The Efficiency Rate of a Steam-Water Injector

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Abstract: This paper analyses the influence of relevant parameters on the efficiency rate of a supersonic injector and its parts. Forced condensation in the injector is achieved by the mixing of cold water and steam with the goal of getting higher pressures of hot water (6-12 bar) with an outlet temperature of 70-80 °C to enable distant transportation of hot water and heating. Although the energy potential of pressure is not significant compared to inlet energy of steam, it is of importance since it represents the potential to realize external work. The complex flow process through the mixing chamber, the most important part of the injector, is presented with a diagram of forced condensation energy change, flow and geometry changes of the mixing chamber and the distribution of the relevant forces on the borders of the mixing chamber. It is shown that higher efficiency rates are achieved if condensation is speeded up by the reduced mixing of cold water and steam. Analyses are done according to analytical laws and experimental investigations of a steam-water injector prototype, with the data enclosed. The efficiency rates of the mixing chamber ranges from 60 to 85%, depending upon the following: the injection coefficient, the inlet pressures and the temperatures of the cold water and steam (and the desired outlet hot water temperature and pressure). Investigations are mostly directed to the particular needs of energy and process devices and the significance of the Mach number was not especially emphasized in them.

Keywords: steam-water injector; efficiency rate; supersonic steam-water injector; forced condensation

1 Introduction

The main purpose of the considered steam-water injector is to substitute the pump and heat exchanger, which is a great challenge for numerous applications [1, 2]. The relatively cheap acquisition of a steam boiler with low-demanded performances eliminates the need for electric energy supply and makes possible the autonomy of certain process. The remaining important question is if it is possible to get low outlet temperature and relatively high pressure (6-12 bar, $\dot{m}_w/\dot{m}_s < 5$) with a steam-water injector.

Due to very high velocities of steam (v_s) at the outlet of Laval nozzle of about 600-800 m/s, velocities of mixture through mixing chamber ($v_{mix\ mc}$) are also high, so the whole flow process of mixing and condensation happens very fast. It is of importance that the transition from supersonic into subsonic flow is realized in vicinity of the throat of mixing chamber, which, as a matter of fact, occurs no matter if all steam is condensed or water is completely mixed with condensate. This fact, experimentally confirmed, indicates that the homogeneity of flow field and uniform mixing are not necessary in order to achieve the transition from supersonic into subsonic flow at the end of mixing chamber or near to it.

Sound velocity (c), i.e. the Mach number (M), is the main criterion for such a complex and quick energy flow process through mixing chamber, which is in accordance with dynamic equation of compressible flow without losses in which balance of flow is reached by equilibrium of only inertial and elastic (compressible) forces. Isentropic-polytrophic changes of flow parameters in the transition zone ($M \approx 1$), shown in diagram in Fig. 1 [3, 4], are valid also for the flow in vicinity of the mixing chamber throat. Smooth transition - without shocks - from supersonic into subsonic flow is achieved also during experimental investigations.

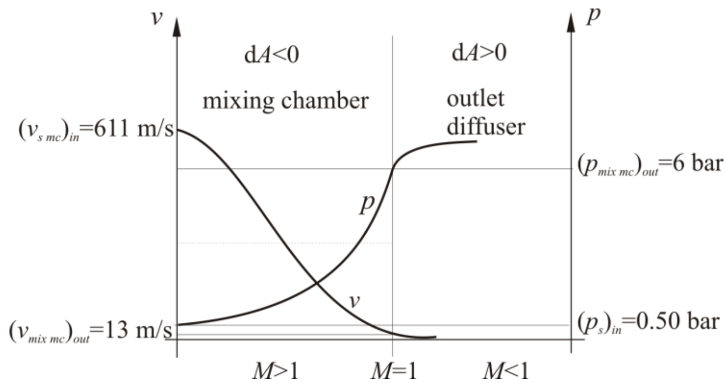

Figure 1

Steam-water mixture polytrophic changes of pressure (p) and velocity (v) through mixing chamber for supersonic transition as a function of Mach number (M) and cross-section change dA

The energy diagram, which is very important for the physical representation and analysis of flow, can be gained indirectly in this case, thanks to the energy diagram of forced condensation. The modified Mollier's diagram of forced condensation, connected with longitudinal geometry change (change of the cross section) of the mixing chamber, accurately presents energy changes along mixing chamber (p, t, h).

The application of reliable momentum equation, which is valid without any limitations for flow process in the mixing chamber, gives data about the pressure change in it, depending on the inlet and outlet parameters.

The efficiency rate of the steam-water injector η_{inj} is presented with:

$$\eta_{inj} = \eta_{L+cw nozz} \cdot \eta_{mc} \cdot \eta_d \quad (1)$$

where

$\eta_{L+cw\ nozz}$ – is the efficiency rate of the Laval nozzle and the cold water nozzle

η_{mc} – is the efficiency rate of the mixing chamber

η_d – is the efficiency rate of the outlet diffuser.

The efficiency rate of the Laval nozzle for steam and outlet diffuser for hot water are well known. The efficiency rate of the mixing chamber is determined according to the known energy characteristics of: cold water at disposal $(p, t)_{cw}$, steam at disposal $(p, t)_s$ and the desired characteristics of the hot water (mixture) in the throat of mixing chamber $(p, t)_{mix}$.

2 Results and Disussion

2.1 Flow Model in Mixing Chamber

The mixing chamber, which is 10 cm long, is fictively divided into 10 sections, each 1 cm long. The inlet cross section is 4.2 cm in diameter, and the outlet is 1 cm. The changeable orifice inlet of cold water can be regulated in a range from 0.5 to 3 mm (Fig. 2).

