Journal of Energy Technology

JET Volume 6 (2013), p.p. 31 - 48

Issue 2, May 2013

http://www.fe.um.si/en/jet.html

USING STEAM AS AN ALTERNATIVE MOTIVE FLUID IN THE EXISTING TURBINE EJECTOR SYSTEM OF THE LJUBLJANA DISTRICT HEATING PLANT

MOŽNOSTI NAPAJANJA OBSTOJEČEGA EJEKTORSKEGA SISTEMA TURBOAGREGATA V TOPLARNI LJUBLJANA Z ALTERNATIVNO POGONSKO PARO

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Keywords: analysis, ejector, heat flow, oscillation, reconstruction, reliability of production, pump system, motive steam, turbine condenser

Abstract

In this paper, we will examine the possibility of using steam as an alternative motive fluid in the existing turbine ejector system of the Ljubljana district heating plant. As alternative motive fluid, steam is of lesser quality and has constant pressure. The ejector pump system will be adjusted to new circumstances. The purpose of the present work is to rationalise and increase the reliability of ejector system operation at the lowest possible investment costs. A computer model of an ejector model will be designed using the measurements and analyses of the existing system. On the basis of the results obtained, the changes required for the reliable operation of the system using steam as an alternative fluid will be defined and the appropriate solutions will be indicated.

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Povzetek

V članku bomo preučili možnosti napajanja obstoječega ejektorskega sistema turboagregata v toplarni Ljubljana z alternativno pogonsko paro. Alternativna pogonska para je slabše kvalitete in konstantnega tlaka. Ejektorski črpalni sistem bomo prilagodili novim razmeram. Namen naloge je racionalizacija in povečanje zanesljivosti delovanja ejektorskega sistema s čim nižjimi investicijskim stroški. S pomočjo opravljenih meritev in matematične analize obstoječega sistema, bomo izdelali računalniški program modela ejektorja. S pomočjo pridobljenih rezultatov, bomo določili potrebne spremembe za zanesljivo delovanje alternativnega sistema z alternativno pogonsko paro in podali ustrezne rešitve.

1 INTRODUCTION

In view of the strategic requirement for a rational use of fuels and more reliable operation of ejector systems, we are forced to seek alternative ideas. Some unexpected failures have occurred in the turbine operation as a consequence of the ejector system malfunctioning. The system was designed to remove any non-condensable gases from the turbine condenser (air). As a result of the pumping, a stable operating vacuum is maintained in the turbine condenser system. Due to pressure oscillation of the motive steam of the ejector pump system supplied via the reducing valves from a boiler, disruptions occur in maintaining proper condenser pressure. The primary cause of motive steam pressure oscillation is poor manual steam pressure control, which changes following the change in the boiler steam pressure. A more suitable source of ejector motive steam has been proven to be the steam generated from the third turbine extraction. The extraction is controlled, and pressure oscillation disruptions should not cause any substantial obstacles. In addition to the constant pressure of the motive steam, another advantage of the new system is that the motive steam expands in the turbine, thus producing useful work. The ejector motive steam of the existing system is generated by damping the high pressure boiler steam without any work being produced.

2 ANALYSIS OF EXISTING MOTIVE STEAM AND ADEQUACY ASSESSMENT OF ALTERNATIVE MOTIVE STEAM

The existing ejector system is supplied with steam generated from a boiler (92 bar and 512°C), reduced to 14 bar by means of manual reducing valves prior to entering the ejector system (Fig. 3). As a result of pressure oscillation in the boiler and poor pressure control of the ejector motive steam, changes in the ejector motive steam occur, leading to fluctuations in the ejector system flow. The fluctuation of the flow rate of gases leads to a pressure rise in the turbine condenser, causing lower turbine efficiency. Fig. 1 illustrates the boiler steam pressure oscillation (for a period of five days). The variations of pressure range from 87 bar to 96 bar. The boiler steam pressure variation leads to the variation of ejector motive steam pressure from 9 bar to 20 bar. The flow rate of gases lowers at the ejector motive steam pressure of 9 bar, causing a pressure rise in the condenser in long-term operation and consequently turbine failure. At the ejector motive steam pressure of 20 bar, the steam consumption for the system operation unnecessarily increases, resulting in reduced pump system efficiency.

It has been proved that the alternative motive steam can be used, specifically that of the third regulated steam extraction as a result of the constant pressure. Fig. 2 shows a variation in relative pressure ranging from 7.83 bar to 7.88 bar, which means oscillation of absolute pressure from 8.83 bar to 8.88 bar, to be used in the analysis. It was established that the steam of the third steam extraction has a enormous advantage over the existing motive steam, i.e. lower operating pressure oscillation, uninterrupted constant availability of steam even in the event of a boiler failure, no additional need for steam reduction and consequently no need for reducing valves. The steam from the third extraction expands in the turbine, thus producing useful work. A drawback of the alternative motive steam is its poorer quality requiring a detailed analysis of the pump system.

