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THE EFFECT OF OUTSIDE AIR TEMPERATURE IN PERFORMANCE OF THE EJECTOR APPLIED FOR DIRECT CONNECTION TO THERMAL NETWORK

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Abstract: This paper describes the usage of water-water ejectors for direct connection by mixing the hot water from district heating network with the cold water used from heating systems. The purpose of this paper is to analyze the temperature of the water supply in the secondary network of thermal consumers, replacing the heat exchangers with ejectors. The hydrodynamic and thermal analysis for this work is based on the maintenance of the water supply temperature in the secondary network, which is connected to the primary heating network of Termokos - district heating company in Prishtina, Kosovo. Analysis of contours of velocity, pressure, and temperature was conducted using Solid Works software 2014 including Computational Fluid Dynamics (CFD) codes for ejectors. Moreover, the analysis was done based on the following mathematical models: Mass Equation, Differential Equations of Navier-Stokes, and Energy Equation. The thermal analysis was realized on ThermoCalc, Microsoft Excel. The results of the analysis showed that thermal energy could be saved by replacing heat exchangers with ejectors (jet pumps). In the future, the obtained results may be used from engineers of district heating networks to utilize thermal energy in a more efficient way.
Keywords: ejector, air temperature, thermal network, CFD modeling, efficiency

1. INTRODUCTION

The water-water ejector (jet pump) uses the principle of a convergent nozzle, which enables the conversion of pressure energy to kinetic energy. The increase in kinetic energy at the outlet of the primary tube allows the absorption of water from the secondary tube due to the decrease of the vacuum metric pressure. At the exit of the ejector, a diffuser is placed whose task is to increase the pressure on the velocity reduction account, therefore, the ejector is said to use the Venturi effect of the converging-diverging nozzle.

Because of their simplicity as mechanical equipment, ejectors are being applied more in thermal and nuclear power plants (steam turbine and condensers). The thermal and hydrodynamic model of the ejector is conducted due to the advantages it has compared to heat exchangers. Ejectors are used in thermal sub-stations for direct connection of heat consumers in thermal networks. Direct connection of heat consumers in thermal networks is done where the water from district heating network is mixed with water of residential or commercial buildings and is used for heating systems. Ejectors are stationary equipment, and therefore have greater longevity than heat exchangers because they do not have moving parts. Additionally, the ejector's size depends on the amount of heat, which requires a thermal consumer.

Hence, ejectors enable a direct connection of heat customers with thermal networks [1]. The following analysis aims to address the issues of contours of velocity, pressure, and temperature through the ejector and their application to district heating of Pristina City compared to heat exchangers, – using Solid Works software – CFD codes, and mathematical models. In mathematical models are including Mass Equation, Differential Equations of Navier-Stokes, Energy Equation, and Thermocalc (Microsoft Excel).

2. ANALYSIS OF THE PROFILES OF VELOCITY, PRESSURE, AND TEMPERATURE BY SOLID WORKS 2014 AND MATHEMATICAL MODELS

Computational Fluid Dynamics (CFD), as a branch of fluid mechanics uses numerical analysis, methods of finite elements, and algorithms for solving the differential equation of fluid flow dynamics. CFD analysis of ejector is helpful for the distribution of velocity, pressure, and temperature through the ejector. The result gained from the CFD analysis can be used from district

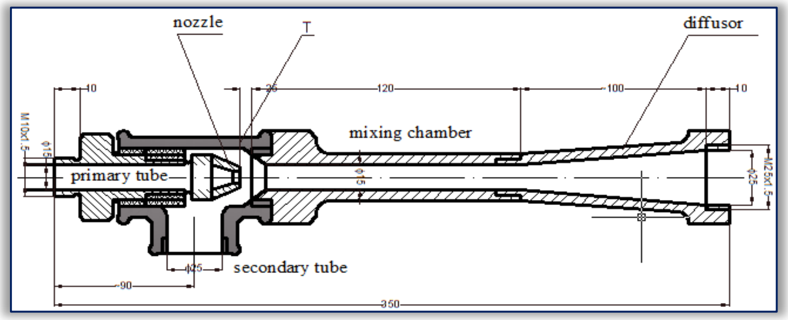

Figure 1.a. The geometry of modeling ejector

heating engineers to assign the right size of ejectors to save thermal energy. The Figure 1 describes the geometric properties of modeling ejector.