The assumption is that forced condensation in the mixing chamber forms following the flow model (Figs. 3, 4 and 5):

- Cold water gradually mixes with steam and condenses, and during this process, it is constantly in contact with mixing chamber walls, with no steam present. The velocity of water part, which mixes (v_{wm}) and is heated, increases until it has reached the velocity of steam and condensate. From that point forward, their velocities are the same (v_{mix}). The rest of water (v_{wrest}), which is located along chamber walls, accelerates from inlet velocity ($(v_w)_{mc\ in}$) gradually to mixture velocity at the outlet of mixing chamber $(v_{mix})_{mc\ out}$. The temperature of both these parts of water is almost the same and increases almost linearly through whole chamber.
- Steam mixes with certain parts of the water for mixing and condensates uniformly along the mixing chamber. Steam fulfills a larger part of mixing chamber. It is present in the throat and after mixing chamber throat (referent point). Steam is dominant at about 80 mm length from inlet section of chamber. In this region are valid compressible flow laws with

parameters of wet steam shown in Fig. 5. The velocity of the steam (v_s) suddenly decreases from inlet velocity from Laval nozzle ($(v_L)_{in}$=611 m/s) to outlet velocity from mixing chamber ($(v_{mc})_{out}$=13 m/s). In the vicinity of the mixing chamber throat, the velocities of steam (v_s) and mixture (v_{mix}) are equal.

- The velocity profiles in certain sections of mixing chamber show a tendency to equalize velocities of all components in the vicinity of the mixing chamber throat.