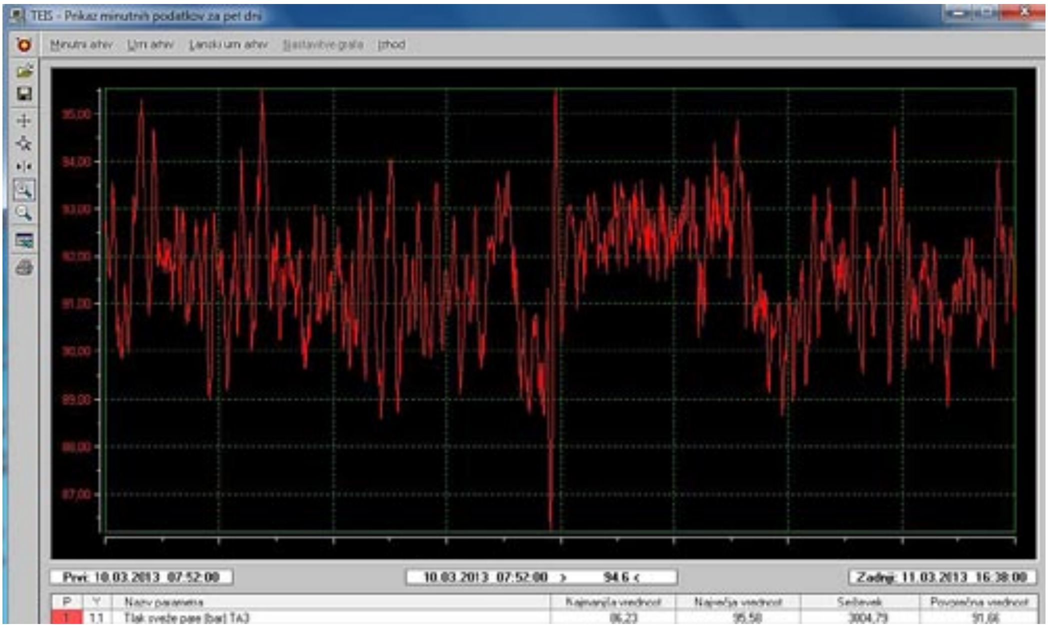

Figure 1: *Pressure oscillation of the steam generated in the boiler, TE-TOL [8]*

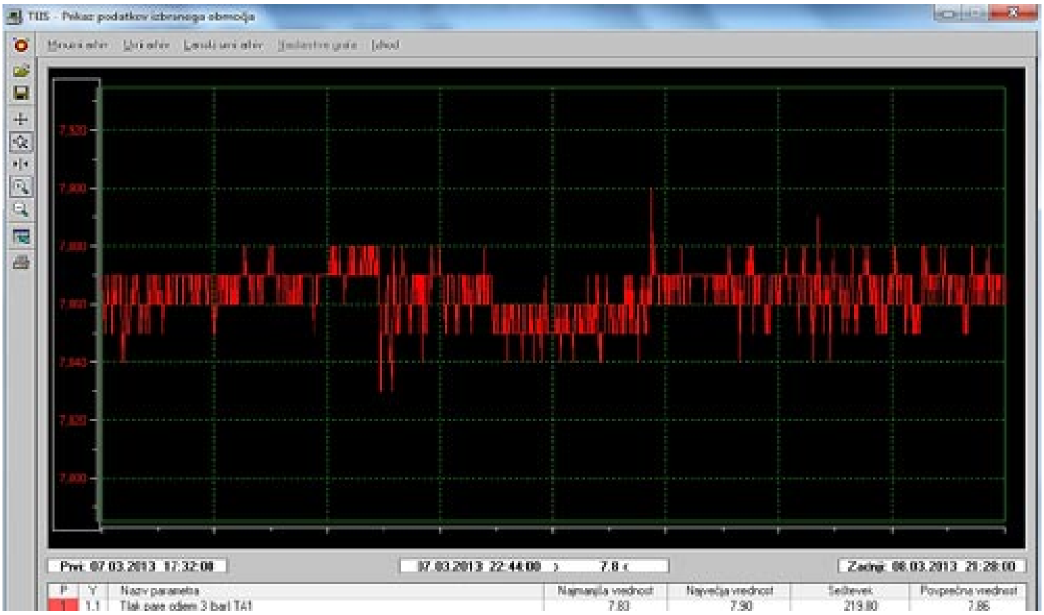

Figure 2: *Pressure oscillation of the third steam extraction, TE-TOL [8]*

3 DESCRIPTION OF EJECTOR SYSTEM OPERATION

Ejectors are devices designed to use the pressure energy of a working fluid for the transport of another working fluid, whereby no mechanical work is supplied or recovered. The working fluid may be liquid, vapour or gas. It is used as a vacuum compressor or a vacuum pump in order to produce vacuum in steam turbine systems, in refrigeration systems, for bulk material transport etc. The actual efficiency is low, ranging from 0.1 to 0.35. The process is non-reversible due to mixing of two flows.

The suction pipe of an ejector pump system is connected to the coldest spot of the turbine condenser, where there is remarkably little steam due to sub-cooling, and therefore almost pure air is sucked out. A steam ejector is a two-stage flow-type compressor. Compression is achieved through fresh steam flow energy. In our case, the device comprises two stages. In the primary stage, i.e. the condensation stage, the sucked out air is compressed at a pressure of approximately 0.25 bar. A mixture of the sucked out gases and working steam from the primary ejector is led to the primary cooler. Most of the steam is condensed here and returned to the condenser through a special barometer loop. The mixture remaining in the primary cooler after the condensation is sucked out at the steam pressure of the secondary (atmospheric) stage and compressed to the pressure slightly higher than the atmospheric pressure, then led, together with the steam from the second stage, to the secondary cooler. The steam is also condensed in this cooler. The condensate passes through the condensate pot to the turbine condenser and the residual extracted air to the atmosphere. Fig. 3 illustrates the ejector system operating principle and the measurements of the existing system.

