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District Heating Rehabilitation in Russia

The hydro-ejector system's replacement with the plate heat exchanger

ABSTRACT

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This thesis has two assigners. The first one is Karelia institute, which wanted more information about the hydro-ejector's functions and Russian heating norms. Besides of that Jupra Oy, which wanted to find out about co-operation possibilities between different counterparts in the Russian Heating sector. The company also wanted to make sure, that it is possible to replace the ejector system with a plate heat exchanger system. Karelia institute works closely together with Oulu University of Applied Sciences, so it was natural to me choose to help them with this task and Jupra Oy functioned as a financier of this thesis.

Background knowledge for this thesis was achieved with the assistance of a lecture of Oulu UAS Veli-Matti Mäkelä and multiple professors and lectures of Petrozavodsk State University. Research was international and was partly made in Finland and partly in the Russian Federation.

Methods of this thesis are based on acquiring information from literature, innovations and unofficial interviews of local experts and officials.

As a result this thesis gives common information about a district heating, market relationships, views of the energy policy, the tariff structure of the Russian Federation, the physical theory of the ejector, measurement methods of the ejector in the Russian Federation, the physical theory of the plate heat exchanger, the plate heat exchanger designing and introduces how to replace the ejector system with the plate heat exchanger system.

There are plenty of options to develop this work even further. Convergence work with different counterparts needs constant initiatives, mathematic calculation programs can be made with base on this thesis, further research on laws and norms can be done. This inspirational work gives also keys, which can lead to the new innovations.

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1. Foreword

The purpose of this work is to give acquirements to understand a Russian district heating system entirety so that it can be replaced with more modern, more energy effective, more service confident, more thermally quality and safer “Closed system”. The term "Closed system" means in this work, a system where primary side's and secondary side's fluids are separated with a plate heat exchanger.

Even though the main themes of this work are the guidelines and analysis of technical implementation, this work also lightly focused on district heating markets, while keeping the point of view of a customer and a subscriber. In some chapters get to know a little bit at Russian and European Union energy policy and their main numbers. It also attempts to find a common need to make alteration for the better and to add closer cooperation. The Russian heating sector reform gives great opportunity to closer cooperation and opportunity to find common interest between counterparts of the district heating sector.

Moreover, this is only theoretical work which tries to show that there is a big potential need or opportunity for this kind of rehabilitation. There are multiple ways to get this rehabilitation started. Possibly the hardest part is to find contracts, so that all the associates would get positive results in participating in the rehabilitation project.

2. District heating

In the district heating system, a heating energy is produced in the heating plants. For example the nuclear-, coal-, peat-, water- and the natural gas heating or the combined electricity and heating facilities. In most cases, the heat distribution is accomplished with a fluid, which is passing through the pipeline system to the consumer. The heat consumption consists of the heat losses of the building, domestic hot water heating, industrial progresses, air-conditioning et cetera.

The operating principle can generally be explained with fluid characteristics and -route. From a district heating facility, fluid leaves on the supply temperature and is transported to the consumer through a supply pipe. The cooled district heating water is transported back to the district heating facility through the return pipe. Power of the district heating water, which is influenced by temperature differences of the fluid, mass flow of the fluid and the heat transfer area of the fluid, is transported to a consumer with form of a fluid circulation.

The district heating supply temperature specification is affected by many variables, such as technical reasons. (11, p. 9.) Besides that, short-term and long-term strategies of the heating sector have also a role on defining the supply temperature (10, p. 24.)This is because determining fluid characteristics affect the pipe sizing, which has an influence on the pipe system's life circle.

The heat energy distribution value and the size of the pipelines can be influenced by raising the temperature difference. The temperature difference rise raises the transmission capacity of pipes in relation to the input power. It means that a bigger temperature difference between the pipelines allows the smaller pipe sizes to be used or in a second hand it allows a bigger energy transportation.

With higher temperatures, vaporization can be a problem and it can be solved by adjusting the pipe size, which aims to increase a static pressure in the pipeline, and/or adding a booster pump or circulation pump. Possible vaporization is influenced by the static pressure adjust. The intention of the methods above is to keep or raise the static pressure level enough, so that vaporization does not occur.

It is notable that in the Russian Federation and the former Soviet Union countries one of the main control of the heating power in the district heating systems is the supply fluid temperature, which controls the heat availability of consumers. The supply fluid measurement temperature is determined regionally, but the real time supply fluid temperature is not a constant value. Heating facilities control the value in the consideration of the external parameters changing.

Compared with the Northern-Europe district heating systems,the main difference can be found from a customer oriented secondary network control equipment, included for example heat exchangers, safety equipment, automation equipment, pumps and controlling devices, which are an essential part of the system in the Northern-Europe whereas in the Russian Federation and the former Soviet Union countries they are rarely used.

Lack of a customer-oriented temperature control is one factor in the energy efficient development in the Russian Federation heating sector. The customer-oriented temperature control will help to adjust the heating power consumption more specifically than a supplier-oriented temperature control. Besides that, it makes easier to control the heat comfortably, if other variables are taken into consideration. For example, variables can be regarded as an air-conditioning, heat losses and air leaks in the envelope of the building and an internal heat loads.

In the Russian Federation and the former Soviet countries, a main principle of secondary side fluid temperature adjusting for the planned level is a hydro-ejector system. The hydro-ejector system is mixing a returning fluid of a heated target with a hot supply fluid of the heat supplier. (11, p. 52.)In normal cases, temperature of the secondary side fluid, is not allowed to exceed 90°C, because of safety and practical reasons (15, c. 6.)

It is notable that a fluid mass flow should stay unchanged in a branch after the hydro-ejector or other component replacement, so that the nearby branch thermal power can stay undisturbed. That should be noted when replacing the open hydro-ejector with a closed heating exchanger system. The power-operated bypass valve and circulation pump can be installed in the purpose of adjustability of circulation in the branch.

3. District Heating Markets

District heating markets are worldwide, from where the Russian share of the district heating markets is significant in comparison with European, Chinese or American markets. Sizes of heat deliveries in the Russian Federation are between 1700000-2400000 GWh per year. Besides that, there are approximately 50000 district heating systems in the Russian Federation. The corresponding value in the Europe is 5000 (4).

3.1 Market Relationships

Marketing relationships have a very important role in industries where customers and suppliers have special needs on a quality of the product or the relationship with a counterpart. In the *DIAGRAM* 1 is a summary of different variables impact on the relationship between suppliers and customers. Information in the table is summarized from the dissertation of Mittilä, T, which is based on the Finnish unit's representatives' interviews in 12 different international companies (8). In diagram 1 the number 1 means minimum impact and 5 means major impact. Original scale of some values was 1…7. They are adjusted to correspond a scale of 1…5.

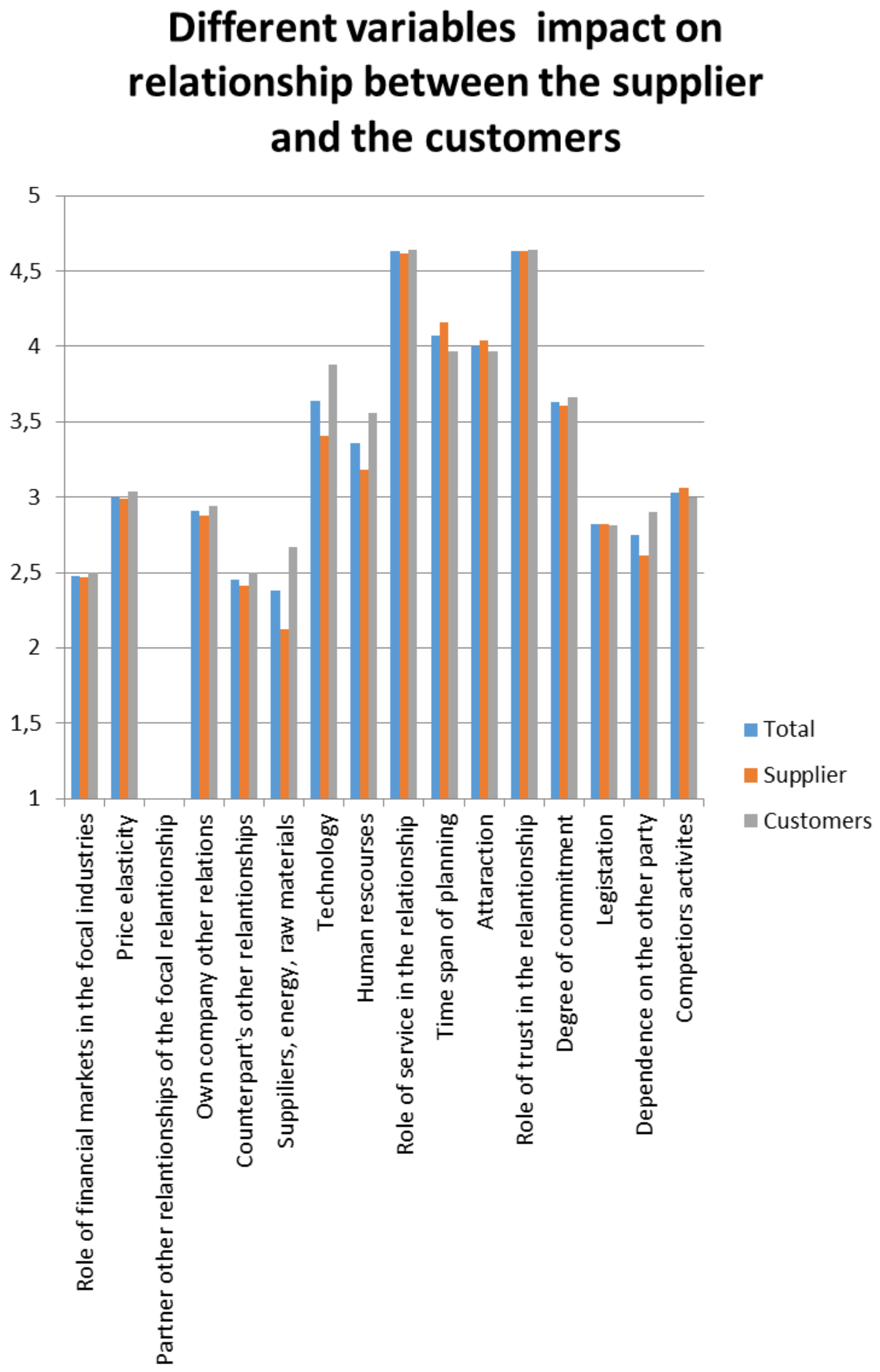


Diagram 1. Different variables impact on a relationship between the supplier and the customer.(8)

4. Russian Energy Strategy up to 2030

Some of Russian energy policy's main objectives are to maximize the effective use of natural energy recourses, improve the quality of the thermal conditions of the population and to raise the economic position of the country (10, p. 21).

"During the implementation of the Energy Strategy of Russia up to 2020 it was confirmed the following that its most important provisions are adequate to the real process of development in the energy sector of the country even in face of abrupt changes of external and internal factors that determine the basic parameters of operation of the fuel and energy complex of Russia."
(10, p. 3).

4.1 Objectives of Heating Sector

The strategic objectives of the heat supply development are to achieve a high level of comfort in residential, public and industrial premises. This includes growth in quality and the volume of the complex of services on the heat supply. The complex of the heat supply includes heating services, cold supply services, ventilation services, air-conditioning and domestic water services.
The Russian Federation heating sector tries to correspond with the objectives mentioned above for the level of leading European countries at the heating sector, as well as quality and affordable prices (10, p. 105).

There is an intention to the cut of the total heat production losses from 19 percent to 8-10 percent at the end of the year 2030 (10, p. 108.) In order to achieve that objective, it is determined that the heat supply service standards need to be tightened, a system structure need to be optimized at the combination of centralized and decentralized heat supply system. An improvement of reliability will be noted and is intended to improve an energy and an economic efficiency of the heat production. There is also an intention to focus on the safety questions and to make favourable conditions to the private investments and investors (10, p. 109).

"Heat power market will be established and relationships between its players will be harmonized"
"The population will be provided with high level of heat comfort corresponding to that of the countries with similar natural and climatic conditions (Canada, Scandinavian countries)." (10, p. 109).

4.2 Strategic Numbers in Table and Diagrams

The *TABLE* 1 and the *DIAGRAMS 2,3,4,5* below shows the direction and the magnitudes of the variables in the Russian energy strategy up to 2030.

Strategic numbers of the heat supply development for the period up to 2030				
	2008	Phase	Phase	Phase
Energy security and the heat supply reliability	(actu-al)	1	2	3
Heat supply cut-off rate, 1/year	0.27	no more than 0.25	no more than 0.20	no more than 0,15
Heat supply cut-off rate due to the fault of sources, 1/(sources•year)	0.06	no more than 0.05	no more than 0.03	no more than 0.01
Renovation of heat supply network (percentage of total length)	2	at least 10	at least 40	at least 90
Innovative development of the heat supply				
The share of systems equipped with new highly effective operation technologies (%)	10	at least 40	at least 80	100
Efficiency of the heat supply				
Increase in energy efficiency of buildings (in % as compared to 2005)	5	at least 10	at least 30	at least 50
Heat losses (percentage of total heat production)	19	no more	no more	no more

		than 16	than 13	than 8–10

Table 1. Strategic Numbers of Heat Supply Development for Period up to 2030. (10)

The *TABLE 1* shows that the strategy of the heat supply development is divided in three phases and the objectives are defined with the minimum or the maximum value of the variable.