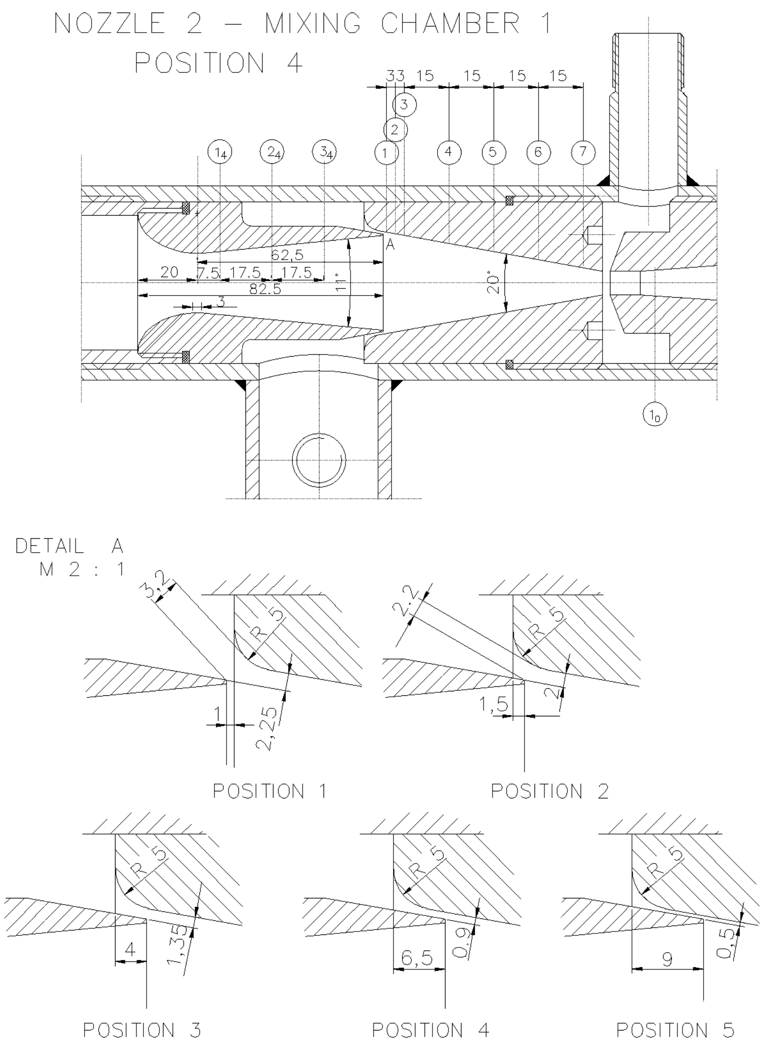

Figure 2
Configuration of cold water nozzles orifices

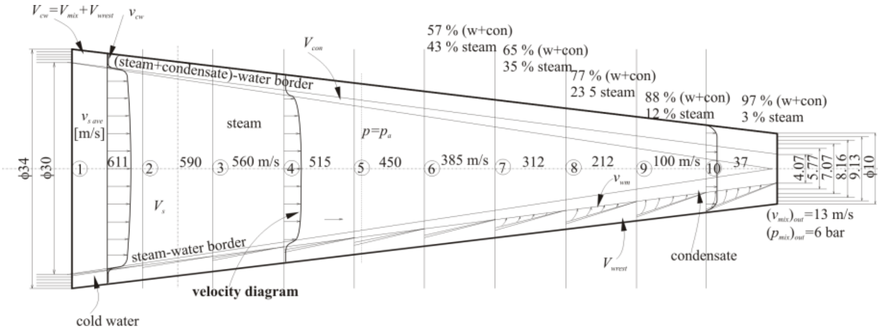

Figure 3

Flow model of forced condensation in mixing chamber

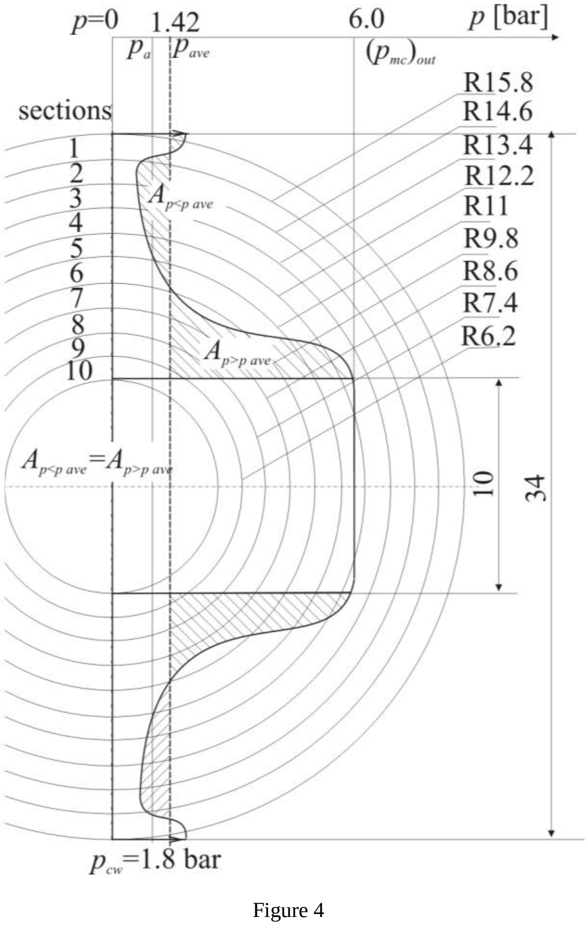

Figure 4

Pressure distribution along mixing chamber walls

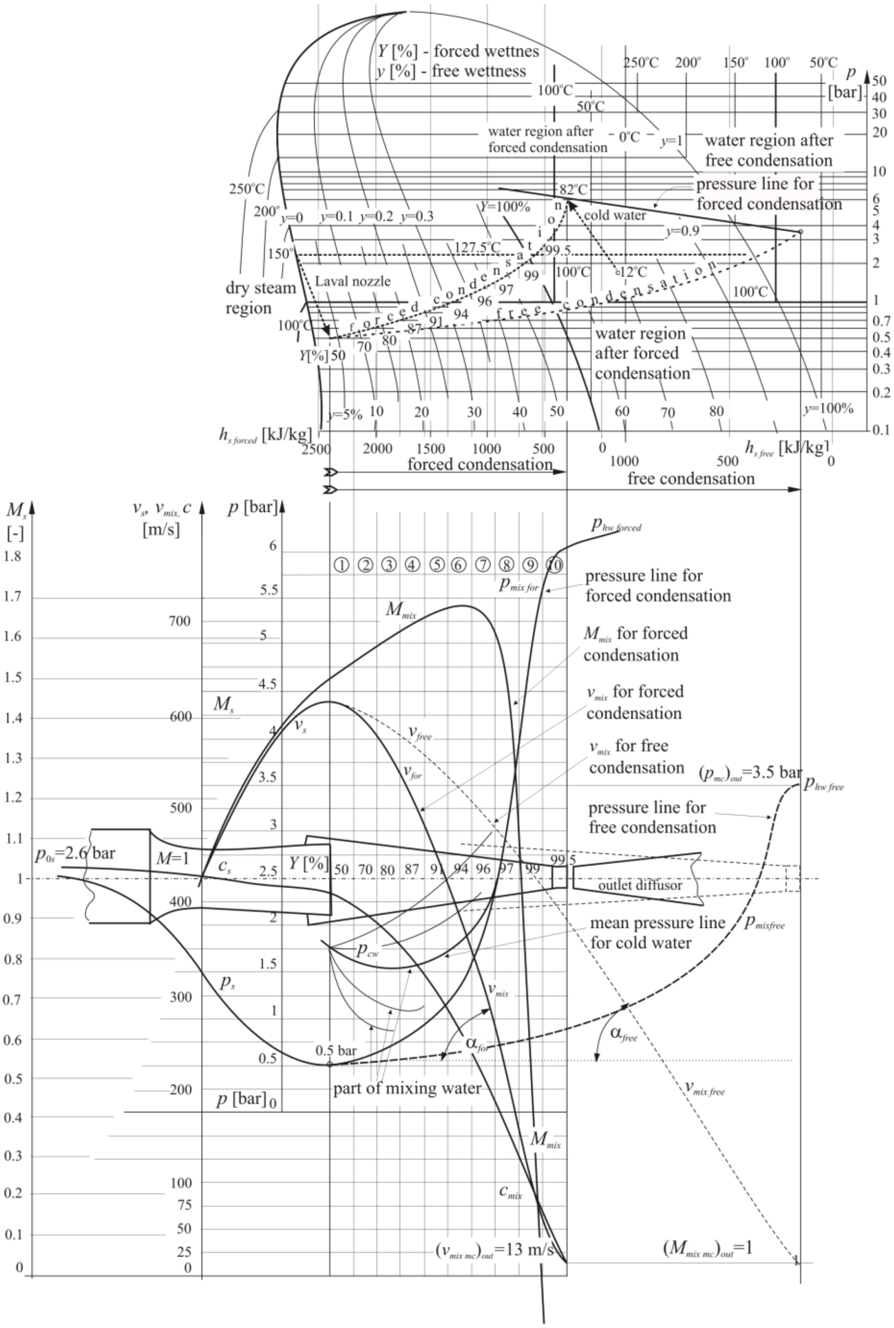

Figure 5

Diagram of Mach number, pressure, velocity, temperature and density for forced condensation

The condensation core's forming rate is greater under forced condensation according to Deich and Filipov [5]. The condensate has significantly lower velocity compared to surrounding steam due to the density difference ($\rho_w/\rho_{s\,ave}$=500).