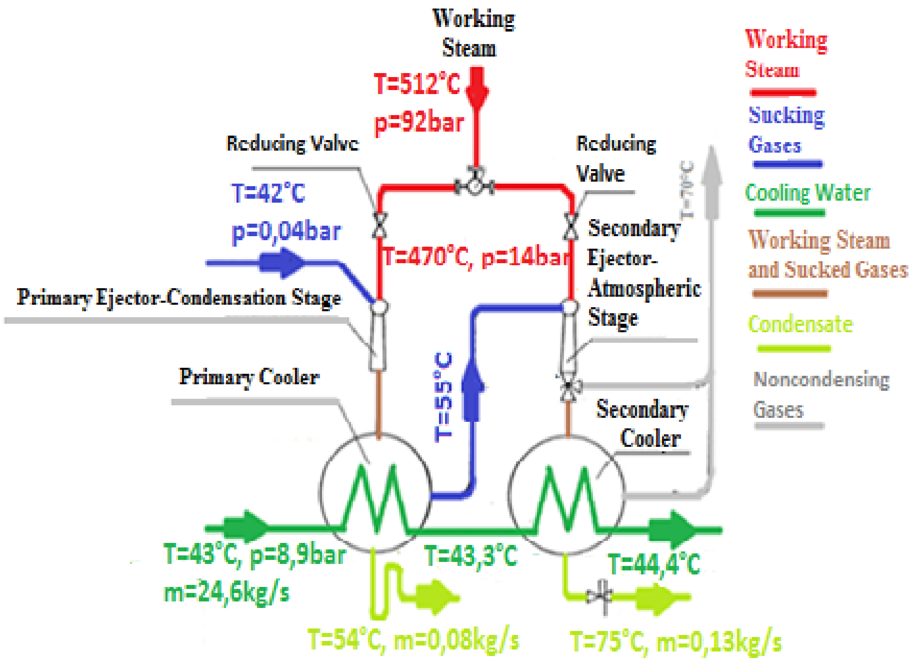

Figure 3: *Turbine ejector system and the measurement results, Strušnik [6]*

4 EJECTOR SYSTEM MATHEMATICAL ANALYSIS

The motive steam enters the ejector at point 0 and flows through the Laval nozzle, where the steam expands at up to supersonic speed. In the mixing chamber *x*, the expanded motive steam sucks gases from the turbine condenser 4, where they mix with each other from point 1 to point 2. The mixture of gases enters a diffuser, where kinetic energy of a gas transforms into pressure energy and at point 3, the gases exit at a higher pressure and a lower speed (Fig. 4).

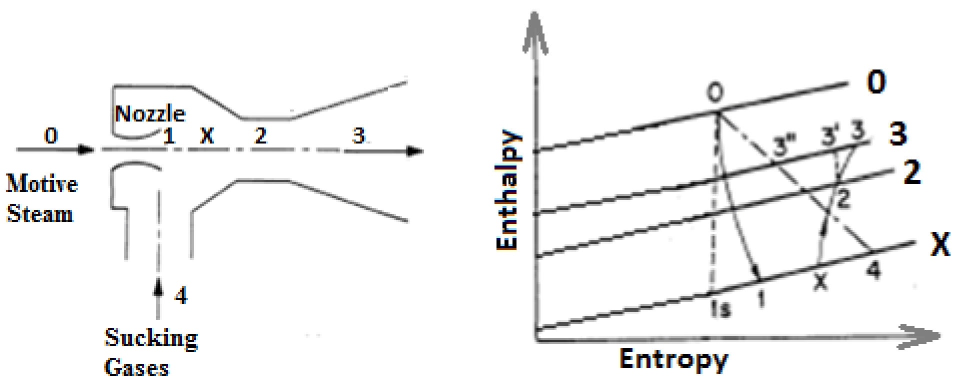

Figure 4: *Ejector model and Mollier h-s diagram, Strušnik [6]*

The mathematical analysis is based on the facts of conservation of momentum, mass and energy balances in each part of the ejector (Laval nozzle, mixing chamber and diffuser)

The following facts are taken into consideration in the calculation:

- There is no heat transfer in the transformations in the Laval nozzle,
- The motive stream expands in the Laval nozzle from the initial state of *p0* up to pressure in mixing chamber *px*,
- We assume that pressure *px* in the mixing chamber equals the gas suction pressure *p4*,
- The gases in the mixing chamber mix with each other and are regarded below as an ideal gas,
- Potential energy is negligibly low and is not taken into consideration,
- Outlet speed from the diffuser is extremely low and is therefore neglected.

4.1 Conditions in the Laval nozzle

It is assumed in the calculation of the mass flow through the Laval nozzle that the motive fluid is an ideal gas. The mass flow equals:

$$\dot{m}_0 = \frac{A_L * p_0}{T_0} \sqrt{\frac{\kappa}{R}\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}} \tag{4.1}$$

where $\dot{m}_0$ - motive gas mass flow rate,

R - gas constant,

AL - cross section of the Laval nozzle,

p_0 - inlet gas pressure,

T_0 - inlet gas temperature,

κ - ratio of specific heats of gas.

The Laval nozzle isentropic efficiency is defined as:

$$\eta_s = \frac{h_0 - h_1}{h_0 - h_{1s}} \approx 0{,}97 \tag{4.2}$$

where η_s - isentropic efficiency of a nozzle,

h_0 - specific enthalpy of inlet motive gas,

h_1 - specific enthalpy of nozzle expansion,

h_{1s} - specific enthalpy of isentropic nozzle expansion.

Speed at exit from the nozzle is calculated as:

$$c_{1s} = \sqrt{2 * \eta_s * (h_0 - h_{1s})} \tag{4.3}$$

where c_{1s} - isentropic nozzle gas velocity.

4.2 Conditions in the mixing zone

The flow of the motive gas entering through the Laval nozzle and of the pumped out gas is mixed in the mixing zone. The thermodynamic state is described by means of the following equations:

Mass balance:
$$\dot{m}_0 + \dot{m}_4 = \frac{A_2 * c_2}{v_2} \tag{4.4}$$

where $\dot{m}_4$ - mass flow of gas pumped out,

A_2 - inlet cross sectional area of the diffuser,

c2 - inlet diffuser speed of gases,

v_2 - inlet diffuser specific volume of gases.

Momentum:
$$m_0 * c_1 + p_4 * A_2 = [\dot{m}_0 + \dot{m}_4] * c_2 + p_2 * A_2 \tag{4.5}$$

where c_1 - exhaust nozzle gas speed,

p_4 - pumped gas pressure,

p_2 - gas pressure at the inlet to the diffuser.

Energy balance:
$$\dot{m}_0 * h_0 + \dot{m}_4 * h_4 = [\dot{m}_0 + \dot{m}_4] * \left[h_2 + \frac{c_2^2}{2} \right] \tag{4.6}$$

where h_4 - specific enthalpy of pumped gas,

h_2 - specific enthalpy at the inlet to the diffuser.

A constant pressure in the mixing chamber *p4=p1* is taken into consideration in the calculation.

4.3 Conditions in the diffuser

Energy balance:
$$h_2 + \frac{c_2^3}{2} = h_3 \tag{4.7}$$

where h_3 - specific enthalpy at the outlet from the diffuser.

By introducing isentropic efficiency of a diffuser we obtain:

$$h_{3'} - h_2 = \eta_d \cdot [h_3 - h_2] \tag{4.8}$$

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where $h_{3'}$ - isentropic specific enthalpy at the outlet from a diffuser,

η_d - diffuser efficiency.