The heat supply cut-off rate with a downward trend, means a more effective heat supply, including controlled heat losses, use of the more effective products and reducing the need of heat. The *DIAGRAM 2* include the determined objectives of the heat supply cut-off rate of the Russian energy strategy up to 2030.

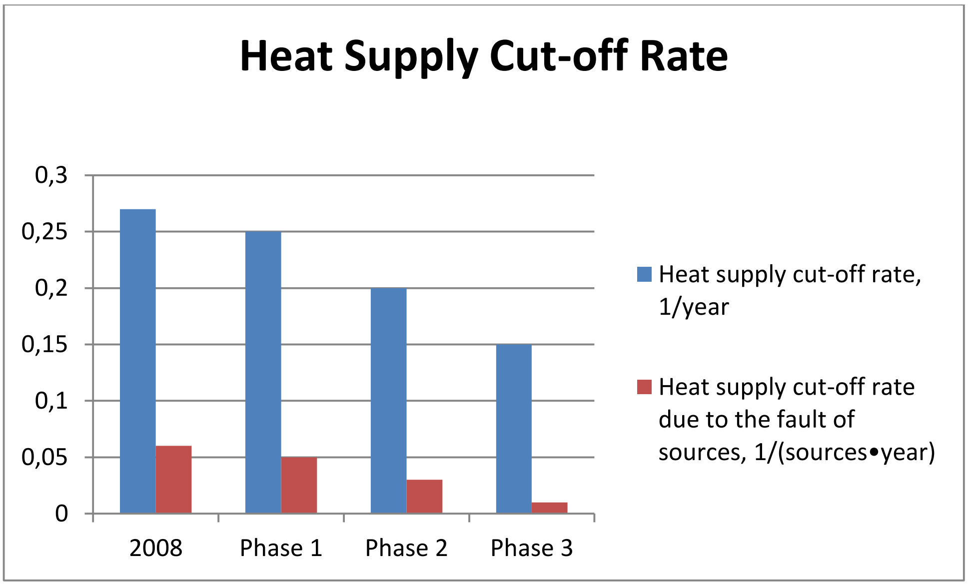


Diagram 2. Heat Supply Cut-off Rate. *(TABLE 1)*

The *DIAGRAM 3* shows that the Russian Energy ministry has plan to renovate over 80 percent of the heat supply network before the year 2030. Trend of the renovation is upward, which can be considered good sign for the investors.

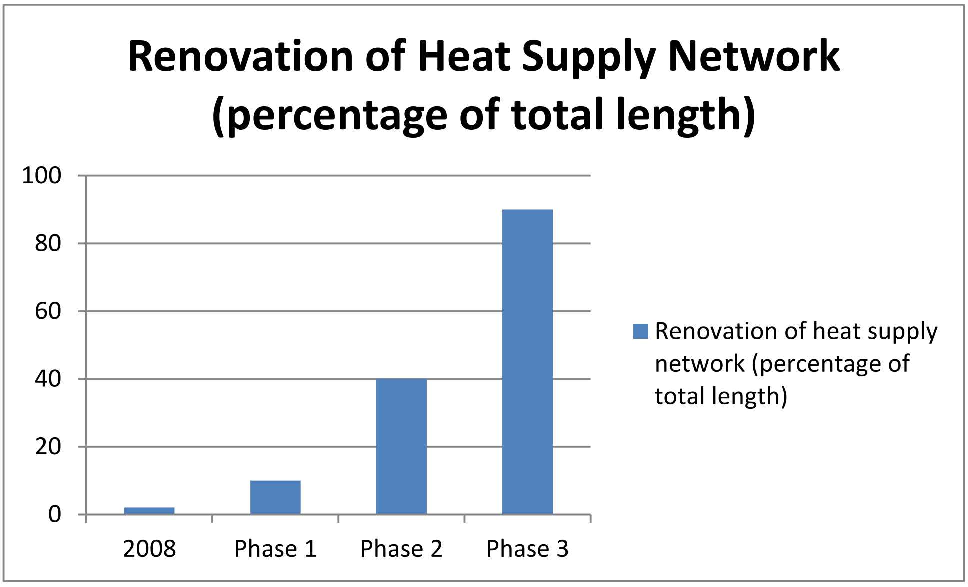


Diagram 3. Renovation of Heat Supply Network (percentage of total length). (TABLE 1)

The *DIAGRAM* 4 shows that the Russian Energy ministry is plan to improve the efficiency of the heat supply multiple ways before the year 2030. The efficiency of the heat supply is divided in to the four segments , whose amendment are demonstrated with form of the bar chart. For example, the planned percentage raise of the energy efficiency of buildings is from five percent in the year 2008 and up to 50 percent in to the year 2030. The energy efficiency of buildings can improve with more accurate control of the heat losses and accuracy of the imported heating power in relation to the needed heating power.

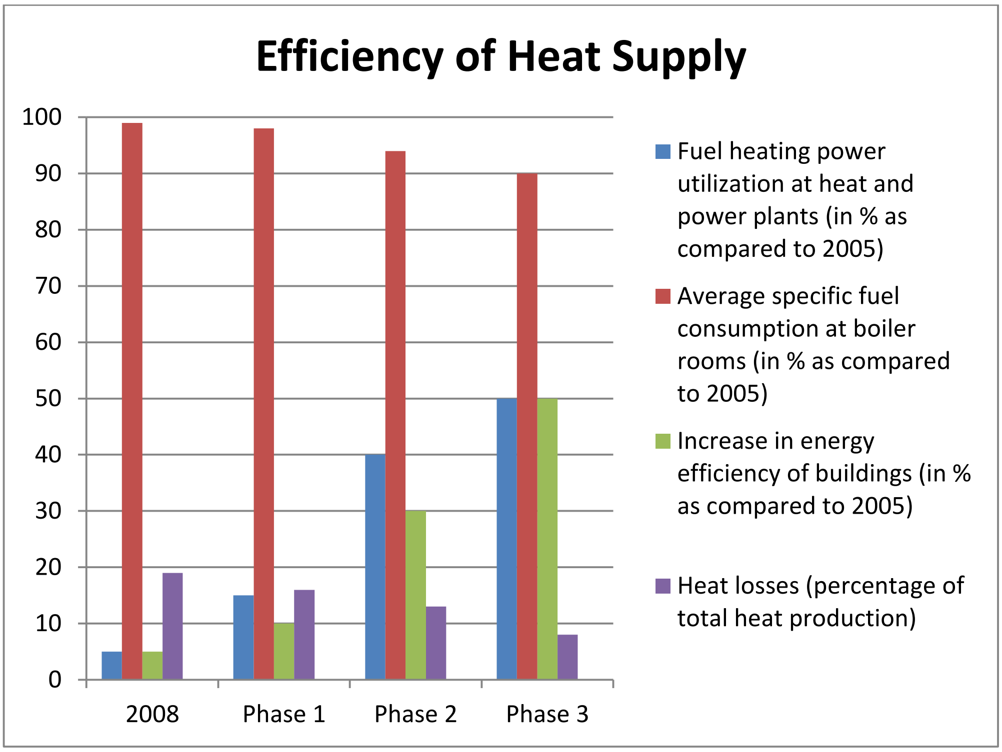


Diagram 4. Efficiency of Heat Supply. (TABLE 1)

The *DIAGRAM* 5 shows that the Russian Energy ministry is plan to improve a technology level of the heat supply before the year 2030. Correllation between variables in the *DIAGRAMS* 2,3,4 and the *DIAGRAM 5* is clearly seen.

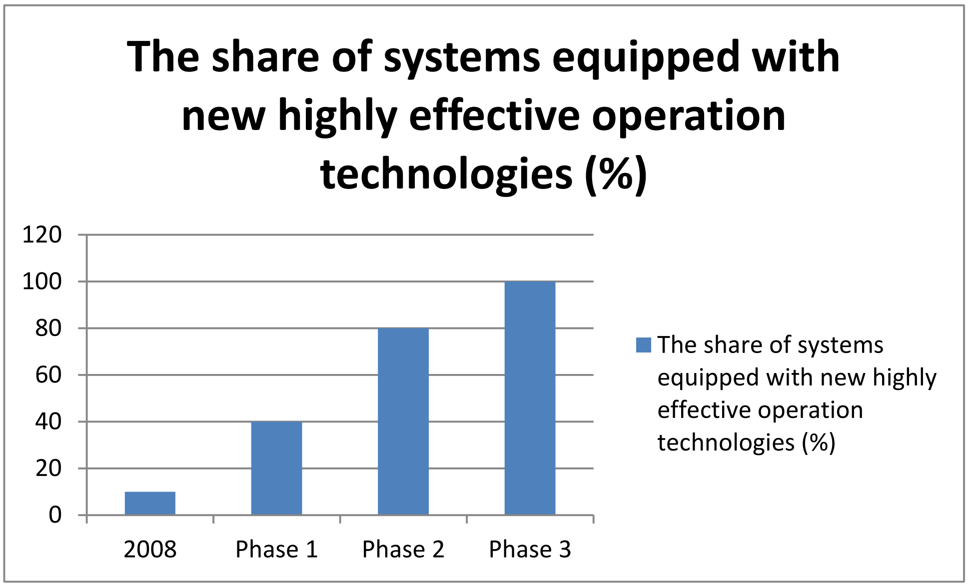


Diagram 5. Share of Systems Equipped with New Highly Effective Operation Technologies. (TABLE 1)

4.3 Potential Goals and Main Challenges

Energy saving potential in the district heating market could be called large in Russian Federation. Therefore one of the main goals of the Russian Government energy strategy is to entice more investors for the energy and the heating sector. It is challenging to make conditions enough favorable to investors, in order for the necessary financing to be accomplished (10, p. 112.) About 70 percent of the district heating infrastructure inevitably needs replacement or maintenance (3, p. (*iii*).)

One of the main challenges of Russia's district heating policy is tariffs (3, p. (*iii*).) Currently tariffs do not cover the full costs of district heating. It is economically sensible that they cover the full costs of the district heating (3, p. 5.)However, increasing a customer's heating costs can lead to socio-political issues.(3, p. 6.)Likewise, such as payment problems of customers.

"*About 73 percent of the Russian population—92 percent in urban areas and 20 percent in rural areas—depend on Russia's district heating sector, the largest in the world*" (3, p. 1.)

The lack of accurate information on actual consumption, losses, and volume supplied may be a problem in a planning process of the heating system. Adding metering devices in the heating system can be one solution to this problem. In the Russian Federation is a new law of energy, which requires metering, which leads to a process where municipalities and district heating companies are working to implement it (3, p. (*iii*)).

"Regulators, however, should ensure that metered data are incorporated into billing and planning. When norms and estimates are used, as has been customary in Russia, they often do not reflect the actual situation and fail to send the right signals to consumers and suppliers about their behavior and ability to improve efficiency of consuming or supplying heat. In situations when loss norms are lower than actual losses and metering is absent, losses tend to be passed to consumers and never addressed." (3, p. (*iii*)).

It is notable that because of the social policy in Russian Federation, the heat supplier is not allowed to cut off the heat supply if a customer has a problem with payment. There are differences compared with some neighbouring countries where the poorest people can be left without heat supply a result of un-payment of bills, for example in Romania (3, p. 10.)

4.4 Tariff Structures in Russian Federation

The Russian law of the heat supply provides four types of tariff structures.

- Cost-plus tariff, which is based on a fixed percentage payment of a profit build. Positive sides are logical and clear calculation methods. Negative sides are that it encourages on inefficient solutions and asymmetry of costs, which is depend on region.
- Return on investment. Investments costs are credited with a certain amount of time period, for example 3-5 year's period. Positive sides include guaranteed rate of return of investment and a drawback includes that it not encourage to improve energy efficiency.
- Tariff indexation. A central government sets yearly index at the tariffs. Positive sides are that it provides strong incentive to improve energy efficiency and cost savings and drawbacks are concerns that unexpected equipment failure costs cannot be covered completely with tariffs.

- Benchmarking. Positive sides are that it provides strong incentive to improve the energy efficiency and cost savings. Drawbacks are lack of knowledge from the heating data and adjust accuracy problems. There have not regulations ready yet in this tariff form (3, p. 7).

5. Ejector's Physical Theory

The ejector's principle operation is effected by the influence of pressure differences, which are created with the size and shape design of different parts of the ejector. Next chapters shows ejector's structure “exploded” and that opens the meaning of an individual component's purpose in the ejector.