The pressure diagram of steam and mixture contains points of inlet pressure of steam, the average pressure of mixture (gained from momentum equation) and the outlet pressure of the mixture – hot water ($(p_{mix})_{out}$) (Fig. 1). An abrupt mixture pressure rise happens in the immediate vicinity of the mixing chamber throat, i.e. where M=1, according to diagrams given in Fig. 1. and Fig. 5. Experiments [3, 4] confirm that the pressure change through mixing chamber is stationary, stable and smooth, without visible and sensed shocks and disturbances. The pressure diagram (Fig. 4) shows the pressure at the inner side of mixing chamber wall. Each circle in Fig. 4 corresponds to a certain section (Fig. 3) for which pressure is demanded.

2.1.1 Losses in Mixing Chamber

The efficiency rate of the mixing chamber depends on different kinds of losses:

- loss due to friction on the chamber walls;
- loss due to local drags, oblique and normal shock waves of different origin, and intermolecular reactions (friction of water, steam and condensate);
- local losses of cold water and steam at the entrance into the mixing chamber and mixture at the outlet of the mixing chamber.

Friction losses on the inner side of conical wrap of the mixing chamber are manifested as a pressure drop which is present in whole flow cross section [6]. This downstream pressure drop diminishes outlet pressure in the mixing chamber throat. Water heat increases due to friction cannot be transformed into pressure energy because of its low energy potential. Losses due to friction on the chamber walls are not small compared to the pressure rise along the chamber. If the average velocity of the rest of the water, near to conical wrap, is v_{wrest}=10 m/s, the pressure drop through mixing chamber is about Δp=3000 Pa, which is 0.5 % compared to the outlet pressure of 6 bar. If the average velocity is 20 m/s, the pressure drop through the mixing chamber due to friction would be 12,000 Pa, which is, compared to outlet pressure of 6 bar, 2%. Because of this, it is important to disable greater velocities of water layer, which is desired to be in constant connection with walls of mixing chamber.

The internal energy of steam consists of intermolecular and intramolecular interactions, as well as the energy of random molecular movements, which depends on temperature. If there are no chemical reactions in the fluid, all changes in internal energy are due to thermal changes in the fluid, i.e. due to changes in the kinetic energy of molecular movement and the molecular interaction. Since in the larger part of mixing chamber there is steam, which can be considered as ideal gas in that part, its internal energy depends only on the starting and end temperatures.

Losses due to local changes of stream flow, as well as the occurrence of waves of different origin, can be considered as irreversible, compared to the pressure drop decrease which appears no matter where the local resistance takes place. Partial heat increase, which occurs downstream from these disturbances, cannot diminish the downstream pressure drop either. Normal stationary shock waves cannot be transmitted or occur in mixing chamber because of the suppressed effect of the inlet cold water.

Losses due to friction and mixing of the steam, water and condensate are hard to estimate, but they can be diminished if continuity in changes of temperature, steam and velocity of all mixture components are reached. Continuity means uniform:

- pressure rise of: steam, condensate, water and mixture,
- velocity drop of: steam, condensate and mixture,
- water temperature increase.

Continuity in velocity distribution through a cross section of the mixing chamber along its whole length, along with maintaining steady water film along chamber walls, probably presents optimal flow model.

Mechanical efficiency rate of mixing chamber of water-water injector, which takes into account the friction of the mixing water, as well as the friction of the water and the walls of the mixing chamber, according to Menegay [7], is 25% for a relative chamber wall roughness of 0.03. If this value is taken for the steam-water injector, with assumption that condensate is a drive fluid and cold water is sucked one, there would appear difference in their kinetic energies at the entrance and exit of mixing chamber of about 2 kW. The efficiency rate of the mixing chamber would change by about 0.5%.

Muffling of the supersonic flow, according to its nature, leads to a muffling of flow anomalies and disturbances. It is known in the transportation of mixtures that the flow of the mixture calms the whole one-phase fluid flow, which indicates that both flows are compatible and collectively characteristics of joint model provides a stable flow.

The reduction of losses in the muffled stream of supersonic mixture increases flow stability, which can be maintained in moderately non-stationary conditions. If increased disturbances result in the impossibility of maintaining a stable operation, these disturbances are source of losses which influence the energy efficiency of the process.

The estimation of pressure drop due to other mentioned resistances demands much more information of an experimental nature.

2.2 Supersonic Flow through Steam-Water Injector

The graphically presented distributions of the Mach number (M), pressure (p), and the velocities of steam and mixture (v_s, v_{mix}) are products of laboratory research [3, 4]. Credible pressure lines are those through Laval nozzle and outlet diffuser. The pressure gradient through mixing chamber is positive, with a steep inflection in front of the throat of the mixing chamber. The main results of the supersonic flow through the steam-water injector are presented in Fig. 5.

The discontinuity of pressure due to appearance of shockwaves at the exit of Laval nozzle, as well as in the mixing chamber, is not registered during the proper working of the device. Stable work requires a smooth pressure line throughout the injector, without shockwaves in any part of injector.

The relation between pressure and Mach number could be recognized in already existing law of compressible flow:

$$\frac{p_0}{p} = \left(1 + \frac{\kappa - 1}{2} M^2\right)^{\frac{\kappa}{\kappa - 1}} \qquad (2)$$

where coefficient (κ-1)/2 and exponent κ/(κ-1) should be exchanged with variable and independent coefficients $(\kappa-1)/2=m$ and $\kappa/(\kappa-1)=n$. It is assumed that in the throat of the mixing chamber prevails almost total pressure, due to the low velocities in the throat.

Deich and Filipov [5] gave the values for κ depending on wetness (y). If an assumption is made that the wetness during forced condensation (Y) is equivalent to the wetness during free condensation (y), according to eq. (2) total pressures may be calculated in the throat of mixing chamber.