4.4 Mach number in the mixing zone

The Mach number in the mixing zone is calculated:

$$M_2 = \frac{c_2}{[\kappa * p_2 * v_2]^{0,5}} \tag{4.9}$$

Where M_2 - Mach number in the mixing zone.

Using Equations (4.4), (4.5) and (4.9) the necessary diameter is expressed and calculated:

$$A_2 = \frac{\dot{m}_0 * c_1}{p_2 * [\kappa * M_2^2 + 1] - p_4} \tag{4.10}$$

Equations (4.7) and (4.8) are used to calculate:

$$h_{3'} - h_2 = \eta_d * \left[\frac{c_2^2}{2}\right] \tag{4.11}$$

Other equations for ideal gases are used, i.e. $h = c_p * T$, $c_p = \frac{\kappa * R}{\kappa - 1}$ and $M = \sqrt{\kappa * R * T}$ to obtain the equation:

$$\left[\frac{p_3}{p_2}\right]^{\frac{\kappa-1}{\kappa}} - \frac{\eta_d * M_2^2 * [\kappa - 1]}{2} = 1 \tag{4.12}$$

If the inlet diffuser cross-section A_2 is known, it is possible to express M_2 depending on p_2 from Equation (4.10) to obtain:

$$M_2^2 = \frac{\dot{m}_0 * c_1 + p_4 * A_2}{\kappa * p_2 * A_2} - \frac{1}{\kappa} = \frac{1}{\kappa} \cdot \frac{\dot{m}_0 * c_1 + (p_4 - p_2) * A_2}{p_2 * A_2} \tag{4.13}$$

By inserting Equation (4.12) into Equation (4.13) we obtain:

$$\left(\frac{p_3}{p_2}\right)^{\frac{\kappa-1}{\kappa}} - \frac{\eta_d * (\kappa - 1)}{2\kappa}\left(\frac{\dot{m}_0 * c_1 + p_4 * A_2}{p_2 * A_2} - 1\right) - 1 = 0 \tag{4.14}$$

If p_3 (pressure behind the ejector) is known, Equation (4.13) is solved, and p_2 is obtained and then M_2 is calculated from Equation (4.12). It is necessary to determine c_2, T_2 and $\dot{m}_4$. Taking into account Equation (4.4) the energy Equation (4.6) can be written as follows:

$$\dot{m}_0 * h_0 + \dot{m}_4 * h_4 = A_2 * c_2 * \frac{p_2}{RT_2}\left[\frac{\kappa * RT_2}{\kappa - 1} + \frac{c_2^2}{2}\right] = A_2 * c_2 * p_2\left[\frac{\kappa}{\kappa - 1} + \frac{\kappa}{2} * M_2^2\right] \tag{4.15}$$

From Equation (4.15) we express c_2 and insert it into Equation (4.5) to obtain, after a transformation, a quadratic equation for the calculation of the flow rate of gases $\dot{m}_4$, whose solution is:

$$\dot{m}_4 = \frac{\dot{m}_0(h_0+h_4) \pm \sqrt{\dot{m}_0^2(h_0+h_4)^2 + 4h_4\left\{\dot{m}_0^2 h_0 - (\dot{m}_0 c_1 + p_4 A_2 - p_2 A_2)\frac{\kappa}{\kappa-1}A_2 p_2\left[1+\frac{\kappa-1}{2}M_2^2\right]\right\}}}{2h_4}$$

(4.16)

4.5 Calculation of flow rate of non-condensable and condensable phase of primary and secondary ejector

In the calculation of the phase quantity of the pumped gases of the primary ejector, the proportion of a mixture of the primary ejector ($\dot{m}_{4,1}$) is taken into account on the basis of experience, i.e. 75% of non-condensable phase and 25% of condensable phase, whereby $\dot{x}_{4,1H2O} = 0{,}25$ and $\dot{x}_{4,1air} = 0{,}75$.

It is necessary to calculate the phase ratio to be sucked into the secondary ejector. In view of the fact that the Laval nozzle of the secondary ejector has a larger diameter than the primary ejector, the secondary ejector pumps a higher quantity of gases. This means that the secondary ejector also pumps a portion of the primary ejector motive steam not condensed in the primary cooler. The quantity of the primary ejector motive steam pumped via the cooler into the secondary ejector is calculated:

$$\dot{m}_{4,2pr} = \dot{m}_{4,2} - \dot{m}_{4,1} \tag{4.17}$$

where $\dot{m}_{4,2pr}$ - quantity of primary ejector motive steam pumped via the primary cooler into the secondary ejector,

$\dot{m}_{4,2}$ - quantity of pumped gases of the secondary ejector,

$\dot{m}_{4,1}$ - quantity of pumped gases from the primary ejector condenser.

The share of non-condensable phase pumped by 2nd rate ejector is calculated:

$$\dot{x}_{4,2no-cond} = \dot{x}_{4,1air} * \dot{m}_{4,1} = 0{,}75 * \dot{m}_{4,1} \tag{4.18}$$

where $\dot{x}_{4,2no-cond}$ - share of non-condensable phase pumped by 2nd rate ejector,

$\dot{x}_{4,1air}$ - share of non-condensable phase,

$\dot{m}_{4,1}$ - quantity of pumped gases from 1st ejector rate condenser.

The share of condensable phase pumped by the secondary ejector is calculated:

$$\dot{x}_{4,2cond} = \dot{x}_{4,1H2O} + \dot{m}_{4,2pr} = 0{,}25 * \dot{m}_{4,1} + \dot{m}_{4,2pr} \tag{4.19}$$

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where $\dot{x}_{4,2cond}$ - share of condensable phase pumped by the secondary ejector,

$\dot{x}_{4,1H2O}$ - share of condensable phase,

$\dot{m}_{4,1}$ - quantity of pumped gases from the primary ejector condenser,

$\dot{m}_{4,2pr}$ - quantity of the primary ejector motive steam pumped via the primary cooler into the secondary ejector.