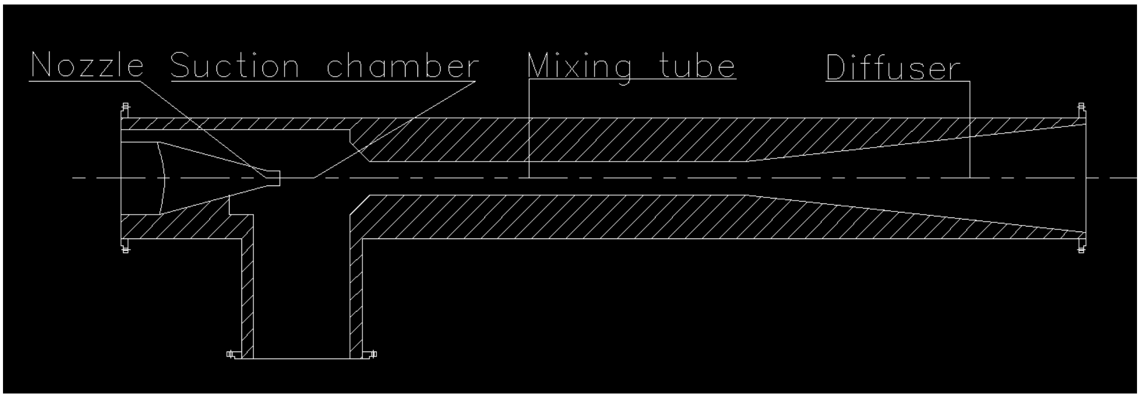


Figure 1. Ejector's structure.

5.1 Nozzle

The *FIGURE* 2 shows the nozzle's principle forms with two-dimensional shape and characteristics, which are needed in calculations related to the nozzle.

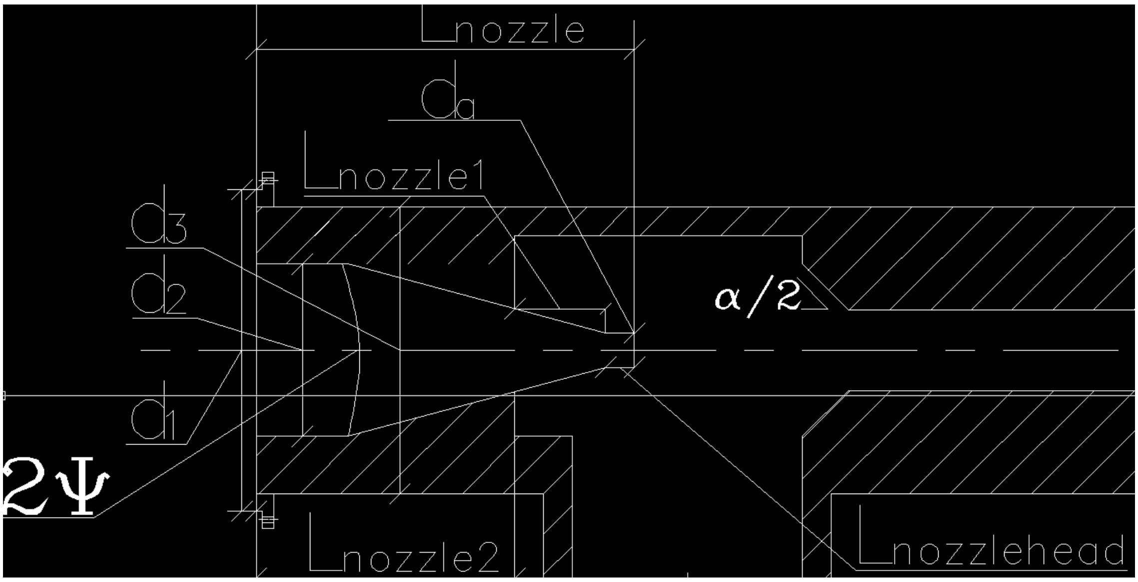


Figure 2. Nozzle.

The purpose of the nozzle is to control a primary flow velocity and give certain pressure loss.The nozzle also points a direction of the fluid shower's.

The nozzle is the smallest part of the ejector and it is also the most critical part of the ejector if it examined the point of view of adjustability of the ejector. In most cases, the only way to adjust a heating power individually in the ejector system is to chance the nozzle's size in the ejector (1). The calculations below help to calculate pressure losses over the nozzle. Besides that the *DIAGRAM* number 6 below shows the influence of a size and a shape to the factor K_0.

$$K_{nozzle} = \frac{f_{nozzle}(1-\beta^4)}{8\sin(\frac{\alpha}{2})} + \frac{f_{tip} l_{tip}}{d_{tip}} + 1 \qquad (1)$$

Where:

$\beta = d_{out} / d_{in}$

f = Fanning's friction factor

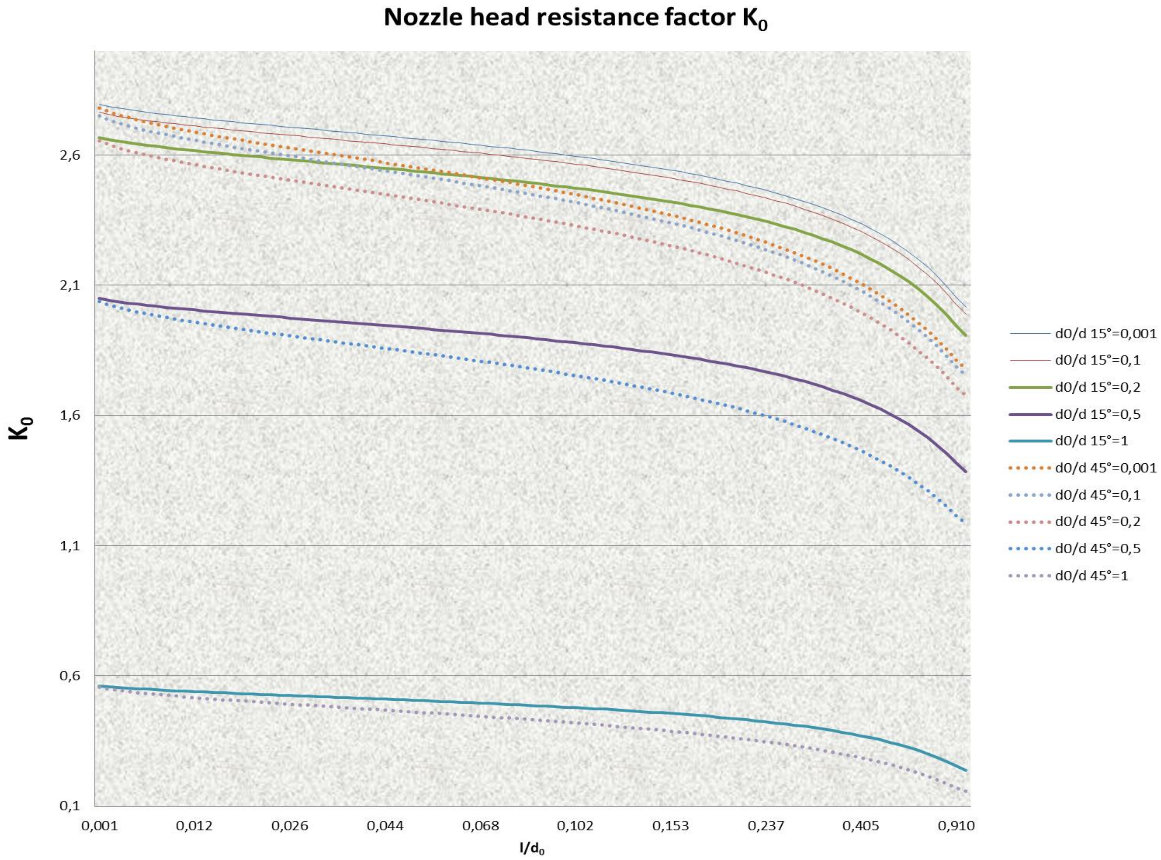


Diagram 6. Influence of shape of nozzle to factor K.

5.2 Suction Chamber

The purpose of the suction chamber is to lead suctioned fluid in to the chamber. The suction chamber's shape and size has not relatively big importance in the ejector when the fluid is water, because the molecule structure of the water does not have significant difference in between two inlets of fluids when the temperature difference is maximum of 55 degrees.

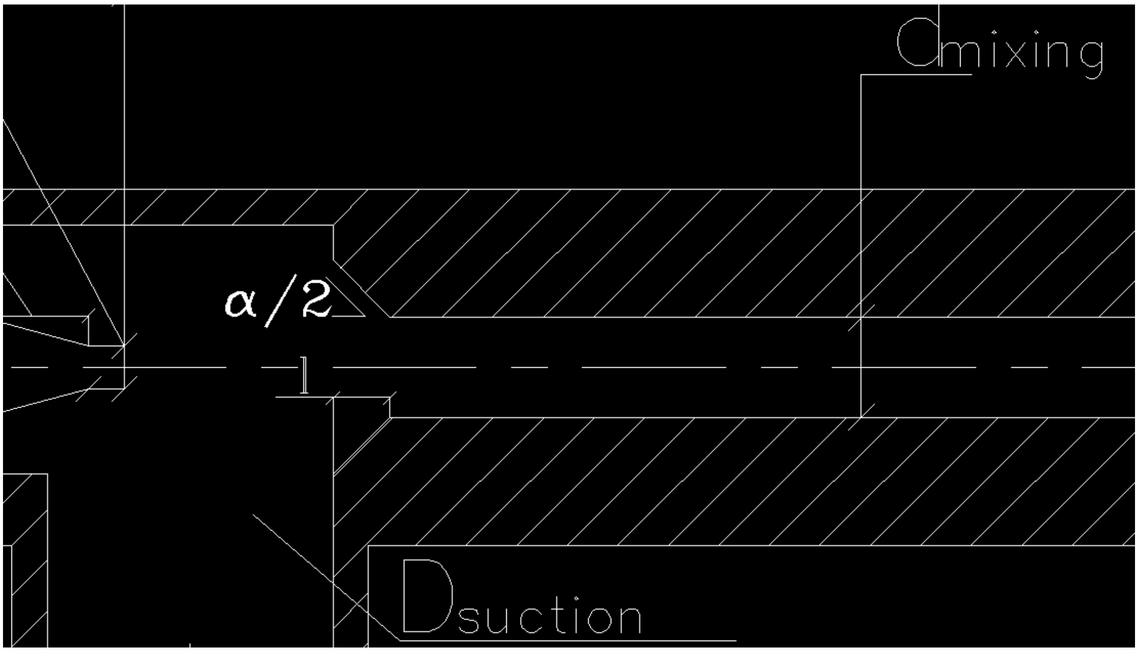


Figure 3. Suction chamber's and mixing tube interface.

K factor in the suction chamber's and mixing tube interface:

$$K_{interface} \sim 0.0696(1 + C_b \sin(\alpha/2) - 1)(1 - \beta^5)\lambda^2 + (\lambda - 1)^2 \quad (2)$$

Where:

α = degree of rounding in the suction camber's and the mixing tube interface.

The jet contraction coefficient:

$$\lambda = 1 + 0.0622\left(1 + C_b\left(\frac{\alpha}{180}\right)^{\frac{4}{5}} - 1\right)(1 - 0.215\beta^2 - 0.785\beta^5) \qquad (3)$$

Where:

The diameter ratio $\beta = d_{mixing}/D_{suction}$

Where:

$D_{suction}$= hydraulic diameter of the suction chamber

$$C_b = \left(1 - \frac{\alpha}{180}\right)\left(\frac{\alpha}{180}\right)^{\overline{\frac{1}{1+l_{rounding}/d_{mixing}}}} \qquad (4)$$

Where:

$l_{rounding}$= length of the rounding
d_{mixing}= diameter of the mixing chamber

5.3 Mixing Tube

The tube of mix purpose is to be a tube, where two fluids are mixed together as good as possible, lest an unmixed fluid do not distract the operation of the diffuser. Its length is determined in the standards of the hydro-ejector. (*TABLE* 2.) Pressure lost of the mixing tubes is calculated a similar way as calculations with the normal tubes.

5.4 Diffuser

The *FIGURE* 4 shows the diffuser's principled form at a two-dimensional shape and characteristics, which are needed in calculations related to the diffuser.