Calculated values of mixing chamber outlet pressure from eq. (2) partly can be identified with polytrophic change in the mixing chamber.

The representative parameters of the conducted experimental research are:

- mass flow rate: steam $\dot{m}_s = 0.17$ kg/s, cold water $\dot{m}_w = 0.85$ kg/s,
- dryness rate at: entrance into injector x=1, exit from Laval nozzle x=0.95,
- cross section: in front of injector ϕ40, exit of mixing chamber ϕ10, exit of diffuser ϕ25,
- velocities: cold water at the entrance of mixing chamber 4.1 m/s, steam at the entrance of mixing chamber 611 m/s, mixture in the throat of mixing chamber 13 m/s, mixture at the exit of diffuser 2.1 m/s,
- pressure of: steam in front of Laval nozzle $p_{0s} = 2.6$ bar, cold water at the entrance into mixing chamber p_{cw}=1.8 bar, mixture (hot water) at the exit of mixing chamber p_{mix}=6.0 bar, hot water at the exit of diffuser 6.66 bar,

- temperature of: cold water t_{cw}=12 °C, steam at the entrance of injector t_s=127.5 °C, mixture at the exit of injector t_{mix}=82 °C,
- enthalpy of: steam (total) in front of injector h_0=2718 J/kgK, at the entrance of mixing chamber h=2530 kJ/kgK (0.5 bar, 0.33 kg/m^3).

The maximal pressure of hot water is achieved by a rise in outlet downstream pressure (back pressure), which represents a hot water pipeline pressure drop. The steam-water injector acts like a volumetric pump since the flow rate is not dependent upon the variation in outlet pressures.

During experimental research were obtained:

- stable flow for:

  - l_{mc}=100 mm and d_0=10 mm

- unstable flow with shocks and vibrations and flow rate decrease for:

  - l_{mc}=80 mm (for d_0=10 mm and d_0=8 mm)

  - l_{mc}=100 mm and d_0=8 mm.

In Fig. 5 is also given the comparative diagram for forced and free condensation (p, v).

2.3 Effect of Energy Transformation in Mixing Chamber

2.3.1 Water Velocity Distribution in the Mixing Chamber

The velocity distribution of the main water flows in the mixing chamber could substantially contribute to determining the complex process properties: mixing, condensation, heat transfer and losses.

Analyses of the flow through the mixing chamber, which are based on present knowledge about the process, show that basic losses through the mixing chamber are connected to losses due to inter-phases mixing and friction, and to the friction of the water (and maybe steam) and the walls of chamber.

Local losses at the entrance of the mixing chamber form a useful initial weak oblique shock of pressure, which is changed into a continual pressure wave and culminates at the entrance of the sonic flow region, in front of the chamber throat. The formation of condensation sources probably influences the pressure wave shape by changing frequencies, but it is neglected in this case.

It is reasonable to expect that the water heats, mixes with steam and condensates within optimal energy conditions. It seems that for the rest of water the following stands:

- heating is homogenous from inlet (12 °C) to outlet (82 °C) temperature, which agrees with experimental investigation of Malibashev [9];
- velocity increases from inlet (4.1 m/s) to outlet velocity (13 m/s);
- pressure grows from inlet (p_{cw}=1.8 bar, p_s=0.5 bar) to outlet (p_{mix}=6.0 bar).

2.3.2 Energy Transformation

The energy equation, which comprises kinetic, pressure and heat energy, is clearly presented through injector efficiency analyses.

The efficiency rate of the injector is defined as the ratio between the energy of outlet hot water and the sum of the energies of the steam and cold water at the entrance of the mixing chamber. It is accepted that all energies are useful: heat energy (q [J/kg]), kinetic energy ($v^2/2$ [m^2/s^2]) and pressure energy (p/ρ [m^2/s^2]). Kinetic and pressure energy have the same dimensions as heat energy (J/kg or m^2/s^2).

The sum of heat and pressure energies is enthalpy (h). The basic inlet energy is gained from steam, which is being condensed. Heat energy of about 2200 kJ/kg is mostly at disposal, and it is latent heat released from the steam during condensation.

All three kinds of energies are present during the forced mixing of cold water and steam through the injector, but with huge differences in energy content. Heat and pressure energy of steam are present in tens and hundreds of kJ, water pressure energy from 0.1 to 1 kJ/kg, and kinetic energy of steam during supersonic flow reaches several hundreds of kJ, while kinetic energy of water during standard flow does not exceed 0.05 kJ.

This means that in order to reach relatively high pressure of water, high heat energy of steam is not essential, since there is 2200 kJ/kg. (E.g. in order to increase pressure of 1 kg of water from 0 to 50 bar, 5 kJ is needed.) Water heating and cooling needs about 4.2 kJ/kgK of heat energy, which is a main energy requirement in comparison with medium kinetic and pressure energies.

Since flow processes are conducted continuously, each kind of energy (q, p/ρ, $v^2/2$, [kJ/kg]) is multiplied with mass flow rate [kg/s] and become power in kW. Therefore, energy efficiency is presented in the relation between exit and inlet powers (kW_{out}/kW_{in}).

The injector is designed according to its application. The investigated steam-water injector is used for heating, distant transportation and warm water distribution for different users. These applications require:

- outlet water which is not too hot (70-80 °C) due to diminishing heat losses along the way;

- outlet water pressure which is high enough to conquer hydraulic losses (6-8 bar);
- the inlet water temperature should be about 70 °C, lower than steam condensation temperature, in order to successfully conduct condensation process;
- the inlet water pressure depends on characteristics of the water sources. It can be atmospheric pressure, but if it is available, water under pressure can be used as well; in that way, the efficiency rate is increased and a higher pressure of warm water is more likely;
- inlet steam pressure can be low, which allows for using less quality steam (p_m=1,5-2 bar).