Mass Equation

The conservation of mass is a fundamental concept of physics. The equation of conservation of mass is applied when the mass of fluid remains constant throughout its boundary volume, namely in the field of flow:

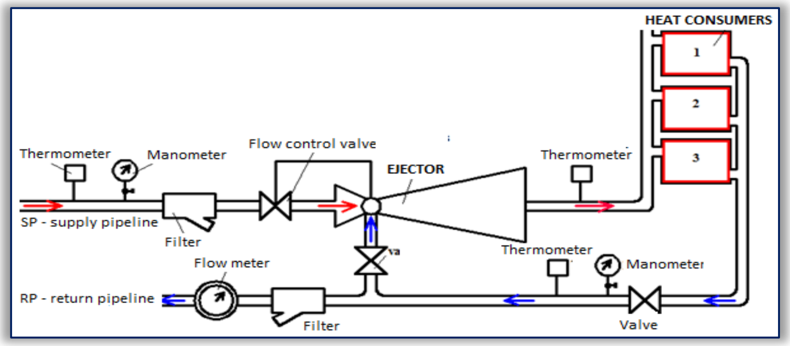

Figure 1.b. Scheme of thermal substation with ejector

$$\frac{dM}{d\tau}=0 \tag{1.1}$$

The conservation of mass for the boundary volume can be written as:

$$\frac{\partial}{\partial\tau}\int_{VC}\rho\cdot dV+\int_{SC}\rho\cdot V\cdot\hat{n}\cdot dA=0 \tag{1.2}$$

Equation (1.1) is known as the equation of continuity, which is applied to boundary volume (VC), is defined by boundary surface (SC). The first integral in the equation (1.2) on the left side shows the mass change in the unit of time at the border of volume. The second integral on the right side shows the net rate mass, which has passed through the boundary surface per unit time.

Differential Equations of Navier-Stokes

Differential equations of the fluid motion are described using equations given by the French mathematician L.M.H. Navier (1758-1856), and Sir British engineering G. G. Stokes (1819-1903), from which it has received the name of the Navier-Stokes equation.

The final form of the equations of Navier-Stokes's for dynamics of non-compressibility fluid is described by the differential equation, which is expressed by the formula:

$$\frac{d\vec{w}}{d\tau}=\vec{F}_{Vm}-\frac{1}{\rho}grad\ \vec{p}+w\Delta^2\vec{w}+\frac{1}{3}w\ grad\ div\ \vec{w} \tag{1.3}$$

The left part of the above equalizer shows the expression of the acceleration of the fluid, while the right part of the equation shows the forces that act on the fluid which causes the fluid to move. The first expression on the right-hand side (1.3) presents the action of volumetric forces on the fluid in the unit of mass. The second term takes into account the effect of the pressure forces. Lastly, the third term takes into account the effect of inertia forces, and the last expression takes into account the compressibility of the fluid.

If the fluid is non-compressible, the divergence of the velocity vector will be equal to zero, so that the vector equation of the Navier-Stokes will take the form:

$$\frac{d\vec{w}}{d\tau}=\vec{F}_{Vm}-\frac{1}{\rho}grad\ \ p+w\Delta^2\vec{w} \tag{1.4}$$

Energy Equation

The first law of thermodynamics for a control volume says: The amount of accumulated energy in a system in the unit of time is equal to the net amount of heat energy transmitted in units of time in the system and the net amount of work done in the system, namely:

$$\frac{d}{d\tau}\int_{system}e\cdot\rho\cdot dV=\left(\sum\dot{Q}_{in}-\sum\dot{Q}_{out}\right)_{system}+\left(\sum\dot{P}_{in}-\sum\dot{P}_{out}\right)_{system} \tag{1.5}$$

$$\frac{d}{d\tau}\int_{system}e\cdot\rho\cdot dV=\left(\sum\dot{Q}_{net}-\dot{P}_{net}\right)_{system} \tag{1.6}$$

In the above equation **e** should include all forms of energy: internal, potential, kinetic and others. We have then:

$$e=u+\frac{w^2}{2}+g\cdot z \tag{1.7}$$

Finally, we can write the energy conservation equation in the system with the expression:

$$\frac{d}{d\tau}\int_{sistem}e\cdot\rho\cdot dV=\frac{\partial}{\partial\tau}\int_{VC}e\cdot\rho\cdot dV+\int_{SC}e\cdot\rho\cdot V\cdot\hat{n}\cdot dA \tag{1.8}$$