5 EJECTOR SYSTEM MODEL IN MATLAB-SIMULINK SOFTWARE ENVIRONMENT

Using Matlab-Simulink software, an ejector was modelled to calculate, by means of the equations described in Chapter 4, the conditions in the ejector, the quantity of pumped gases, the consumption of ejector motive steam and the work done by the turbine if the ejector motive steam expanded in the turbine. The ejector model consists of the main model (Fig. 6) and three sub-models. The first sub-model (Fig. 5) calculates the conditions in the Laval nozzle. The main model computes the quantity of the pumped gases for each ejector separately by means of the computed parameters. The second sub-model computes the phase ratios of gases pumped into the secondary ejector. On the basis of the motive gas quantity used for the ejector system operation, the third sub-model computes the power produced as a result of the expansion of the ejector motive steam in the turbine. The model is conceived so that the narrowest nozzle diameter and the motive steam quality may be manually selected.

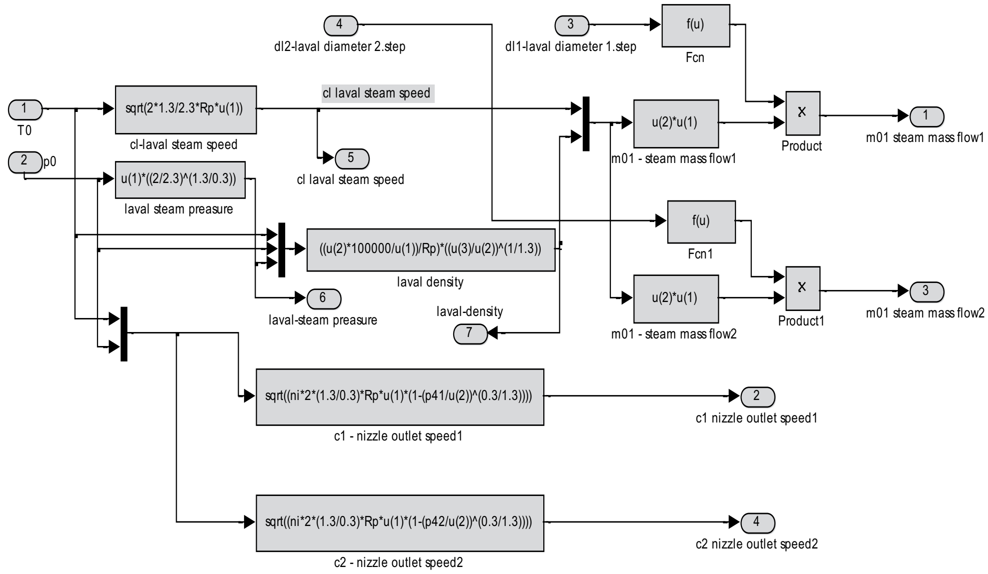

Figure 5: *Ejector sub-model, Matlab [5]*

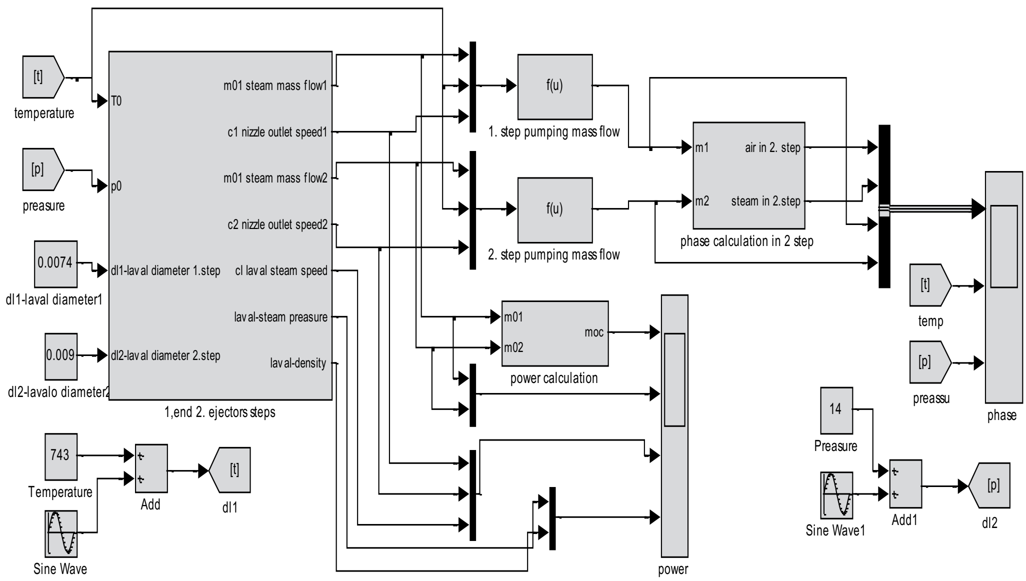

Figure 6: *Ejector model, Matlab [5]*

6 ANALYSIS OF THE EXISTING EJECTOR SYSTEM AND DETERMINATION OF GEOMETRICAL DATA OF THE ALTERNATIVE EJECTOR SYSTEM

In order to ensure a good turbine condenser vacuum, the amount of gases pumped using the existing ejector system is important. The alternative ejector system must ensure an equivalent flow rate to the existing system. In view of the fact that the alternative system motive steam is of poorer quality, an appropriate nozzle diameter must be defined. To this end, an ejector model is to be used.

The measured data of the existing system (Fig. 3) and geometrical data (Table 1) were put into the ejector model.

Table 1: *Existing ejector system data*

Designation	Meaning	Primary ejector	Secondary ejector
d_L	Narrowest Laval nozzle diameter	7.4 mm	9.0 mm
d_1	Laval nozzle outlet diameter	35.6 mm	23.6 mm
d_2	Diffuser inlet diameter	49.6 mm	32.3 mm
d_3	Final diameter of a diffuser	102.5 mm	87.2 mm
δ	Diffuser expansion angle	10°	10°
η_s	Nozzle efficiency	0.97	0.97

η_d	Diffuser efficiency	0.75	0.75
p_0	Motive steam pressure	14 bar	14 bar
T_0	Motive steam temperature	470°C	470°C
T_c	Temperature of gases in a condenser	42°C	42°C

On the basis of the results obtained of the flow rate of the existing ejector system (primary ejector pumps 0.04483 kg/s of gases and secondary ejector 0.05734 kg/s), an alternative ejector system was conceived. It was established that the alternative ejector system provides sufficient flow rate when the narrowest primary Laval nozzle diameter measures 9.5 mm and the secondary one measures 12 mm. The alternative primary ejector system with the above-indicated Laval nozzle dimension pumps 0.04509 kg/s and the secondary one 0.05777 kg/s of gases. The alternative ejector system pumps a sufficient amount of gases to maintain a proper pressure condition in the condenser, as it pumps an amount higher than the amount pumped by the existing system. The data of the alternative ejector system is indicated in Table 2.