2.4 Efficiency Rate Definitions

Efficiency rates can be defined in different ways, with respect to the basic postulate that the efficiency rate is a relation between useful (or needed) and engaged. The terms "useful" and "needed" could be applied to numerous variables: energy (power, work), bulk (mass, volume), flow surfaces, pressures or their differences (inlet, outlet), concentrations, granulometric fractions, etc.

The definitions which describe process quality or quality of one of its segment are: universal efficiency rate, thermodynamic efficiency rate, kinetic energy efficiency rate, efficiency rate of mixing, etc.

Natural - free condensation of steam (static conditions) is conducted under constant pressure and almost constant temperature in the whole two-phase area. Numerous pressure diagrams through cylindrical, and sometimes conical mixing chambers, accept this law for flow processes too.

Water velocity at the inlet section of the mixing chamber $(v_{cw})_{mc\ in}$ and mixture velocity in the mixing chamber's throat $(v_{mix})_{mc\ out}$ are assumed according to the energy diagram.

Since the difference of latent heat of evaporation $\Delta\lambda$, for pressures ranges from 1 to 10 bar is only 3.5%, it can be considered that pressure and temperature do not influence the heat exchange between water and steam in the two-phase region.

The efficiency rate of the mixing chamber η_{mc} is:

$$\eta_{mc} = \frac{\left(h_{mix} + \frac{1}{2}v_{mix}^2\right)_{mc\ out} \dot{m}_{mix}}{\left[\left(h_s + \frac{1}{2}v_s^2\right)_{L\ out} \dot{m}_s + \left(h_{cw} + \frac{1}{2}v_{cw}^2\right)_{mc\ in} \dot{m}_{cw}\right]} \tag{3}$$

The efficiency rate which is used for heat pumps is validated with one of the following pressure increase efficiency rates ($(\eta_{cw})_p$, $(\eta_s)_p$):

$$\left(\eta_{cw}\right)_p = \frac{p_{mix}}{p_{cw}};\ \left(\eta_s\right)_p = \frac{p_{mix}}{p_s} \tag{4}$$

The commonly used efficiency rate η_p for small injection coefficients $u_{cw\ p}<3$ ($u_{cw\,p} = \dot{m}_{cw}/\dot{m}_s$) is given as following:

$$\eta_p = \frac{p_{mix} - p_s}{p_s - p_{cw}} \tag{5}$$

where:

p_{mix}, p_{cw} and p_s are the pressures of mixture, water and steam [Pa].

For higher injection coefficients the next relation is valid:

$$\eta_p = \frac{p_{mix} - p_{cw}}{p_s - p_{cw}} \tag{6}$$

For both relations, the denominator is the same and can be considered as engaged pressure.

2.5 Efficiency Rate of Laval Nozzle η_l

In the ideal case, the flow through the nozzle and diffuser is isentropic. But in the actual case, friction exists and affects in following ways:

- reduces the enthalpy drop and the final velocity of steam
- increases the final dryness fraction and specific volume of the fluid
- decreases the mass flow rate.

The efficiency of the nozzle depends upon:

- the material it is made of and its smoothness,
- the size, shape and angle of the nozzle divergence,
- the nature of fluid flowing and its state,
- the fluid velocity and the turbulence in nozzle flow.

The efficiency rate of the Laval nozzle for superheated and wet steam flow is determined by many experimental results with satisfying accuracy. So far, the transformation of friction through the Laval nozzle into thermal energy is not explained enough. The nature of energy loss in the case of a normal shockwave in the diffuser of the Laval nozzle can be explained. The inlet parameters of the steam in the mixing chamber can be determined if the efficiency rate of the Laval nozzle is known.

The efficiency rate of the Laval nozzle is based on knowing the investigated features of the convergent nozzle, whose efficiency rate is dominant in the equation for η_L.

The complete diagram of the efficiency rate of Laval nozzle η_L is shown in Fig. 6 with three curves: real nozzle (with friction) η_L, theoretical η_{Lth} and with post expansion η_{Le}.

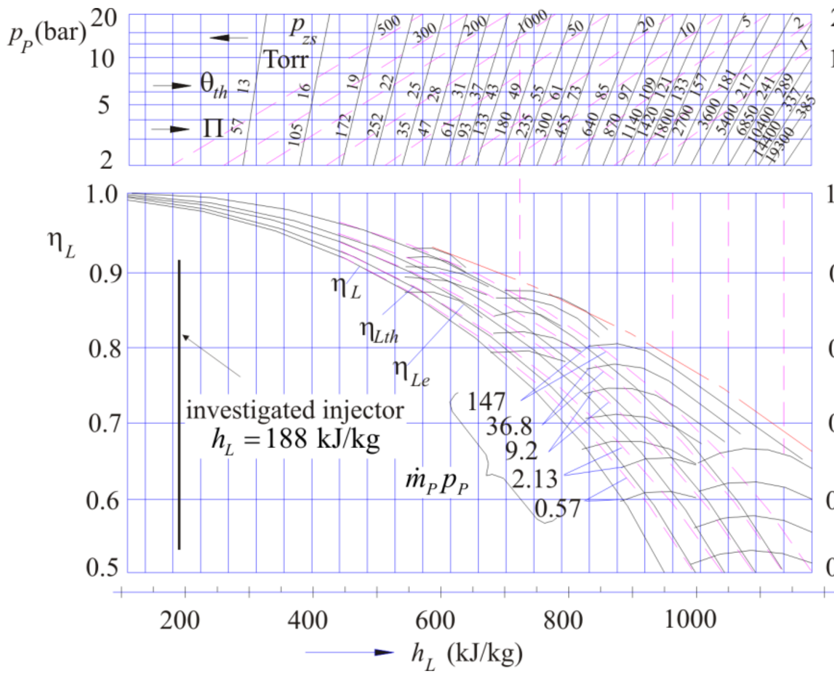

Figure 6

Efficiency rates of Laval nozzles for smooth surfaces and shock less processes ($p_S\,\dot{m}_S\,[\mathrm{bar}\cdot\mathrm{kg/s}]$; $\Delta h_L = h_{Lin} - h_{Lout}$) according to Hess [4]

The efficiency rates' theoretical curves of the Laval nozzle with post expansion $\eta_{Le}=f(h_L)$ are parallel and with one maximal value. Only experiments can show the magnitude of post expansion in order to achieve the maximal efficiency rate for the given Laval nozzle.