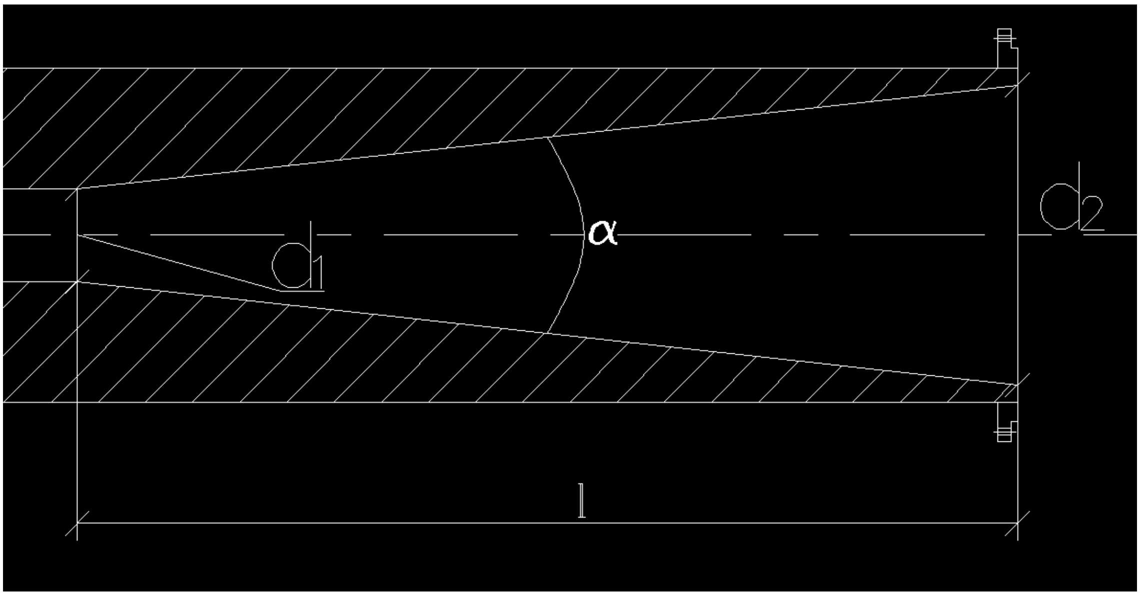


Figure 4. Diffuser.

The diffuser's purpose is to recover the fluid's static pressure with a minimal loss of a total pressure while reducing the flow velocity. Diffusers' angles in the hydro-ejectors are always below 20˚, consequently the review at the coefficient of the diffuser can be made with the calculations below.

The length of the diffuser:

$$l_{diff} = \frac{d_2 - d_1}{2\tan(\frac{\alpha}{2})} = d_1 \left(\frac{\frac{1}{\beta} - 1}{2\mathrm{l}/d_1} \right) \tag{5}$$

Where:

α = angle of the diffuser

d_x= diameter in section x

β = diameter ratio d_1/d_2

The angle of the diffuser:

$$\alpha_{diff} = 2\text{atan}\left(\frac{d_2-d_1}{2l}\right) = 2atan\left(\frac{\frac{1}{\beta}-1}{2l/d_1}\right) \quad (6)$$

The coefficient of the diffuser:

$$K_{diff} = \left(1-\eta_{diff}\right)(1-\beta^4) \quad (7)$$

Where:

η_{diff}= efficiency of the diffuser.

The coefficient factor of the diffuser:

$$K_{diff} = 8.30[tan(\alpha/2)]^{1.75}(1-\beta^2)^2 + \frac{f(1-\beta^4)}{8\sin(\frac{\alpha}{2})} \quad (8)$$

If requirements $0^\circ<\alpha< 20^\circ$ and $0> \beta< 1$ are filled.

Where:

α = angle of the diffuser

β = diameter ratio d_1/d_2

5.5 Example Calculations of Ejector Parts

Used size values on these calculations are defined with values of the ejector number 3, which can be found in the *TABLE* number 2.

The standard measurements of ejector					[m]											
nro.	L	L1	L2		Dmixing	Din	Ddiff,suction	Lnozzle	LnozzleA	LnozzleB	D1nozzle	D2nozzle	D3nozzle	Lnozzlehead	Dnozzle4	Dnozzle5
1	0,425	0,09	0,11	1	0,015	0,04	0,05	0,11	0,065	0,045	0,044	0,032	0,039	0,004	0,01666	0,01495
2	0,425	0,09	0,11	2	0,02	0,04	0,05	0,1	0,065	0,035	0,044	0,032	0,039	0,002	0,01666	0,01495
3	0,625	0,135	0,155	3	0,025	0,05	0,08	0,145	0,105	0,04	0,056	0,044	0,049	0,005	0,02644	0,02412
4	0,625	0,135	0,155	4	0,08	0,05	0,08	0,135	0,105	0,035	0,056	0,044	0,049	0,003	0,02644	0,02412
5	0,625	0,135	0,155	5	0,035	0,05	0,08	0,125	0,105	0,02	0,056	0,044	0,049	0,003	0,02644	0,02412
6	0,72	0,18	0,175	6	0,047	0,08	0,1	0,175	0,13	0,045	0,088	0,072	0,081	0,002	0,04191	0,03895
7	0,72	0,18	0,175	7	0,059	0,08	0,1	0,175	0,13	0,025	0,088	0,072	0,081	0,002	0,04191	0,03895

Table 2. Standard measurements of the ejectors. (1)

5.5.1 Design of Nozzle

It is assumed that the mass flow through the nozzle is 1.3 kg/s and the nozzle's output diameter is 7mm.

Reynolds number in the nozzle:

$$Re_{nozzle} = \frac{\dot{m}_x D_h}{A_{cannel}\mu_x} = \frac{\frac{1{,}3kg}{s} * 0.007m}{3.85*10^{-5}m^2*0.00018Pa*s} \sim 1390000 \quad (9)$$

With the *FIGURE* 5, a friction factor (*f*) of the nozzle head and its tip can be obtained. It is assumed that the nozzle surface is smooth.

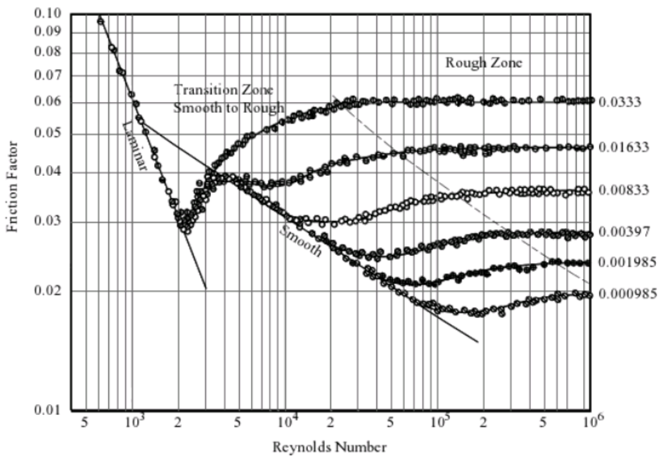


Figure 5. Friction Factor Relation to the Reynolds number. (11)

The contribution factor of the nozzle:

$$K_{nozzle} = \frac{f_{nozzle}(1-\beta^4)}{8\sin(\frac{\alpha}{2})} + \frac{f_{tip}l_{tip}}{d_{tip}} + 1 = \frac{0{,}017(1-\beta^4)}{8\sin(\frac{\alpha}{2})} + \frac{f_{tip}l_{tip}}{d_{tip}} + 1 \sim 1.03 \qquad (10)$$

Where:

$\beta = d_{out} / d_{in}$=7mm/44mm=0.16

$f_{nozzle} = f_{tip}$ ~0.017

l_{tip}= 3mm

d_{tip}= 7mm

The angle of the nozzle:

$$\alpha_{nozzle} = angle\ of the\ nozzle =$$
$$2\text{atan}\left(\frac{d_{in}-d_{out}}{2l}\right) = 2\text{atan}\left(\frac{0{,}044m-0{,}07m}{2*0{,}135\text{m}}\right) = 15.6° \qquad (11)$$

The pressure loss over the nozzle:

$$H = f\frac{L}{D}\frac{v^2}{2g} \sim 1.77m = 177kPa \tag{12}$$

Where:

$$K_{nozzle} = f\frac{L}{D} \tag{13}$$

The velocity in the nozzle:

$$v = \frac{\dot{m}}{\rho A} = \frac{1.3\ kg/s}{\frac{1000kg}{m^3} * \pi 0.0035^2} \sim 33.8\ m/s \tag{14}$$

Where:

g =gravity = 9.81 m/s^2

$$A = \pi {r_{nozzlehead}}^2 = \pi 0.0035^2 \tag{15}$$

ρ = density [kg/m^3]

5.5.2 Design of Suction Chamber

The suction chamber size parameters are assumed to correspond ejector number 3 values, which can be found in the *TABLE* 2.

Themixing ratio ω is assumed as 2.53. That will lead:

$$\dot{m}_{suction} = \omega * \dot{m}_1 = 2.53 * 1.3\frac{kg}{s} \sim 3.29kg/s \tag{16}$$

K factor in the suction chamber and mixing tube interface:

$$K_{interface} \sim 0.0696\left(1 + C_b \sin(\frac{\alpha}{2}\right) - 1)(1 - \beta^5)\lambda^2 + (\lambda - 1)^2 =$$
$$0.0696(1 + 0.31\sin(74/2) - 1)(1 - 0.08^5)0.08^2 + (0.08 - 1)^2 \sim 0.85 \tag{17}$$

Where:

α = degree of rounding in the suction camber and the mixing tube interface.

The jet contraction coefficient:

$$\lambda = 1 + 0.0622\left(1 + C_b\left(\frac{\alpha}{180}\right)^{\frac{4}{5}} - 1\right)(1 - 0.215\beta^2 - 0.785\beta^5) =$$

$$1 + 0.0622\left(1 + 0.31\left(\frac{74}{180}\right)^{\frac{4}{5}} - 1\right)(1 - 0.215 * 0.08^2 - 0.785 * 0.08^5) \sim 1.01$$

(18)

Where:

The diameter ratio $\beta = d_{mixing}/D_{suction} = 0.08$

Where:

$D_{suction}$= the hydraulic diameter of the suction chamber

$$= \frac{4A}{U} = \frac{4*2\pi rh}{circuit} = \frac{4*2\pi*0.025*0.152}{2*0.025+2*0.152} \sim 0.27m \quad (19)$$

$$C_b = \left(1 - \frac{\alpha}{180}\right)\left(\frac{\alpha}{180}\right)^{\frac{1}{1+l_{rounding}/d_{mixing}}} = \left(1 - \frac{74}{180}\right)\left(\frac{74}{180}\right)^{\frac{1}{1+0.01/0.025}} \sim 0.31$$

(20)

Where:

$l_{rounding}$ = 0.01m

d_{mixing} = 0.025m

$$\alpha_{rounding} = 2\,\mathrm{atan}\left(\frac{d_2 - d_1}{2l}\right) = 2atan\left(\frac{0.05-0.035}{2*0.01}\right) \sim 74° \quad (21)$$

The pressure loss over the interface:

$$H = f\frac{L}{D}\frac{v^2}{2g} \sim 0{,}24m = 24kPa \quad (22)$$

Where:

$$K_{interface} = f\frac{L}{D} \tag{23}$$

$$v = velocity = \frac{\dot{m}}{\rho A} = \frac{1.3\ kg/s + 3.29 kg/s}{\frac{1000kg}{m^3} * \pi 0.025^2} \sim 2.34\ m/s \tag{24}$$

Where:

g =gravity = 9.81 m/s^2

$$A = \pi {r_{mixing\ chamber}}^2 = \pi 0.025^2 \tag{25}$$

ρ = density [kg/m^3]

5.5.3 Design of Mixing Chamber

The mixing chamber size parameters are assumed to correspond ejector number 3 values, which can be found in the *TABLE* 2.

$$\dot{m}_{mixing} = \dot{m}_1 + \dot{m}_2 = (1.3 + 3.29) kg/s = 4.59 kg/s \tag{26}$$

$$Re_{mixing} = \frac{\dot{m}_x D_h}{A_{cannel} \mu_x} = \frac{\frac{4.59kg}{s} * 0.025m}{4.91 * 10^{-4} m^2 * 0.00018 Pa * s} \sim 1300000 \tag{27}$$

The pressure loss over the mixing chamber:

$$H = f\frac{L}{D}\frac{v^2}{2g} \sim 0.03m = 3kPa \tag{28}$$

Where:

L= 0.15m

D = 0.025m

f_{mixing} ~0.017 , can be obtained in the *FIGURE* 6.

Velocity:

$$v \ = \ velocity \ = \frac{\dot{m}}{\rho A} = \frac{4.59 \ kg/s}{\frac{1000kg}{m^3} * \pi 0.025^2} \sim 2.34 \ m/s \tag{29}$$

Where:

g =gravity = 9.81 m/s^2

$$A = \pi {r_{nozzlehead}}^2 = \pi 0.0035^2 \tag{30}$$

ρ = density [kg/m^3]

5.5.4 Design of Diffuser

The diffuser size parameters are assumed to correspond ejector number 3 values, which can be found in the *TABLE* 2.

.

The mass flow of the diffuser:

$$\dot{m}_{diffucer} = \dot{m}_1 + \dot{m}_2 = 4.59kg/s \tag{31}$$

The alpha degree of the diffuser:

$$\alpha_{diff} = 2\mathrm{atan}\left(\frac{d_2 - d_1}{2l}\right) = 2\mathrm{atan}\left(\frac{0.08 - 0.025}{2*0.3}\right) \sim 10.5° \tag{32}$$

Where the length of the diffuser is assumed as 0.3m.

Dimensions d_1=0.025m and d_2=0.08m can be found in the *TABLE 2*.