Table 2: *Alternative ejector system data*

Designation	Meaning	Primary ejector	Secondary ejector
d_L	Narrowest Laval nozzle diameter	9.5 mm	12.0 mm
d_1	Laval nozzle outlet diameter	35.6 mm	23.6 mm
d_2	Diffuser inlet diameter	49.6 mm	32.3 mm
d_3	Final diameter of a diffuser	102.5 mm	87.2 mm
δ	Diffuser expansion angle	10°	10°
η_s	Nozzle efficiency	0.97	0.97
η_d	Diffuser efficiency	0.75	0.75
p_0	Motive steam pressure	8.8 bar	8.8 bar
T_0	Motive steam temperature	290°C	290°C
T_c	Temperature of gases in a condenser	42°C	42°C

The ejector model graphic results are presented below with each figure containing two graphs. The upper set of graphs shows the results of the existing ejector system model along with the data indicated in Table 1, whereas the lower set of graphs illustrates the alternative ejector model results along with the data in Table 2. To provide a more explicit presentation of the results, a simulation of the sinusoidal oscillation of the ejector motive steam was made as shown in Fig. 7. The amplitude of temperature fluctuation, marked in yellow, is 10°C. The amplitude of the sinusoidal pressure fluctuation, marked in red, is 0.5 bar.

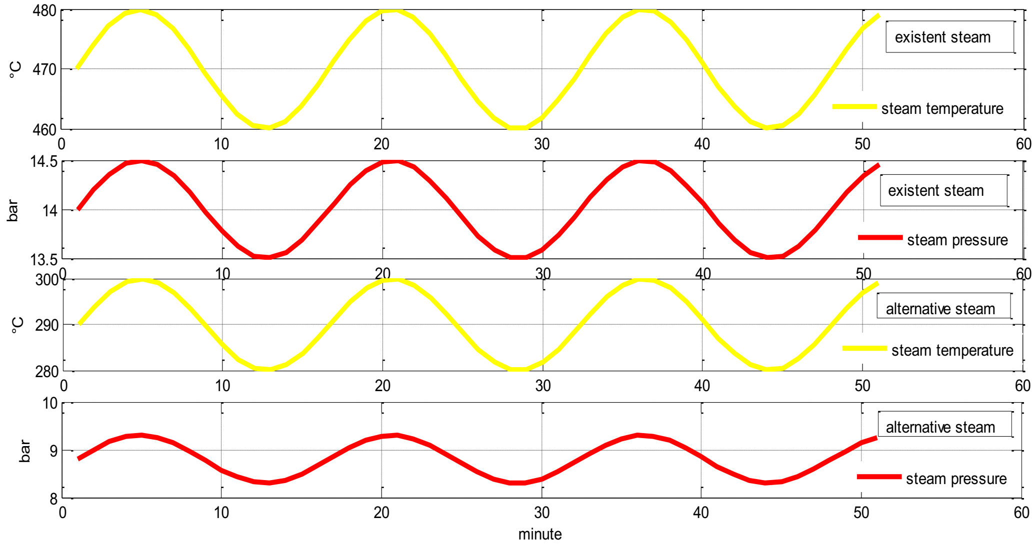

Figure 7: *Sinusoidal oscillation of ejector motive steam, Matlab [5]*

Fig. 8 shows the conditions of Laval nozzle in Laval cross-section. The yellow curve shows the pressure and the pink curve the ejector motive steam density. A 2.8 bar decrease in pressure in the Laval cross-section and a slight decrease in density are observed in the operation of the ejector with alternative motive steam.

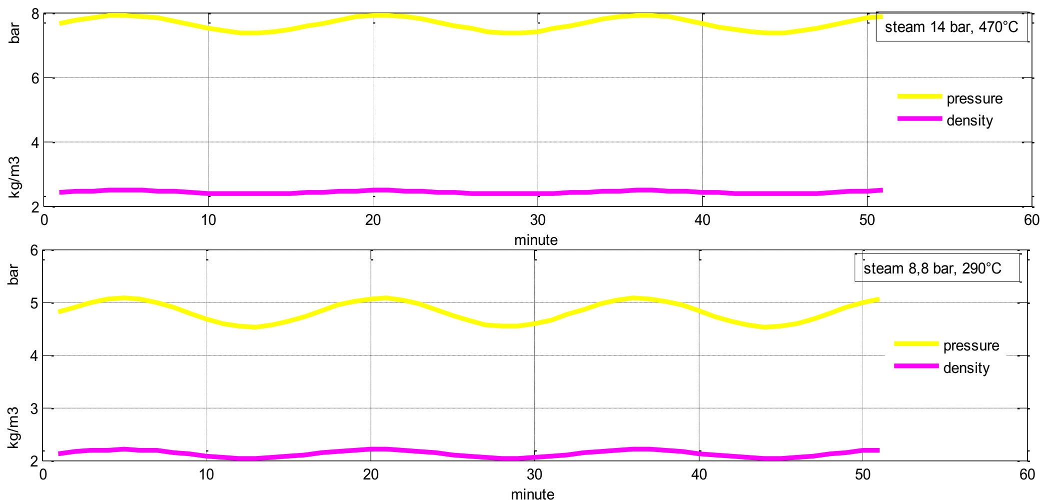

Figure 8: *Conditions in the Laval nozzle Laval cross-section, Matlab [5]*

Fig. 9 shows Laval nozzle velocities. The yellow curve shows the outlet nozzle speed of the primary motive steam, the pink curve the outlet nozzle speed of the secondary motive steam and the blue curve the motive steam speed in Laval cross-section. Lower velocities are observed in the alternative ejector system. The primary and secondary outlet nozzle velocities are lower by 220 m/s. The velocity of steam in Laval cross-section is lower by 50 m/s.