The general form of the energy equation for a control volume is:

$$\left(\sum \dot{Q}_{net} - \dot{P}_{net}\right)_{system} = = \frac{\partial}{\partial \tau}\int_{VC}\left(u + \frac{w^2}{2} + g \cdot z\right) \cdot \rho \cdot dV + \int_{SC}\left(u + \frac{w^2}{2} + g \cdot z\right) \cdot \rho \cdot V \cdot \hat{n} \cdot dA \qquad (1.9)$$

where h is specific enthalpy given by: $(u + p/\rho) = h$.

Revision of literature

Jiankai Dong [2], established calculation of the water stability in ejector, and some factors that might influence the stability of the water ejector, such as the system form, the design outdoor air temperature, the design supply /mixing/ discharge water temperatures, the number of the floors and the heat transfer coefficient were analyzed. The application of ejector in the centralized heating system had also drawn some attention, such as in the high and low regions direct connection system by Wang [3], in primary and secondary network direct connection system by Yen [4] and in the heating system by Ren [5].

3 RESULTS AND DISCUSSION

After drawing the ejector in Solid Works 2014, we defined the type of grid, namely the size of cells. The smaller the cells, the obtained results are more accurate. The boundary conditions such as pressure, flow mass and temperature in the primary tube, secondary tube and at the exit of the diffuser were also defined in Solid Works. The data values are shown in the following figures.

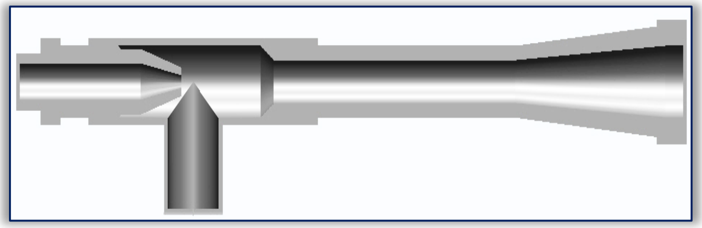

Figure 2. Drawing of water ejector with Solid Works 2014

Figure 3 shows the effect of the outside air temperature at the water supply temperature, which is heated in "Termokos" heating plant. The heated water is sent to the heating network of the city of Prishtina. Figure 3 shows that with the increase of the outside air temperature, the heating water temperature is linearly reduced. The following example shows the temperature dependence of the hot water temperature (primary network) on the outside air in a linear fashion: $t_{ws1} = 0{,}0002 \cdot t_{oair}^2 - 2.9173 \cdot t_{oair} + 87.19$ (°C).

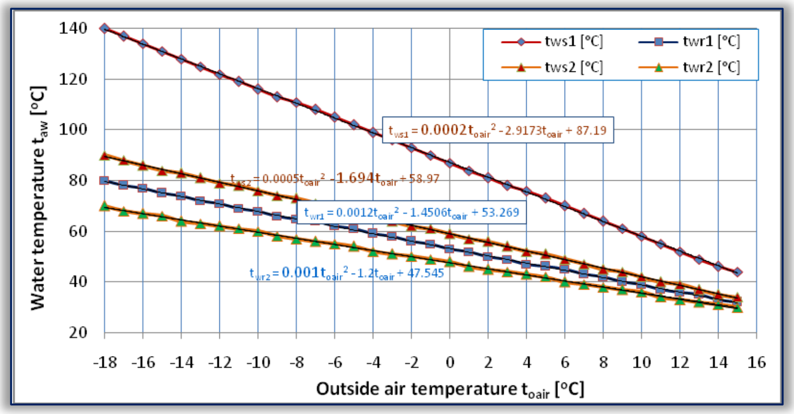

Figure 3. Water temperature in the primary and secondary network as a function of outside air temperature [6]