Each of the curves is given for the certain value of the Reynolds number or characteristics of flow $\dot{m}_s p_s$. Divergent section is with central angle of 10-20°. In this range of angles, the change in efficiency rate is within tolerable limits. During the flow of steam through the Laval nozzle, the boundary layer is turbulent.

In the representative case, where $\dot{m}_s = 0.17$ kg/s, p_{0s} =2.6 bar, h_0=2718 kJ/kg, $(h_0)_{L\ out}$=2530 kJ/kg, Δh_L=188 kJ/kg and $\dot{m}_s p_{0s} = 0{,}442$;

- the efficiency rate of the Laval for all kind of energies is η_L=0.98.
- the power loss through Laval nozzle is 0.17·188·0.02=0.64 kW.
- The steam power at the outlet of the Laval nozzle is 2718·0.17-0.64=461.42 kW, where
  - kinetic power is $((611^2/2)\cdot 0.17)/1000$=31.73 kW
  - enthalpy power (thermal and pressure power) is 2530.0.17=430.1 kW.

The outlet pressure should not be too low compared to the nozzle throat pressure, in order to eliminate the possibility of a shock wave forming in the supersonic nozzle section, which does not contribute to the proper operation of the injector. The outlet pressure from the Laval nozzle should be about 0.4-0.5 bar (absolute pressure), which allows for the usage of low-pressure steam in suction of cold water on atmospheric pressure.

2.6 Overall Efficiency Rate of Laval and Cold Water Nozzle

According to [10], the efficiency rate for designed cold water nozzle for lost kinetic energy (without thermal energy) is estimated as η_{cw}=0.90.

For outlet water velocity of a nozzle of 4.1 m/s:

- the intake kinetic power of the cold water is $(4.1^2/2)\cdot 0.85$=7.144 kW,
- the lost kinetic power through the cold water nozzle is 7.144·0.1=0.714 kW
- the kinetic power at the inlet of the mixing chamber is 7.14·0.90=6.43 kW
- the efficiency rate of the cold water nozzle, which refers to all kind of energies, is (42.738+7.14·0.9+0.153)/(42.738+7.144+0.153) =49.317/50.035=0.986.

The power loss through the Laval nozzle is 0.64 kW, and for the cold water nozzle it is 0.714 kW.

The total steam power at the entrance of the injector (462.06 kW), and cold water (50.035 kW) is 512.09 kW.

The total power of steam and cold water at the inlet of the mixing chamber with embraced losses is:

512.09-0.714-0.64=510.74 kW.

Overall efficiency rate of Laval and cold water nozzle is

$\eta_{L+cwnozz}$=510.74/512.09=0.99.

2.7 Efficiency Rate of Outlet Diffusor

The efficiency rate of the outlet diffuser is discussed according to several assumptions:

- The diffuser is filled with an homogeneous stream of hot water without steam bubbles;
- The heat losses are neglected;
- Condensation is completely finished in the throat of mixing chamber.

With assumption that velocity profile uniformly decreases through diffuser cross section, the efficiency rate (without thermal energy) – cold η_{dc} is:

$$\eta_{dc} = \frac{\left(p_{out} - p_{in}\right)_d}{\frac{1}{2}\rho\left(v_{in}^2 - v_{out}^2\right)_d} = \frac{\left(p_{out} - p_{in}\right)_d}{\frac{1}{2}\rho\left(v_{in}\right)_d^2\left[1 - \left(\frac{A_{in}}{A_{out}}\right)_d^2\right]} \tag{7}$$

The outlet diffuser is designed with diameters ratio 1:3 (10:30) and half angle α=6°; for which is, according to [10] efficiency rate η_{dc}=0.95. The efficiency rate of the outlet diffuser of 0.95 refers to the lost kinetic energy of 5%. For η_{dc}=0.95 and an inlet velocity of 13 m/s and an outlet velocity of 2.1 m/s, the lost power is $P_{lost} = \left(v_{in}^2 - v_{out}^2\right)\dot{m}_{mix}\left(1-\eta_{dc}\right)/2 = 0{,}0042 \text{ kW}$. From this it follows that the efficiency rate of outlet diffuser is η_d=1.

According to equation (7), which is in accordance with experimental results, the absolute pressure of the hot water at the outlet of the diffuser is:

$(p_d)_{out}$=6.66 bar

2.8 Efficiency Rates of Mixing Chamber (η_{Mc}) and Injector (η_{Inj})

According to known characteristics of hot water at the outlet of the diffuser, as well as the diffuser's efficiency rate, the parameters in the throat of the mixing chamber are: $p_{mc\ out}$=6.0 bar, $v_{mc\ out}$=13 m/s, $t_{mc\ out}$=82 °C, and $\dot{m}_{mc\,out} = 1.02$ kg/s. Steam is completely condensed at the outlet of the mixing chamber.

Hot water power at the outlet of the mixing chamber is:

(82·4.19+132·0.5+0.6)·1.02 = 437.25 kW.