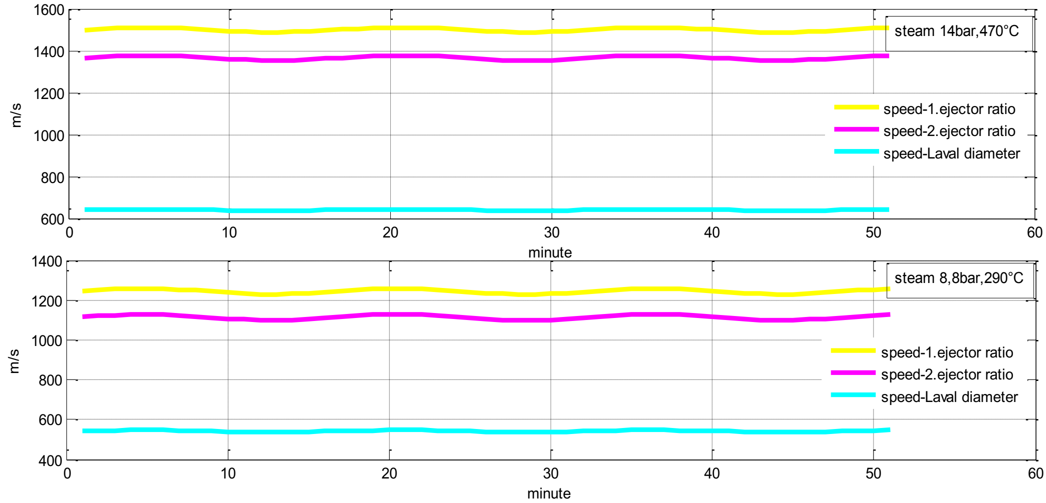

Figure 9: *Laval nozzle velocities, Matlab [5]*

Fig. 10 shows the mass flow rates of the ejector motive steam. The yellow curve illustrates the consumption of the primary ejector motive steam and the pink curve the consumption of the secondary ejector motive steam. Due to a larger Laval nozzle, the motive steam consumption in the alternative ejector system increases. The primary ejector consumes 0.014 kg/s more motive steam for pumping and the secondary ejector 0.036 kg/s more motive steam. For its operation, the existing ejector system consumes 0.17 kg/s steam in total but the alternative one comsumes, 0.22 kg/s of steam. This means that the total motive steam consumption of the alternative ejector system is higher by 0.05 kg/s.

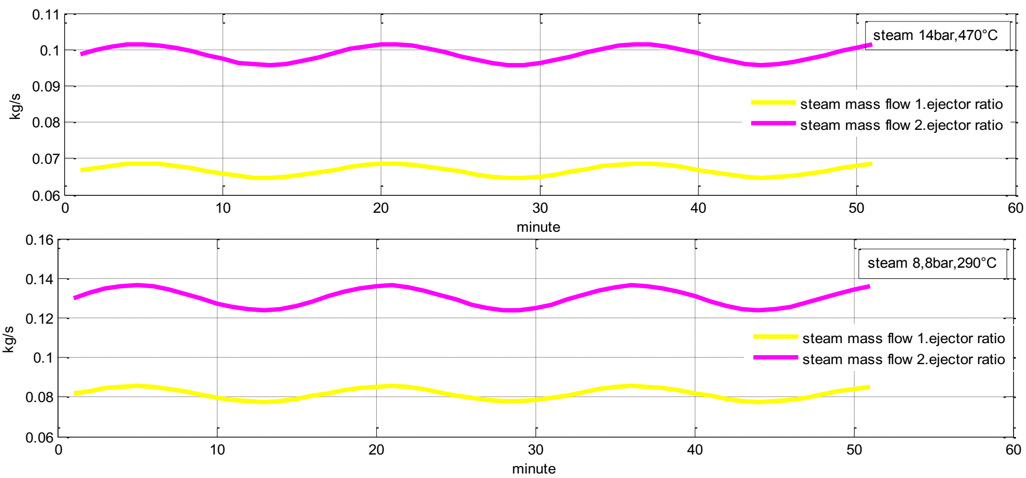

Figure 10: *Ejector motive steam flow rates, Matlab [5]*

Fig. 11 shows the flow rates of the pumped gases. The yellow curve illustrates the amount of non-condensable gases (air), pumped by the secondary ejector. The violet curve shows the amount of condensable gases (water vapour), pumped by the secondary ejector. The red curve shows the total flow rate of the secondary ejector, being the sum of the flow rate of non-condensable and condensable gases. The blue curve shows the primary ejector total flow rate.

It is observed that the secondary ejector pumps a larger quantity than the primary ejector, which means that the secondary ejector also pumps a portion of the motive steam of the primary ejector. An important item of information regarding the alternative ejector system dimension is the equivalent flow rate of the existing system. Fig. 11 shows the primary and secondary ejector flow rates (blue and red curves).