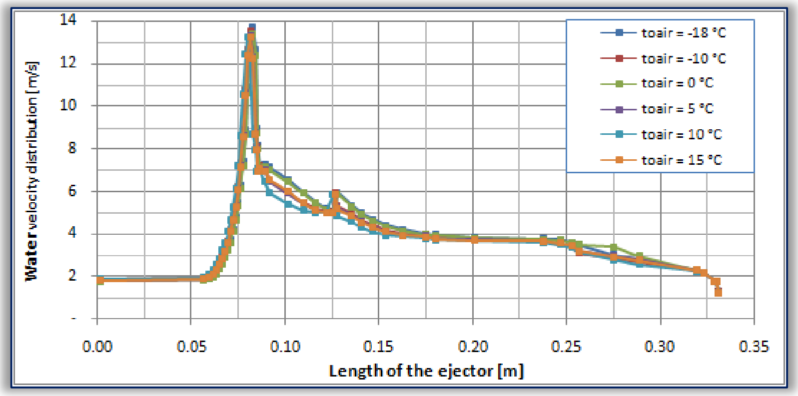

Figure 4. Contours of water velocity through ejector

For the various outdoor air temperature values = {-18, -10, 0, 5, 10, 15} °C, the following figures show the ejector modeling results such as the speed, water pressure, temperature.

Other parameters such as absolute pressure in primary tube 2.01325 bar [7], and static pressure in secondary tube 1.01325 bar [1], have remained constant to regulate the temperature of water mixed with mass flow at the outlet of the diffuser of 0.6 kg/s = const, which is sent to the heat consumer in order to meet the demands for thermal energy. The above-mentioned parameters have remained unchanged for several modeling of outside air temperatures.

Figure 4 shows the water velocity distribution through parts of the ejector, depending on the outside air temperature ranging from -18°C to 15°C. It is seen that the effect of the outside air temperature in water velocity streams through the ejector is relatively small. Said differently, this range of outdoor air temperatures does not affect the profile of water velocities.

For the water to flow, usually, the velocity in the secondary network (heat consumer) should be about 1.2÷3.0 m/s. In Figure 4 it is seen that the water velocity varies from 2.0 m/s to the primary tube and reaches the maximum velocity 13.6 m/s. at the exit of the nozzle. After leaving the nozzle, the water velocity drops sharply to 1.2 m/s. The velocity from the outlet stream in the primary tube allows the entraining of the water from the secondary network at a speed of 6.2 m/s. The water mixed with the flow due to the expansion of the section in which it flows decreases up to 5 m/s at the entrance of the mixing chamber. In the mixing chamber, the speed is constant, while during the flow through the diffuser, the speed continues to decrease to 1.2 m/s, where it is sent to the thermal consumer. Again, these values do not change considerably with the change of the outside air temperature.

From Figure 5 we can conclude that the outside air temperature ranging from -18 °C to 15 °C does not have any visible effect in the pressure drop characteristics through the ejector.

In Figure 6, the distribution of water temperature is determined by the ejector for different outdoor air temperatures. As can be seen from Figure 6, for a higher water temperature in the primary tube, respectively for the lower values of the outside air temperature, at the outlet of the ejector diffuser can be read the hot water temperature in the secondary network, e.g. for outdoor air temperature -18°C, the water temperature from the primary network is 414K of supply voltages for the secondary network 361 K, respectively for 0 °C 361.05 K and 339.46 K.