Power at the inlet of the mixing chamber is 510.74 kW.

The efficiency rate of the mixing chamber is

$\eta_{mc} = 437.25/510.74 = 0.856$.

The efficiency of the whole injector is given by eq. (1):

$\eta_{inj} = \eta_{L+cw\,nozz} \cdot \eta_{mc} \cdot \eta_{d} = 0.99 \cdot 0.856 \cdot 1.0 = 0.847$.

The small difference in the efficiency rates of the mixing chamber and the whole injector is a result of the small powers that are spent in: the Laval nozzle, the cold water nozzle and the outlet diffuser. The efficiency rate of the Laval nozzle is the relation between the total energies at its outlet and inlet section, while in the case of the cold water nozzle and the outlet diffuser, losses are connected only to the change of kinetic energy, which is several times smaller than total power that is being transmitted through them.

The mixing chamber efficiency rate cannot be determined without knowing the flow-phases and energy transformation through them. Without any doubt, part of the pressure increase is fulfilled due to the kinetic energy of the steam which enters into mixing chamber.

The basic question without proper answer is whether it is possible to control and manage the process of complex changes through the mixing chamber or whether the process depends, to the greatest extent, on the thermo dynamical nature of the free condensation of steam. No matter the answer to this unanswered question, the efficiency rate of the mixing chamber can be determined for some assumed cases. The flow through injector is characterized by complex processes of steam, condensate, cold and hot water inter reactions.

Conclusions

In addition to the known and frequent transformation of thermal into kinetic energy, the transformation of steam thermal energy into pressure energy of a mixture in a supersonic stream, which is possible by condensation, using latent heat, was also confirmed.

During experimental investigation, the stable work of a steam-water injector (without pulsations and shocks) was established. In order to reach stable work, it is necessary that the flow process through all parts of the injector is in balance. This means: The sound velocity of steam in the throat of Laval nozzle and supersonic velocity are reached at its outlet; and supersonic compressible flow prevails through the mixing chamber and subsonic flow of hot water prevails through the outlet diffuser of steam-water injector.

The flow through the Laval nozzle depends on the geometry of the nozzle and the inlet state of the steam. The parameters at the outlet section of the Laval nozzle are dictated by the steam. The inlet cold water pressure can vary depending on the water source.

During the proper operation of the steam-water injector, water does not evaporate in the mixing chamber; otherwise it would be noticed by unsteady effects. The steady range of the steam-water injector operation depends on a limited coefficient of injection.

The inlet section of the mixing chamber is filled with cold water (orifice with small thickness), i.e. mixture of steam and cold water (Y=0.5). In the greater part of mixing chamber is a two phased region, which is completely condensed in the vicinity of the mixing chamber throat. Supersonic flow exists in the whole mixing chamber; at the beginning, the average velocities of steam are high, and in the vicinity of the throat, the velocity of the mixture are considerably lower. At the entrance of mixing chamber, the Mach number is greater than unity, M>1, because of the expansion of the steam through the Laval nozzle, and in the environment of the throat $M\approx1$, because of very low sound velocities in the very wet steam.

In the throat of the mixing chamber, changes in the flow parameters without shocks are with high gradients and pretty independent of the size and changes of the Mach number through the mixing chamber.

In this paper is presented the recommended velocity distribution of the active and inert components of cold water. It is impossible to determine losses in the mixing chamber without knowing the velocity distribution of the water that is actively and directly mixed with steam (v_{wmix}) and of the water which flows further on and continuously, partially and actively mixes with steam (v_{wrest}). The efficiency rate of the injector device shows that major losses happen in the mixing chamber. The losses are the immediate consequences of considerable friction between the steam, condensate, and active and rest (inert) water. It seems that the mixing chamber walls cause the most significant loss, which means that the better velocity profile is the one with higher gradient in the zone of intensive steam and water mixing.

Forced condensation, compared to free condensation, increases the mixing chamber outlet pressure of hot water. The supersonic mixing chamber is actually a diffuser through which pressure rises downstream. Pressure rise during supersonic flow has the same nature as flow through diffuser in subsonic flow.

The average pressure in the mixing chamber, gained from momentum equation, enables a more accurate determination of the pressure distribution through the mixing chamber. In this way the determined average pressure agrees with the actual pressure.

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Appendix

Nomenclature

A	[cm^2]	cross section area
c	[m/s]	sound velocity
d	[m]	diameter
h	[J/kg]	enthalpy
l	[m]	length

M	[-]	Mach number
m	[-]	coefficient
$\dot{m}$	[kg/s]	mass flow rate
n	[-]	coefficient
P	[W]	power
p	[Pa]	pressure
Q	[m^3/s]	flow rate
q	[J/kg]	heat energy
T	[K]	temperature
t	[°C]	temperature
u	[-]	injection coefficient
V	[m^3]	volume
v	[m/s]	velocity
X	[%]	dryness during forced condensation
x	[%]	dryness during free condensation
Y	[%]	wetness during forced condensation
y	[%]	wetness during free condensation
α	[°]	half angle of diffuser
η	[-]	efficiency rate
κ	[-]	isentropic exponent
λ	[J/kg]	latent heat
ρ	[kg/m^3]	density

Subscripts

0	total
a	atmospheric
ave	average
c	cold
con	condensed
cw	cold water
d	diffuser
hw	hot water
i	section number (*i*=1,2,...10)
in	inlet
inj	injector
L	Laval nozzle
Le	Laval nozzle with post expansion
l	losses
lost	lost
m	manometric,
mc	mixing chamber
mix	mixture
nozz	nozzle
out	outlet
s	steam

th throat
w water
wm water for mixing
wrest the rest of water

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