The coefficient of the diffuser:

$$K_{diff} = (1 - \eta_{diff})(1 - \beta^4) \tag{33}$$

Where:

η_{diff}= efficiency of the diffuser.

The coefficient factor of the diffuser:

$$K_{diff} = 8.30[tan(\alpha/2)]^{1.75}(1-\beta^2)^2 + \frac{f(1-\beta^4)}{8\sin(\frac{\alpha}{2})} =$$

$$8.30[tan(10.5/2)]^{1.75}(1-0.31^2)^2 + \frac{0.017(1-0.31^4)}{8\sin(\frac{10.5}{2})} \sim 0.13 \quad (34)$$

If requirements 0˚<α< 20˚ and 0> β< 1 are filled.

Where:

α = angle of the diffuser

β = diameter ration d_1/d_2~ 0.31

The pressure loss over the diffuser:

$$H = f\frac{L}{D}\frac{v^2}{2g} \sim 0.001m = 1kPa \quad (35)$$

Where:

$$K_{diffuser} = f\frac{L}{D} \quad (36)$$

And:

$$v_{average} = \frac{\left(\frac{\dot{m}}{\rho A_1} + \frac{\dot{m}}{\rho A_1}\right)}{2} =$$

$$\left(\frac{1.3\frac{kg}{s} + \frac{3.29kg}{s}}{\frac{1000kg}{m^3} * \pi 0.025^2} + \frac{1.3\frac{kg}{s} + \frac{3.29kg}{s}}{\frac{1000kg}{m^3} * \pi 0.080^2}\right)/2 \sim 1.18\, m/s \quad (37)$$

Where:

g =gravity = 9.81 m/s^2

$A = \pi {r_x}^2$

ρ = density [kg/m^3]

5.5.5 Pressure Loss Over All Parts

The pressure loss over all parts is calculated with a summing method, which is determined below.

The pressure loss over all parts:

$$H_{tot} = H_{nozzle} + H_{suction} H_{mixing} + H_{diffuser} = (177 + 24 + 3 + 1) kPa = 205\ kPa \quad (38)$$

6. Ejector's Measuring Methods in Russia

The *FIGURE* 6 shows the ejector's structure at a two-dimensional shape and characteristics, which are needed in calculations related to the ejector.

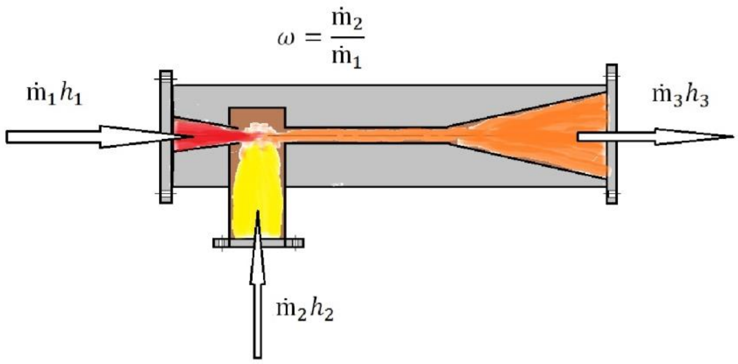


Figure 6. Hydro-ejector's structure.

6.1 Common

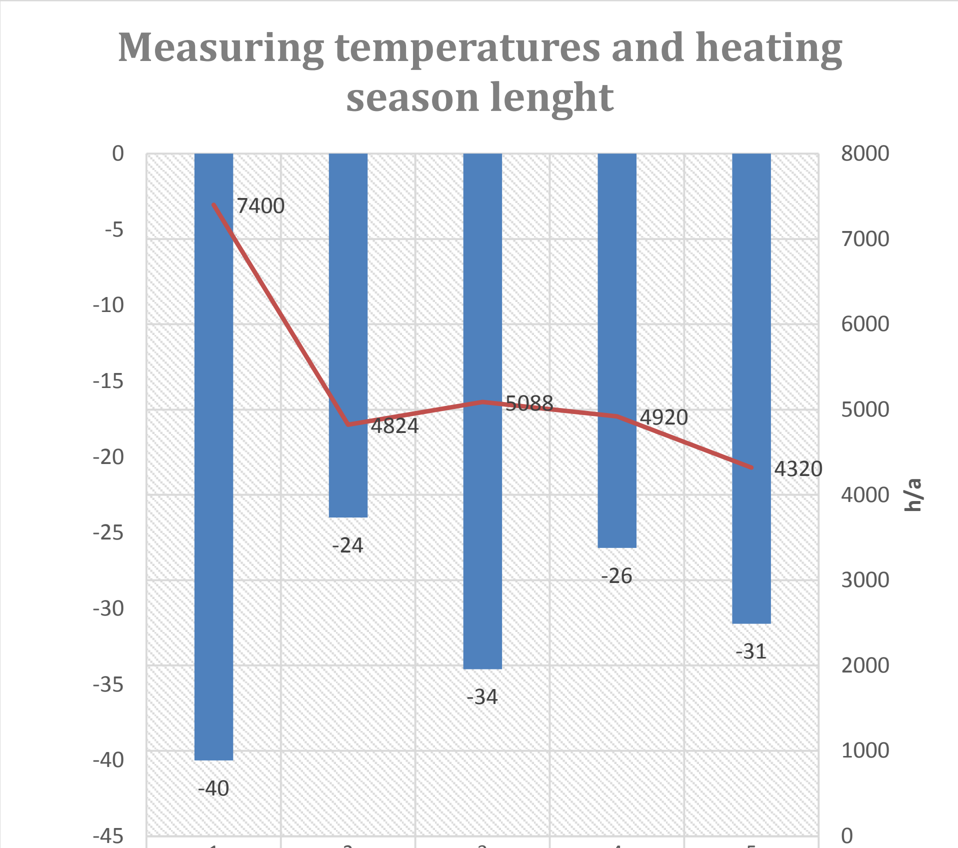
There are many different variations how to calculate the ejector's size and pressure drop and this is only one style, which is combined from multiple references (1,6,9,11,14.) The reason for ending up with this solution was the easy comprehensibility of the calculations and best fit in this contest.

6.2 About Russian heating norms

The maximum temperature of the district heating water is 200°C and its pressure is up to 2.5 MPa. With steam the equal values are 440 °C and 6.3 MPa (16, c. 1.) In Russian study books (1, p. 54 &9, p. 330.), branches before the ejector are planned with 150 °C temperature and with 1,6 MPa maximum pressure, but in a real situation designing temperature is defined by the local heat supplier. The heating network is built using a two- piped system (95-70) °C or singular pipe system (105-70) °C. In a residential buildings the maximum domestic hot water temperature is 95 °C (6, p. 26.) In special buildings such as hospitals, retirement homes, kindergartens and etc. the designing temperature is lower than residential buildings' designing temperature (85...65) °C (16.)

6.3 Example of Regional Measuring Temperatures and Heating Season Length

The *DIAGRAM 7* below has the examples of measuring temperatures and the length of the heating season in example cities. Values of the measuring temperature are used, for example, in the thermal loss calculations and values of the heating season length are used in energy calculations.



	1	2	3	4	5
°C	-40	-24	-34	-26	-31
h/a	7400	4824	5088	4920	4320

Diagram 7. Measuring Temperatures and Heating Season Length in Different Cities in Russia.

Where:

1 =Alatur, 2 = Vladivostok, 3 =Blagoveshensk, 4 = Moscow and 5 = Khabarovsk (6, p. 22.)

6.4 Necessarry Preminilary Information

The measurement of the hydro-ejector starts with knowledge of a preliminary information. The preliminary information includes the Russian heating norms and standards (15,16), which gives the maximum levels of the pressures and temperatures, material qualifications and safety rules.

Some measurement values are also needed, including the inlet measurement temperature, which is obtainable from the local authorities. In the measurement calculations of the ejector, depending on a calculation method, is the operation ratio, which is usually 2.2 or 2.53.

When determine the mixing chamber size, the secondary side pressure drop has to be known, which is assumed to be in next chapter calculations between 10 and 20 kPa. Also it is good to know, either the total calculated thermal power or the suction mass flow.

6.5 Calculations of Ejector

The operation ratio:

$$u = 1.15\frac{T_1 - T_3}{T_3 - T_2} \quad (39)$$

The *FIGURE* 7 shows the nozzle's size of the ejector, when variables are the inlet temperature and the mass velocity.

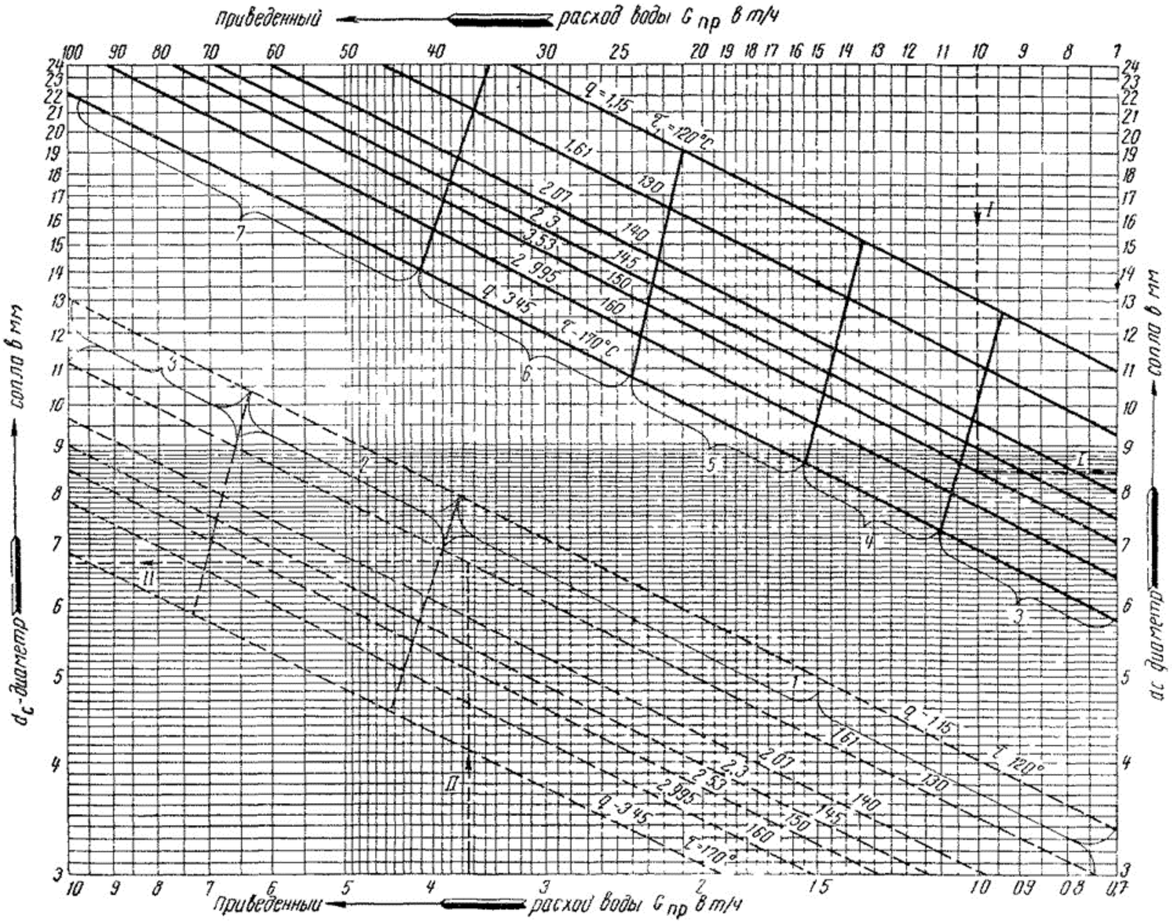


Figure 7. Diameter of Nozzle. (9)

The mixing ratio:

$$\omega = \frac{\dot{m}_2}{\dot{m}_1} \tag{40}$$

Where $\dot{m}$ is the fluid mass flow.

The energy balance:

$$\dot{m}_1 h_{1+} \dot{m}_2 h_2 = \dot{m}_3 h_3 \tag{41}$$

Where:

h = the specific enthalpy.

The combine of educations 2 and 3:

$$\omega = \frac{h_1 - h_3}{h_3 - h_2} \approx \frac{T_1 - T_3}{T_3 - T_2} \tag{42}$$

The temperature level of heating circle inlet:

$$T_3 = \frac{T_1 + \omega T_2}{1 + \omega} \tag{43}$$

The mass flow balance:

$$\dot{\mathrm{m}}_1 + \dot{\mathrm{m}}_2 = \dot{\mathrm{m}}_3 \tag{44}$$

The mass flow in the hydro-ejector's inlet:

$$\dot{\mathrm{m}}_1 = \mu_{nozzle} A_{nozzle} \sqrt{2\rho(P_{in} - P_{nozzle})} \tag{45}$$

Where μ_{nozzle} is the nozzle factor, which usually is between values 0.97-0.99. Furthermore P_{in} is the pressure level before the nozzle and the P_{nozzle} is the pressure level beyond the nozzle.– (12, p. 23.) It is notable,that calculations with methods obtained from different references can give a different nozzle factor. For example in the calculation (10) the nozzle factor value 1.03 is obtained.

