power plants operating in compliance with Rankine cycle. In such cycle the so called regeneration heating of water feeding the boiler takes place within shell-and-tube heater. It is planned to use steam-water injectors instead of shell-and-tube heaters, that could significantly lower investment cost of power plant.

The model tests of two-phase steam-water injector performed in IMP PAN were aimed at determination of the coefficient of heat transfer between steam and water in mixing chamber as well as the maximum pressure of water discharged from the injector.

In the theoretical part of the work the injector's model based on the balance equations of mass, momentum and energy, written for particular injector elements, was elaborated. In the model correlation of heat exchange within mixing chamber was used to determine temperature difference between steam and water in the mixing chamber's throat. The model makes it possible to determine, apart from flow parameters in characteristic control cross-sections, also the maximum pressure p_{4max} at outlet from the injector. Correctness of the model, in the aspect of the pressure, was examined with taking into account the authors' experiments as well as those performed in SIET (Piacenza, Italy). Good comformity between results of the calculations and experiments was achieved as the error amounted to about 15% within a wide range of flow paramters at inlet to injector.

The experimental test results of heat exchange in injector mixing chamber confirmed that to achieve very high values of the heat transfer coefficient $\overline{\alpha}$, reaching even 700 kW/m² K, is possible. They are over 100 times greater than the heat transfer coefficient k in shell-and-tube heaters (which reaches the value of about 3 kW/m² K). The so large values of $\overline{\alpha}$ could make it possible to decrease, by about a hundred times, surface area of steam-water contact in mixing chamber (at the same value of transferred heat flow rate), that could potentially result in the decreasing of gabarites of an alternative heat exchanger in the form of injector, by about 10-times. Hence, such application of the injector could significantly lower investment cost of steam power plant.

The lowering of gabarites of steam power plant devices is especially important in case of merchant and naval ships where shortage of space takes place. By applying injector water heaters their number could be increased, that could result in increasing Rankine cycle efficiency and -in consequence -lowering operational costs. Another factor which leads to improving operational parameters of the cycle fitted with feed-water injectors is lack of thermal degeneration. In injectors, in contrast to shell-and-tube heaters, heat transfer occurs by direct contact of vapour phase with liquid one, hence in such devices heat transfer effectiveness is not worsened due to accumulation of contaminations on heat transfer surface during their long-lasting operation. Preliminary estimations indicate that the increase of Rankine cycle thermal efficiency by about 0.2 %, resulting from the above, can be expected. Another gain can be achieved due to injector pump effect which makes it possible to decrease feed-water pump power by a few dozen percent.

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Nomenclature

A	– cross-section area, m^2 ;
a	– sonic velocity, $a = (dp/d\rho)^{1/2}_{s=const}$;
A_{MC}	– surface area of mixing chamber wall, m^2 ;
b	– specific exergy, kJ/kg;
D, d	– diameters, m;
G	– volumetric water flow rate, m^3/h;
$\dot{m}$	– mass flow rate, kg/s;
M	– Mach number, M = w/a;
Nu	– Nusselt number, $Nu = \overline{\alpha}\, d_e / \lambda_L$;
h	– specific enthalpy, kJ/kg;
h_{fg}	– heat of condensation, kJ/kg;
p	– pressure, Pa;
Re	– Reynolds number, $Re = w \rho\, d_e / \mu$;
s	– thermal capacitance, kJ/kg K;
S	– slip coefficient, $S = w_{V1} / w_{L1}$;
T	– temperature, K;
$\Delta \overline{T}_{MC}$	– logarithmic mean temperature in mixing chamber, K;
U	– injection coefficient, $U = m_{L0}/m_{V0}$, ;
w	– flow velocity, m/s;
x	– steam quality, $x = \dot{m}_V / (\dot{m}_V + \dot{m}_L)$;
$\overline{\alpha}$	– mean heat transfer coefficient for mixing chamber, W/m^2 K;
δ	– water nozzle gap, mm;
λ	– thermal conductivity coefficient, W/(m*K);
η	– injector efficiency;
μ	– dynamic viscosity, kg/(m*s);
ξ	– parameter of physical properties, $\xi = (\rho_{L1} / \rho_{V1})^{0.5} (\mu_{V1} / \mu_{L1})^{0.1}$
ρ	– density, m^3/kg.

lower indices:

0	– at injector inlet;
1	– at inlet to mixing chamber;
2	– at mixing chamber throat;
3	– behind shock wave;
4	– at outlet from injector;
b	– (stands for) exergy;
e	– equivalent;
max	– maximum;
L	– (stands for) liquid (water);
p	– (stands for) pump;
SN	– (stands for) steam nozzle;
V	– (stands for) vapour (of water).