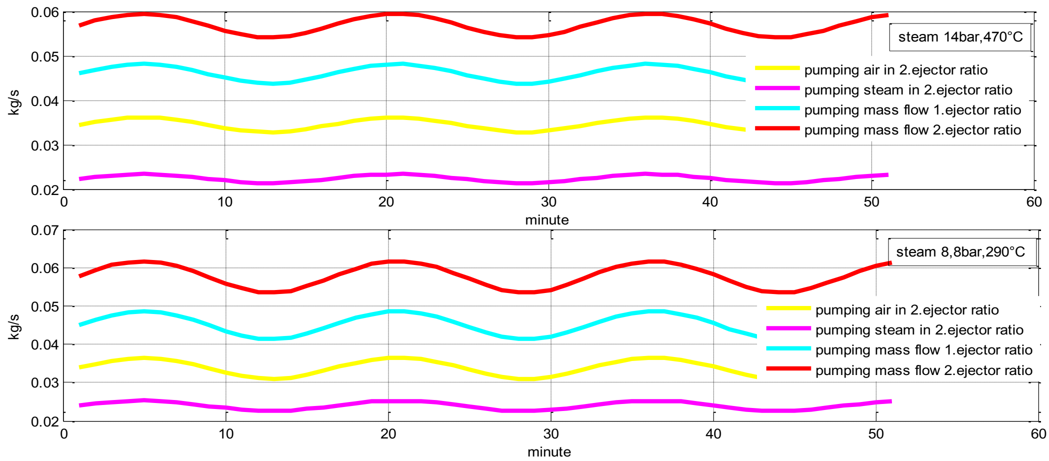

Figure 11: *Quantity of pumped gases, Matlab [5]*

Fig. 12 shows the generated power that would be developed by the ejector motive steam expansion in the turbine. The yellow curve shows the generated power in the case of expansion of the total quantity of the motive steam in the turbine that is used to drive the primary and secondary ejectors. In this case, the steam expands from the parameter of the quality of the steam produced in the steam boiler and expansion in the turbine to the pressure state in the turbine condenser. The pink curve shows the generated power that would be developed by the quantity of the ejector motive steam in the expansion of the steam in the turbine, from the state of quality of the steam produced in the boiler to the pressure state of the Turbine 3 extraction.

The alternative ejector system is supplied with the steam of the third turbine extraction. For its operation, the existing ejector system uses the quantity of the motive steam that would produce in the turbine expansion from the pressure state in the condenser an additional amount of 197 kW of electric power. For its operation, the alternative ejector system uses a larger quantity of motive steam that would produce additional 240 kW of electric power in a turbine expansion to the pressure state in the condenser. However, the alternative ejector system is supplied from the third turbine expansion, which is why the ejector motive steam of this system expands in the turbine to the pressure state of the third turbine expansion and generates 60 kW of electric power. As the motive steam of the alternative ejector system actually expands in the turbine to the state of the third expansion, the power of expansion to the state of the third expansion has to be deducted from the total generated power of 240 kW (expansion to the pressure state in the condenser). The actual generated power of 180 kW is thus obtained that would be developed by the alternative motive steam. The comparison of both ejector systems shows that the alternative ejector system actually generates only 17 kW more electric power due to the expansion of the ejector motive steam in the turbine.

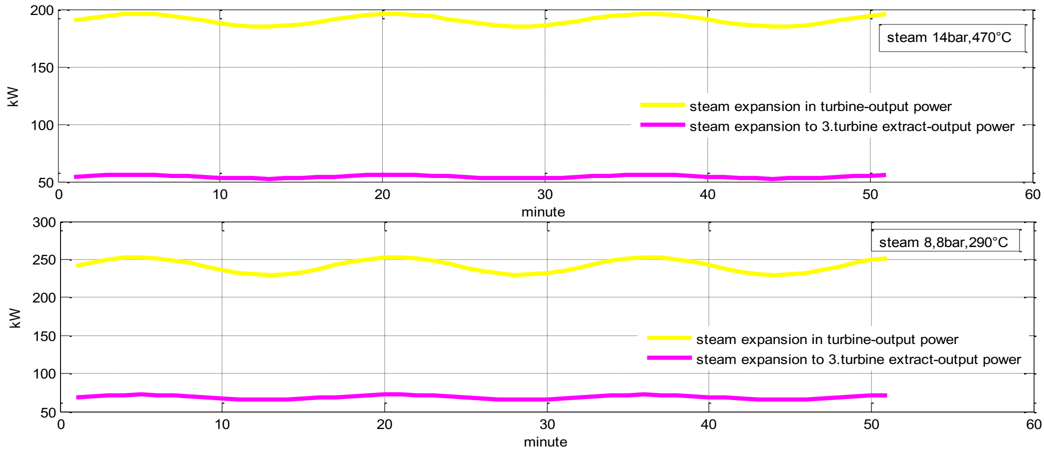

Figure 12: *Power that would be developed by the ejector motive steam in steam expansion in the turbine, Matlab [5]*

7 CONCLUSION

The measurements and the ejector model data show that a reconstruction of the system is necessary in order to adjust the existing ejector system to the new motive steam parameters.

Certain parts of the ejector need to be adjusted to the new computed dimensions (Table 2). Both Laval nozzles have to be replaced and the ejector system coolers enlarged by 15% due to the higher consumption of the ejector motive steam. As a result, the secondary ejector mainly pumps non-condensable gases. The automated system operation may be achieved by installing electric shut-off valves. All the other ejector system elements, such as diffusers, mixing chambers, connecting fittings and flanges could be used without reconstruction. No manual throttle valves are needed as the steam would have a constant pressure. Due to the replacement of the nozzles and the constant pressure of the alternative motive steam, the ejector mixing chamber would have a constant volume flow and consequently smoother operation and uniform pumping of gases from the condenser. The existing ejector system may also be manually controlled. Within the reconstruction, the steam electric valves would be mounted to a more accessible position. The steam valve operation would be remotely controlled. The alternative ejector system motive steam pressure would be ensured from various sources and would not be dependent only on the boiler operation.

The main observation regarding the system reconstruction is that the ejector motive steam previously expands in the turbine to the third turbine expansion pressure and that the ejector system is not directly supplied with the boiler-reduced steam. Additionally, 60 kW of electric energy are generated, and the motive steam enthalpy drop is used, which is lost in the original case of damping. A weakness of the alternative ejector system is that it uses a larger quantity of motive steam and that only 17 kW of additionally generated electric power in the turbine is saved.

The ejector system reconstruction is reasonable only for the sake of a more reliable and safer alternative system operation.

The advantage of the ejector system lies in the easy maintenance of its driving parts and its reliable operation. It would be reasonable to consider a replacement of the ejector system with an electric vacuum pump characterised by lower energy consumption and a considerably higher efficiency.

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[8] **TE-TOL.:** *Thermal power plant*, information system (TIS).

Nomenclature

(Symbols)	(Symbol meaning)
A	cross section
c	speed
h	specific enthalpy
M	mach number
$\dot{m}$	mass flow
p	pressure
R	gas constant
T	temperature
v	specific volume
x	flow rate
κ	ratio of specific heats
η	efficiency