The mass flow in the hydro-ejector's suction chambers inlet:

$$\dot{\mathrm{m}}_2 = \mu_{suction} A_{suction} \sqrt{2\rho(P_{suction} - P_{mixing\ chamer})} \tag{46}$$

Where:

$\mu_{suction}$= the suction factor, which can be assumed as value 1.0, because in the hydro-ejector system, where fluid is the water, the meaning of suction chamber shape and designing got only minor influence on the fluid characteristics.

$P_{suction}$= the pressure level before suction chamber
$P_{mixing\ chamber}$= the pressure level before mixing chamber.

The factor μ consist of:

$$\mu_x = \varphi_x \alpha_x \tag{47}$$

Where φ_x= velocity factor and α_x = constriction factor.

The diameter of the mixing chamber:

$$d_c = 15{,}5 \frac{{G_a}^{0,5}[\frac{t}{h}]}{{\Delta P_{secondary}}^{0,25}[kPa]} [mm] \tag{48}$$

Where:

G_a= the mass velocity between the suction chamber's input and the diffuser's output $\Delta P_{secondary}$= the pressure drop in the secondary side.

The mass velocity between the suction chamber's input and the diffuser's output:

$$G_a = \frac{0{,}86 * \emptyset[W]}{(T_3 - T_2)[°C] * 1000} [\frac{t}{h}] \tag{49}$$

Where:

Ø= the thermal power in the heating circle.

The diameter of the nozzle:

$$d_a = \frac{d_{c,tablevalue}}{1+u} [mm] \tag{50}$$

Where:

$d_{c,tablevalue}$=the mixing chamber table value

The mixing chamber's table value can be found in the table 2. A value, which will be selected, is a following bigger value than the calculated value.

u= the operating ratio

The pressure loss of the hydro-ejector:

$$\Delta p_{ejector} = 0.64 \frac{{G_{tot}}^2[\frac{t}{h}]}{{d_{nozzle}}^4[cm]}[kPa] \tag{51}$$

Where:

G_{tot}= the total mass velocity of ejectors outlet.

A pressure loss over the ejector can also calculated with the formula:

$$\Delta p_{ejector} = 1.5 + (1+u)^2 \Delta p_{secondary}[kPa] \tag{52}$$

6.6 Example Calculation of Ejector

Next calculations are try to simulate a typical situation with the open district heating system in Russian Federation. The subject is ten flat residential building which is attached to the centralized district heating facility with the hydro-ejector system. The calculated heating power($\emptyset$) in that building is 180kW, the inlet temperature (T_1) , which is gained from heating facility, is 150 °C. Moreover, the residential building's inlet temperature (T_3) is determined as 95°C and suction fluid's temperature is 70°C. The pressure loss on the secondary side is determined as 12 Kpa.

It is notable that the inlet measurement temperature has a specific value, depending on which region the heating system is located. Information at the regional inlet temperature can be gained from the local authorities.

The operation ratio:

When the all known characters are placed in the calculation, figured out that u is only unknown.

$$u = 1.15\frac{150\ ℃ - 95℃}{95℃ - 70℃} \sim 2.53 \tag{53}$$

Other way around gives the temperature in inlet of the suction chamber.

$$T_2[℃] = -1 * (1.15 * \frac{150\ ℃ - 95℃}{2.53} - 95℃) = 70℃ \tag{54}$$

After resolving the operation ratio and needed temperature values, a further step is to figure out mass velocity between the suction chamber's input and the diffuser's output.

The mass velocity in the suction chamber's input:

$$G_2 = \frac{0.86 * \emptyset[W]}{(T_3 - T_2)[℃] * 1000} [\frac{t}{h}]$$

$$G_2 = \frac{0.86 * 180000[W]}{(90 - 75)[℃] * 1000} \left[\frac{t}{h}\right] = 6.192 \left[\frac{t}{h}\right] \sim \left[\frac{m^3}{h}\right] \tag{56}$$

This leads to the mass flow of the suction chamber:

$$\dot{m}_{suction} = 6.192 * \frac{1000}{3600} = 1.72kg/s \quad (57)$$

It is notable that typically the mass velocity is ton per hour in the Russian study books' characteristics.

There is only two significant variables in the ejector's sizing process. The diameter of the mixing chamber and the diameter of the nozzle. Other variables are determined in standards.

The diameter of the mixing chamber:

$$d_c = 15.5 \frac{{G_2}^{0.5} [\frac{t}{h}]}{{\Delta P_{secondary}}^{0.25} [kPa]} [mm]$$

$$d_c = 15.5 \frac{6.192^{0.5} [\frac{t}{h}]}{12^{0.25} [kPa]} [mm] = 20.72 \quad (58)$$

Which leads to choose from table (Table2.) value's nro.3 (0.025m)

The diameter of the nozzle:

$$d_a = \frac{d_{c,tablevalue}}{1+u} [mm]$$

$$d_a = \frac{25}{1+2.53} = 7.08\ [mm] \quad (59)$$

The total pressure loss over the ejector:

$$\Delta p_{ejector} = 0.64 \frac{{G_{ejector}}^2 [\frac{t}{h}]}{{d_{nozzle}}^4 [cm]} [\mathrm{kPa}]$$

$$\Delta p_{ejector} = 0.64 \frac{8.64^2 [\frac{t}{h}]}{0.708^4 [cm]} \sim 190\ kPa \quad (60)$$

Where:

$$G_{tot} = G_1 + G_2 = (6.19 + 2.45)[\frac{t}{h}] = 8.64[\frac{t}{h}] \tag{61}$$

Where:

$$G_1 = \ G_{suction}/\ u = 6.192/2.53 \sim 2.45[\frac{t}{h}] \tag{62}$$

A pressure loss over the ejector alternatively can determined with formula:

$$\Delta p_{ejector} = 1.5 + (1 + u)^2 \Delta p_{secondary}$$
$$= 1.5 + (1 + 2.53)^2 12kPa \sim 225kPa \tag{63}$$

It is notable, that there is a slight difference between the results of the pressure loss calculations. With that knowledge it can be extrapolated that in order to obtain more accurate information about the pressure lost over the ejector it should be measured with pressure difference measurement equipment.

7. Plate heat exchanger

The *FIGURE* 8 is a three dimensional description of the plate heat exchanger, implemented by the manufacturer Danfoss.

Figure 8. Plate Heat Exchanger (PHE).(17)

7.1 Introduction

The purpose of this part of the design guide is to introduce the reader on mathematics calculations, thermo-hydraulic design, material choices, future problems and benefits of the plate heat exchanger's (PHE's). This can also give great arguments for questions, "Why choose the PHE?" "Why to replace the open hydro-ejector system with the closed PHE system?"

Answers of those questions are simple. The PHE is relatively cheap to manufacture (13, p. 21.), it is reliable, adjustability of the system is relatively accurate and accuracy of the thermal conditions is high compared to the "open system". In the case of "hydraulic difference", which means a leakage or damage on the secondary side of the district heating network, the damage or the leakage stays local and thus does not affect the whole network's ability to function. District heating water's quality parameters are easier to control in the closed system.

There are also some disadvantages, concerns or arguments against the PHE system. Electricity is used in the heat exchanger system, which can be a problem in areas, where the electricity contribution is unstable. A possible drawback is that the heat exchanger may clog after a period of time if fluid includes occlusive ingredients. In some areas of Russian Federation electric breakdowns are common and they must be considered when designing PHE- system in that kind of area. Important buildings, for example hospitals are equipped with automatic spare energy systems, whereas it is rare to have any spare systems in residential buildings. Those disadvantages can be avoided with good preparation and planning.

7.2 Material

When choosing a material for the PHE, first it must be ensured that it is suitable for the type of physical construction. Moreover, the temperature ranges of the fluid, possible corrosion effect of the fluid and the material contact must be known before the material choose. Material costs also got a big role in a decision make for the material and it must be optimized with the costs of whole process.

"The gaskets, being typically manufactured of a suitable rubber, are the main factor limiting the temperature and pressure ranges possible to achieve with gasketed PHEs" (13, p. 22.) The gaskets, which are manufactured from the rubber, needs regular service or substitution. Alternatively, gaskets manufactured with method of braze are service free.

Plates of PHE are usually chosen as stainless steel types 304 or 316 or aluminium or titanium, depending on the material availability, cost and manufacturing equipment. The carbon steel is usually an unsuitable option, because its vulnerability to corrosion. Plates are typically only 0.4 to 1.4 mm thin, so there is no space for corrosion (13, p. 22-23).

7.3 Plate Heat Exchanger Design

The plate heat exchanger design accuracy will increase possibility to avoid unnecessary material costs and to get the best possible thermal efficiency.

7.3.1Flow- and Thermodynamic Dimensioning

For hot fluid the heat transfer rate is:

$$q = \dot{m}_1 Cp_1 (T_{1in} - T_{1out}) \tag{64}$$

Where:

$\dot{m}_1$= the mass flow rate for the hot fluid

And:

Cp_1= the specific heat for the hot fluid.

The temperature subscripts are displayed in the flow demonstration *FIGURES 9 and 10* below.

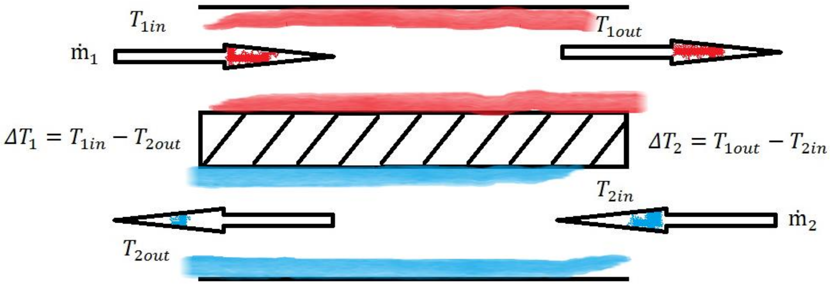


Figure 9. Principle of the counter flow in PHE.

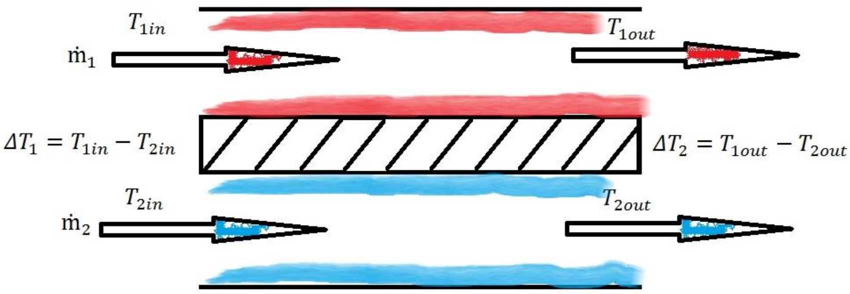


Figure 10. Principle of the parallel flow in PHE.

For cold fluid the heat transfer rate is:

$$q = \dot{m}_2 Cp_2 (T_{2out} - T_{2in}) \tag{65}$$

Where:

$\dot{m}_2$= the mass flow rate for the cold fluid

Cp_2= the specific heat for the cold fluid.

The temperature subscripts are displayed in the flow demonstration in the *FIGURE* 10.

The common heat transfer rate is:

$$q = UAF\Delta T_{lm} \tag{66}$$

Where:

U = the overall heat transfer coefficient

A = the heat transfer surface area at the hot or cold side

F = the correction factor, depending on the flow arrangements.

It is notable that:

$$UA = U_1A_1 = U_2A_2 \tag{67}$$

The logarithmic temperature is:

$$T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{68}$$

Where:

$\Delta T_1 = T_{1in} - T_{2in}$ in parallel flow (69)

And $\Delta T_2 = T_{1out} - T_{2out}$

Where:

$\Delta T_1 = T_{1in} - T_{2out}$ in counter flow (70)

And $\Delta T_2 = T_{1out} - T_{2in}$ (71)

The heat transfer area is:

$$A_1 = P_1 L \tag{72}$$

Or

$$A_2 = P_2 L \tag{73}$$

Where: P_x = the perimeter of hot or cold fluid channel.