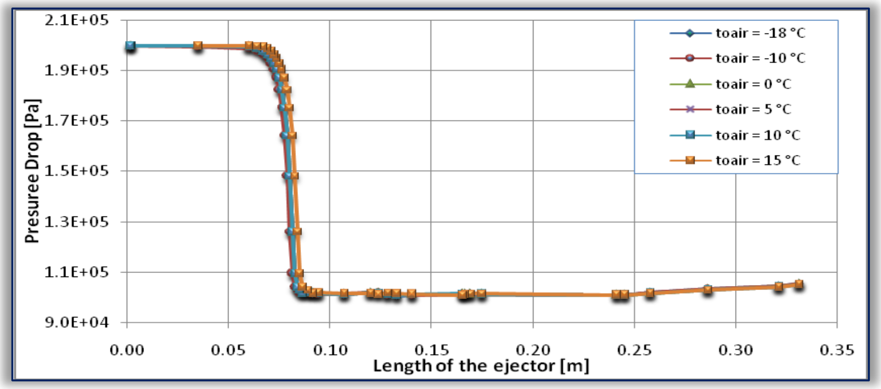

Figure 5. Contours of pressure distribution through parts of the ejector

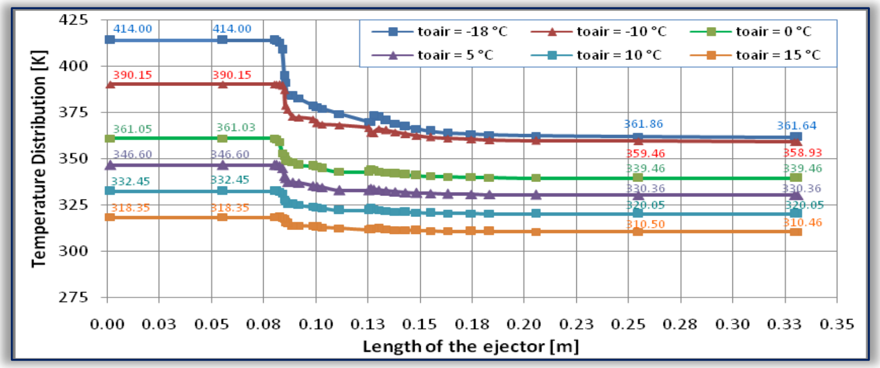

Figure 6. Temperature of water distribution through parts of the ejector

Water supply temperatures at the secondary network when ejectors are used are significantly higher than water supply temperatures, which are applied with heat exchangers by the enterprise of Termokos. This means that for the same heat demands by the thermal consumer when their ejectors are applied, the mass flow of hot water supply is lower.

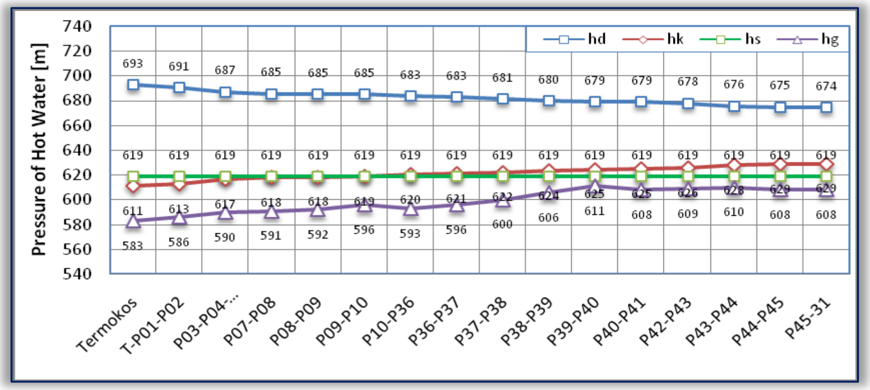

Figure 7. Piezometric pressure h_d [m] in the thermal network of the main pipeline from the enterprise of "Termokos" to the neighbourhood of Sunny Hill

In the diagram of Figure 7 with a blue line is shown the change of the piezometric pressure h_d [m] in the thermal network of the main pipeline from the enterprise of "Termokos" to the neighborhoods of Sunny Hill. With, hk [m] is drawn the piezometric pressure on the secondary network, h_s [m] is the static pressure, which is generated by enterprise "Termokos" and is measured when the heat exchanger of the district heating system is not working by corresponding to the pressure of 3.4 bar. With hg [m] is presented the geodetic height of these sectors, respectively the level of the track. To apply the direct connection of the thermal ejector for heat consumers, it is necessary that the pressure created between the primary and return network (h_d- h_k) to be greater than 15 m to enable normal operation of the ejector.

In Figure 8 with the red line is presented the water supply temperature in the secondary network when heat exchanger are installed in thermal substations, while with blue line is presented the water supply temperature in the secondary network when ejectors are installed in thermal substations. Also, the equations presented describe the temperatures of using the ejector, t_{Es2}=f(toair) , and the temperatures of water provided by "Termokos", t_{TCcs2}=f(toair) depending on the outside air temperature.