The steady heat balance to the control volume for the hot fluid provides:

$$\dot{m}_1 Cp_1 T_1 - \dot{m}_1 Cp_1 (T_1 + dT_1) - dq = 0 \quad \text{(Parallel flow)} \tag{74}$$

This leads:

$$\frac{d_q}{\dot{m}_1 Cp_1} = -dT_1 \tag{75}$$

The steady heat balance to the control volume for the cold fluid provides:

$$\dot{m}_2 Cp_2 T_2 - \dot{m}_1 Cp_1 (T_2 + dT_2) - dq = 0 \quad \text{(parrallel flow)} \tag{76}$$

This leads:

$$\frac{d_q}{\dot{m}_2 Cp_2} = -dT_2 \tag{77}$$

Combining educations (75) and (76):

$$d_q \left(\frac{1}{\dot{m}_1 Cp_1} + \frac{1}{\dot{m}_2 Cp_2}\right) = -dT_1 + dT_2 = -d(T_1 - T_2) \tag{78}$$

The local differential heat transfer rate is:

$$d_q = \frac{T_1 - T_2}{\frac{1}{UdA}} = UdA(T_1 - T_2) \tag{79}$$

Inserting equation (78) into equation (79):

$$UdA(T_1 - T_2)\left(\frac{1}{\dot{m}_1 Cp_1} + \frac{1}{\dot{m}_2 Cp_2}\right) = -d(T_1 - T_2) \tag{80}$$

Rearranging this gives:

$$\frac{-d(T_1 - T_2)}{T_1 - T_2} = U\left(\frac{1}{\dot{m}_1 Cp_1} + \frac{1}{\dot{m}_2 Cp_2}\right) dA \tag{81}$$

The inlet temperature difference = $T_{1in} - T_{2in}$

The outlet temperature difference = $T_{1out} - T_{2out}$.

Integrating the both sides of Equation (81) gives:

$$\int_{\mathrm{T1in-T2in}}^{\mathrm{T1out-T2out}} \frac{-d(T_1 - T_2)}{T_1 - T_2} = U\left(\frac{1}{\dot{m}_1 Cp_1} + \frac{1}{\dot{m}_2 Cp_2}\right)\int dA \tag{82}$$

Which yields:

$$-ln\left(\frac{T_{1out} - T_{2out}}{T_{1in} - T_{2in}}\right) = U\left(\frac{1}{\dot{m}_1 Cp_1} + \frac{1}{\dot{m}_2 Cp_2}\right) A \tag{83}$$

Equations (82) and (83) are rearranged for the inverse of the product of the mass flow rate and the specific heat, which are substituted into equation (84)

$$-ln\left(\frac{T_{1out} - T_{2out}}{T_{1in} - T_{2in}}\right) = UA\left(\frac{T_{1in} - T_{1out}}{q} + \frac{T_{2out} - T_{2in}}{q}\right) = UA\left(\frac{(T_{1in} - T_{2in}) - (T_{1out} - T_{2out})}{q}\right) \tag{84}$$

Solving for q provides:

$$q = UA\frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{85}$$

Where:

$$\Delta T_1 = T_{1in} - T_{2in} \text{ in parallel flow} \tag{86}$$

And:

$$\Delta T_2 = T_{1out} - T_{2out} \tag{87}$$

Where:

$$\Delta T_1 = T_{1in} - T_{2out} \text{ in counter flow} \tag{88}$$

And:

$$\Delta T_2 = T_{1out} - T_{2in} \tag{89}$$

The noncircular diameter D_h (hydraulic diameter):

$$D_h = \frac{4A_c}{P_{wetted}} = \frac{4A_cL}{P_{heated}L} = \frac{4A_cL}{A_t} \tag{90}$$

Where:

P_{wetted}= the wetted perimeter

A_t=the total heat transfer area

L =the length of the channel

The mass velocity G is defined as:

$$G = \rho w \tag{91}$$

The mass flow rate ṁ is defined as:

$$\dot{m}_x = G_x A_{cannel} \tag{92}$$

The Reynolds's number is defined as:

$$Re_x = \frac{\rho_x D_h}{\mu_x} = \frac{\dot{m}_x D_h}{A_{cannel}\mu_x} = \frac{G_x D_h}{\mu_x} \tag{93}$$

The equivalent diameter, which is often used in the heat transfer calculations, is defined as:

$$D_e = \frac{4A_c}{P_{heated}} \tag{94}$$

Where:

P_{heated}= the heated perimeter.

7.3.2 PHE Dimensions and Pressure Loss

A problematic with determination of the PHE dimension characters and pressure characters are boned to each other. This leads that a parametric chance at the dimension variable affects also in the pressure lost variables and contrarily. It is notable that there is limitations, which are determined by the manufacturer, for the maximum pressure loss of the PHE.

The *FIGURE11* in the next page shows the PHE's principled form at the two-dimensional shape and characteristics, which are needed in calculations related to the PHE. Calculations related to the PHE's dimensions can be found as reference (7).

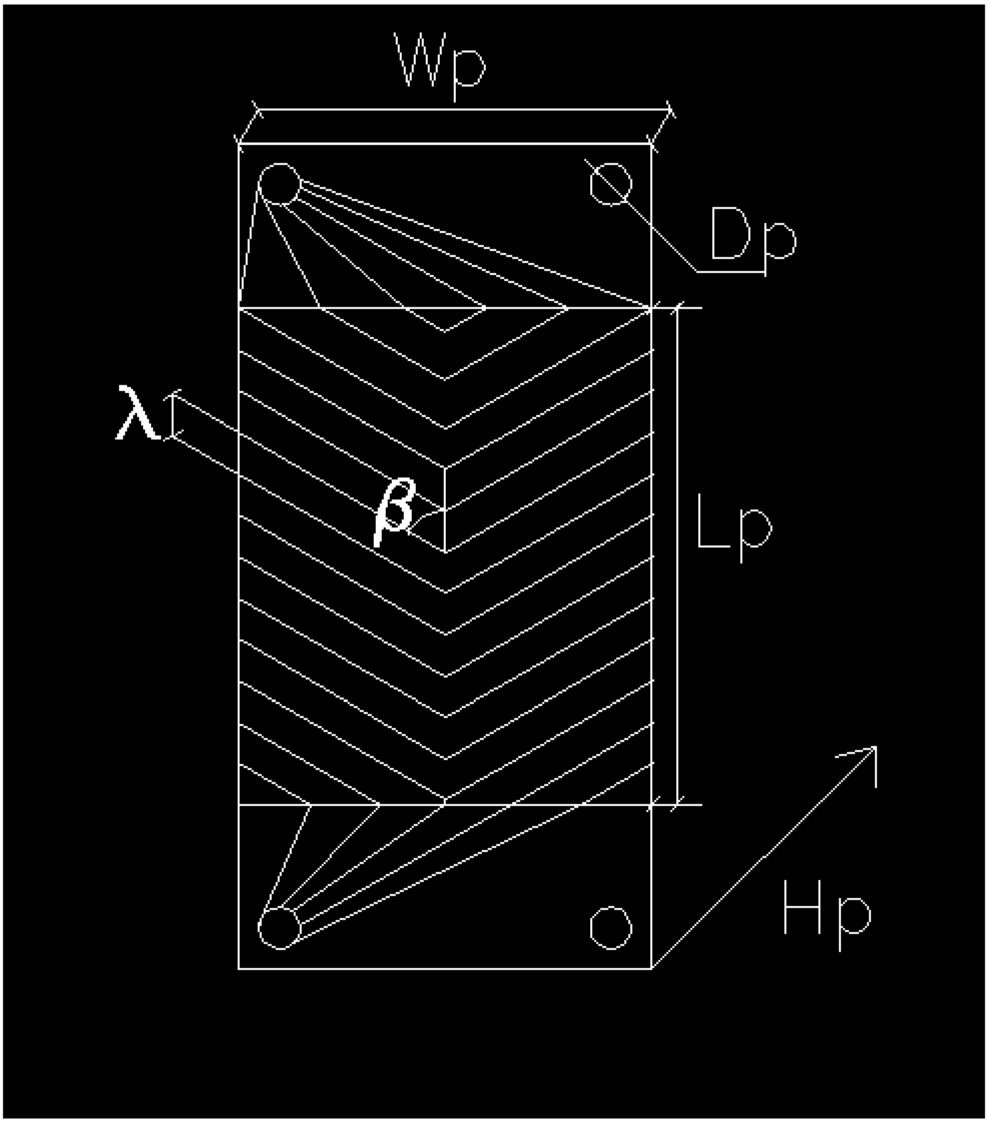


Figure 11. PHE's dimensional characteristics.

Number of plates including dimension W_p:

$$N_\lambda = \frac{W_p}{\lambda} \tag{95}$$

Where:

λ = the wavelength

Number of channels per fluid:

$$N_c = \frac{N_{tot}+1}{2N_p} \tag{96}$$

Where:

N_{tot} is the total number of plates

N_p is the number of channels per pass

The amplitude:

$$a = \frac{1}{2}\left(\frac{H_p}{N_{tot}+1} - \delta\right) \tag{97}$$

Where:

H_p is the PHE height

δ is the plate thickness

The corrugation aspect ratio:

$$\gamma = \frac{4a}{\lambda} \tag{98}$$

When ɣ=0 plate its flat-parallel. Increasing ɣmakes the surface area larger, but too high ɣ may cause vortexes at cannel heads. That will induce the heat transfer reducing.
Optimal ɣ to the PHE its 0.2-0.6.

The wavelength:

$$L_\lambda = \int_0^\lambda \sqrt{1 + (\frac{2\pi a}{\lambda})^2 \cos(\frac{2\pi x}{\lambda})^2\, dx} \tag{99}$$

Where:

a = amplitude

The heat transfer area per fluid:

$$A_t = 2L_\lambda N_\lambda L_p N_c \tag{100}$$

The free-flow area:

$$A_c = 2aW_p N_c \tag{101}$$

The surface enlargement factor:

$$\Phi = \frac{2L_\lambda N_\lambda L_p N_c}{2W_p L_p N_c} = \frac{L_\lambda N_\lambda}{W_p} \tag{102}$$

The hydraulic diameter:

$$D_h = \frac{4A_c L_p}{P_{wet} L_p} = \frac{4A_c L_p}{A_t} = \frac{4(2aW_p N_c)L_p}{2L_\lambda N_\lambda N_p N_c} = \frac{4a}{\Phi} \tag{103}$$

Where:

P_{wet}=wet circle

The fanning friction factor:

$$f = \left[\frac{\cos\beta}{(0.045\tan\beta + 0.09\sin\beta + f_0/\cos\beta)^{0.5}} + \frac{1-\cos\beta}{\sqrt{3.8f_1}}\right]\wedge -0.5 \tag{104}$$

Where:

$$f_0 = \frac{16}{Re}\ ; if\ Re < 2000\ and\ (1.56\ln Re - 3.0)\wedge -2\ if\ Re > 2000 \tag{105}$$

$$f_1 = \frac{149.25}{Re} + 0.9625\ ; if\ Re < 2000\ and\ \frac{9.75}{Re^{0.289}}\ if\ Re > 2000 \tag{106}$$

Nusselts's number

$$Nu = \frac{hD_h}{k_f} = 0.205Pr^{\frac{1}{3}}(fRe^2 \sin 2\beta)^{\wedge}0.374 * \left(\frac{\mu}{\mu_s}\right)^{\wedge}(1/6) \quad (107)$$

Where:

10°<β<80°

k_f= thermal conductivity of the fluid.

μ_s= dynamic viscosity at the wall temperature.

Assumption:

μ/μ_s=1 if μ changes moderately with the temperature.

Prandtl's number:

$$Pr = \frac{C_p\mu}{k_d} \quad (108)$$

8. Replacing Hydro-Ejector System with PHE- system.

In an idea of replacing the hydro-ejector system with the PHE- system include multiply benefits. A temperature control tolerance is changed dramatically in a more accurate direction, which gives opportunity to use a more energy effective construction style and materials and it give opportunity to use a more effective air conditioning. It is also give more freedom to the customer to decide of his/her individual temperature needs, which are hardly accomplished in the centralized hydro-ejector system.

The pipeline service can be done in much smaller area with the PHE- system, it leads situation where the service is cheaper and it needs less planning. There are also researches of the temperatures influence to human's anabolic state. For example in the winter, the heat load, which is produced with the hydro-ejector system, can raise up over 30 decrees in an accommodation or a working place. Exceptions of the temperature of the work zone are proven to influence at the human's anabolic state such that the working power be erroneously lower than in ideal conditions. An ideal working temperature is 18 degrees and the most comfortable zone is in 21 degrees (2, p. 284 & 4, c. 7.) Higher or lower temperatures gives disadvantage to a coping at work and the temperature conformability is lower than the ideal. With the PHE- heat exchanger system that problem is solved and temperature tolerance can easily be kept +/- 1 degrees or better.

The biggest disadvantage in the PHE- system may be, that thereto related motorized valves needs the electricity, which can be little problematic in areas where are common electrical shutdowns. However, it can be solved for example with emergency power generators or batteries.

In summary, there is a big energy saving potential in the Russian Federation district heating systems and also system rehabilitation can make conditions of the customers more comfortable.

In the *FIGURE 12* is described the district heating substation in the Finland. In this case, the substation includes two heat exchangers. The first one is for the domestic heating water and the second one is for heating equipments.