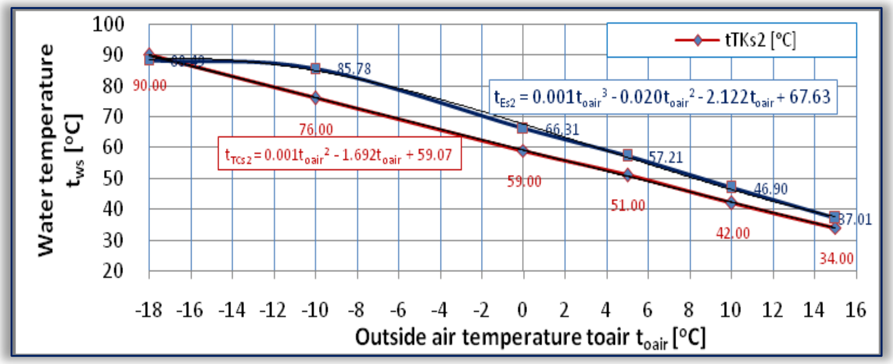

Figure 8. Relative compressions of energy saving derived from water supply temperatures in the secondary network depending on outside air temperature when there are used heat exchangers t_{TCs2} and ejectors t_{Es2}

As presented in the graph, when the outside air temperature is between intervals from -10 to 0°C, the thermal energy savings reach the maximum. This means that the higher the difference between t_{Es2} and t_{TCs2} is, the greater thermal energy savings are achieved. For outside air temperature -18°C this difference is invisible, while when toair=15°C there is shown a considerable energy saving.

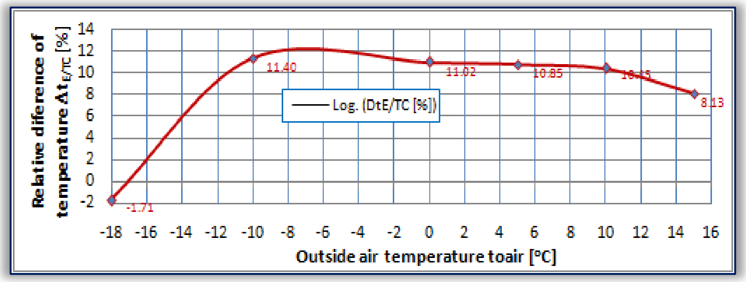

Figure 9. Percentage of thermal energy savings with using of ejector

In Figure 9 is shown the percentage of thermal energy savings achieved. This figure shows that when the outside air temperature is equal to the outside design temperature -18°C, the ejector application is not suitable. When the outside air temperatures vary from -10°C to 10°C, the thermal energy savings from the direct connection modes reach a percentage over 10%.

4. CONCLUSIONS

From the above analysis of Figures 2, 4, 5, 6 it can be concluded that the geometrical dimensions of the ejector play a key role in the distribution of pressure, temperature, and velocity of the water flow. The effect of various water temperatures in the primary and secondary tube has little impact in compression of the distribution of contours of pressure, temperature, and velocity of the water temperatures through ejector.

Also, the possibility of applying ejector equipment to the thermal network of Pristina city compared with heat exchangers enables using thermal energy in a more efficient way.

From the above analysis, it is concluded that when the outside air temperature is between intervals of -10 to 0°C, the effect of thermal energy saving has reached its maximum. This means as much higher the difference between t_{Es2} and t_{TSs2} to be the higher thermal energy savings are achieved. For outside air temperature -18°C this difference is invisible, while when t_{oair}=15°C there is shown a considerable energy saving. Moreover, from the above analysis of Figure 9, it is seen that thermal consumers from the enterprise of "Termokos" to the neighborhood of Sunny Hill can realize the direct connection to a thermal network using ejectors because the disposable pressure is above 15 m.

In the future, the obtained results from the analysis may be used from engineers of district heating networks to utilize thermal energy in a more efficient way.

Nomenclatures:

a [m/s^{-2}] - acceleration of the fluid;
m [kg] - mass of fluid
e [J/kg] - the total energy per unit mass
p [N/m^2] - pressure of the fluid
u [J/kg] - internal energy per unit mass
w [m/s] - the speed of movement of the fluid
γ [N/m^3] - specific weight of the fluid
ρ [kg/m^3] - density of fluid
τ [s] - time
μ [kg/ms] - dynamic viscosity of fluid
λ [W/mK] - coefficient of thermal conductivity
c_p [kJ/kgK] - specific heat of the fluid
A [m^2] - surface
F [N] - deterministic force of movement
V [m^3] - the volume of fluid
Q [W] - the power of heat
P [W] - the mechanical power

Index:

in - inlet
out - outlet
s - supply
r - return
dx - difference
Δ - Laplace operator
SC - boundary surface
VC - boundary volume

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