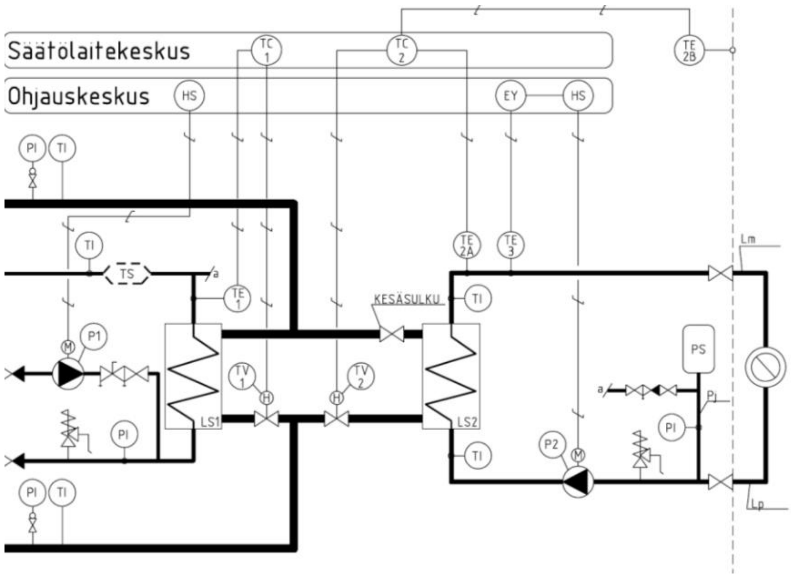


Figure 12. District Heating Substation. (18)

8.1 Nesesarry Preminilary Information

Before replacing the system, there are certain things that should be considered. It has to be done in a service period (summer) because in centralized system fluid flow cannot be stopped or chanced without interference with other parts of the system. The material choices are good to think with the consideration of the conditions, quality needs, life cycle of the product and knowledge of the national standards. Calculated values, which are needed, are the pressure drop over the ejector, the inlet pressure drop, the maximum temperature, the inlet measurement temperatures, the mass flow or the velocity and the needed heating power. Also is needed to choose suitable motored and mixing valves, pumps and an expansion tank.

It is notable that the system mass flow has to stay unchanged, lest flow characteristics in nearby branches won't be disturbed. Ergo, in the heating branch must be built a pumping circle before PHE, which keeps the main flow characteristics stable. To avoid complicated arrangements, it is recommended to replace the hydro-ejector systems with the PHE -systems in area of the whole branch in time of the service period.

8.2 Example Calculation of PHE

The subject is a ten flat residential building which is attached in the centralized district heating facility with the hydro-ejector system. The calculated input mass flow $(\dot{m}_1)$ in that building is 1.3 kg/s .The inlet temperature (T_1) , which is gained from heating facility, is 150 °C. Moreover the inlet temperature of the residential building (T_3) is determined as 95°C and temperature of the suction fluid is 70°C. The pressure loss on the secondary side is 12 kPa and the total pressure drop over the ejector is 200 kPa.

Firstly have to figure out how to keep a pressure lost and a flow stay constant with the assumption that the replacement of the system is limited on an individual part of the branch. The problem is solved when making a flow circle before the PHE and make sure that it got the same pressure lost than a replaceable system. It can be done for example with an adjustment valve, constant flow pump, tree-way valve and maintenance valves. The principle of connection it is explained the *FIGURE 13* below.

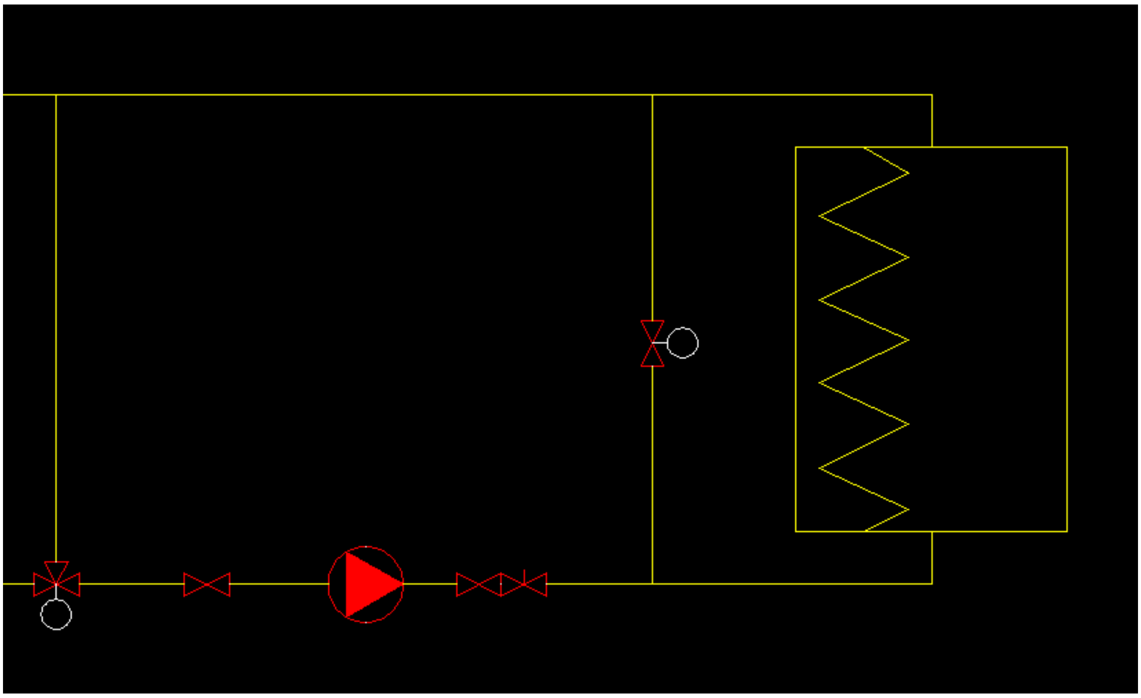


Figure 13. Principle of PHE connection

In the *FIGURE 14* is example of the substation, implemented by the manufacturer Danfoss. The substation includes the PHE and its relative components. For example, the substation includes automated valves, shut down valves, controlling center and measuring devices.

Figure 14. Substation. (19)

8.2.1Choose of Mixing Valve

A pressure lost over the valve and other components must be the same as in a replaceable system, because flow and pressure characteristics must say unchanged in the main line. The total pressure loss over the replaceable part is 200 kPa in this example, which can be assumed with the knowledge attained from the earlier examples of this thesis at Chapter 6 and 7. The pipe size is decided to be DN50, which gives pressure lost per meter, with 1.3 kg/s mass flow, about 180 Pa / m. DN50 is a good choice, because the pressure drop is ideal with this pipe size and the mass flow, if it compared with a bigger or a smaller pipe size. The replaceable hydro-ejector size number 3. got also the same size if assumed the size of the hydro-ejector be the same as in earlier examples.

When choosing a valve type must be consider the maximum pressure drop over the valve, the diameter of the valve and which kind of fluid it is purposed. Adjustability of operation must be ensured with authority of the valve, which must be bigger than 0.5 in 2-way valves and 0.3 in 3-way valves.

In this example is used 3-way valve and 2-way valves as the pump adjust valve and the PHE flow adjust valve**.**

In the *FIGURE15* is a dimensioning diagram for dimensions of the valves.

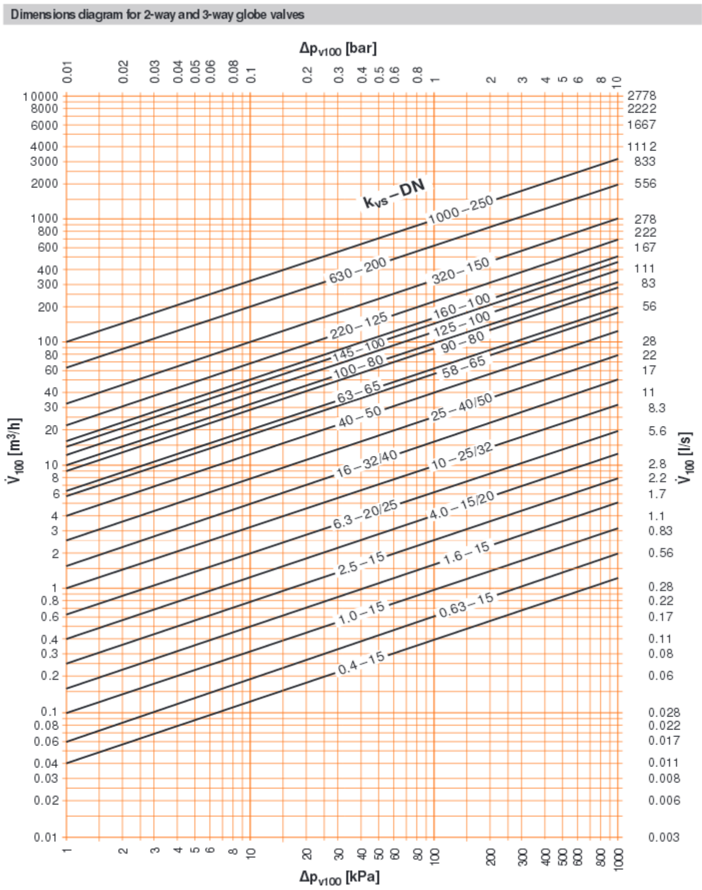


Figure 15. Dimension diagram for 2-way and 3-way globe valves. (20)

Kvs-value of the adjustment valve is calculated as:

$$kvs = \frac{q_v[\frac{m^3}{h}]}{\sqrt{\frac{\Delta p[kPa]}{100}}} \tag{109}$$

Which yields:

$$\Delta p = \left(\frac{q_v[\frac{m^3}{h}]}{kvs}\right)^2 * 100 \tag{110}$$

Suitable valves can be chosen from the dimension diagram (*FIGURE* 15).
Selecting:

3-way, DN 15 valve with kvs-value 2.5.
2-way, DN 15, withkvs-value 6.3.
The principled location of the valves can be found in the picture number 13.

Pressure losses over valves:

$$\Delta p_{3-way,allowed} = \left(\frac{q_v[\frac{m^3}{h}]}{kvs_{3-way}}\right)^2 * 100 = \left(\frac{4.68[\frac{m^3}{h}]}{2,5}\right)^2 * 100 \sim 350\ kPa \tag{111}$$

$$\Delta p_{2-way,allowed} = \left(\frac{q_v[\frac{m^3}{h}]}{kvs_{3-way}}\right)^2 * 100 = \left(\frac{4.68[\frac{m^3}{h}]}{6,3}\right)^2 * 100 \sim 55\ kPa \tag{112}$$

Authority of the adjustment valve is calculated as:

$$\beta = \frac{\Delta p_{valve}}{\Delta p_{allowed}} = \frac{\Delta p_{wanted} - \Delta p_{pipes} - \Delta p_{heat\ exchanger} - \Delta p_{valves}}{\Delta p_{allowed}} \tag{113}$$

$$\beta_{3-way} = \frac{\Delta p_{valve}}{\Delta p_{allowed}} = \frac{(200-20-35-5)kPa}{350kPa} = 0.4 \tag{114}$$

$$\beta_{2-way} = \frac{\Delta p_{valve}}{\Delta p_{allowed}} = \frac{35kPa}{55kPa} = 0.64 \tag{115}$$

It is notable, that if the authority value is too low, valves' adjustment doesn’t necessary work with full capacity. Values 0.4 and 0.64 are both over the minimum authority limit, ergo both valves are acceptable.

8.2.3 Pump selection

Before the selection of the pump it must be explored, what kind of pressure and flow characteristics are needed.

Pressure loss over the pump can be calculated with knowledge of the total pressure loss over the flow circle, in addition the pressure loss of the pump adjustment valve, as well a pressure loss of the PHE's primary side flow.

Pressure loss over the pump:

$$\Delta p_{pump} = \Delta p_{exchanger} + \Delta p_{pipes} + \Delta p_{valve} = (35 + 20 + 5) = 60kPa \quad (116)$$

Volume velocity:

The volume velocity is gained from calculations (111 & 112), because the volume velocity over the valve has same value than the volume velocity over the pump. The value is 4.68 m^3/h.

With the knowledge of the needed pressure loss and the needed volume velocity, pump can be selected from diagrams or it can be selected with a pump selection program.

8.2.4 Variables of Sizing of PHE

For the hot fluid heat transfer rate is:

$$q = \dot{m}_1 Cp_1 (T_{1in} - T_{1out}) = \frac{1.3kg}{s} * \frac{4.2kJ}{kg} * (150^{\circ}C - 70^{\circ}C) = 436.8kW \quad (117)$$

It is notable that if more accurate calculations are wanted, the specific heat must be determined with knowledge of the mean temperature of the process.

Where:

$\dot{m}_1$= the mass flow rate for the primary side fluid and:
Cp_1= the specific heat for the primary side fluid.
Because the laws of thermodynamics the heat transfer rate for the cold fluid are the same than above. Chancing variables in the secondary side are mass flow characteristics and fluid temperatures.

The heating power:

$$q = UAF\Delta T_{lm} \tag{118}$$

Where:

F = the correction factor, depending on the flow arrangements, for example the fluid type and the flow type, which is either a parallel or a counter flow. The correction factor can be assumed as 1 if unknown. The factory F influencers are heat capacity rations $R = \frac{\dot{C}_{hot}}{\dot{C}_{cold}}\left[\frac{W}{K}\right]$and temperature effectiveness. (119)

Which leads:

$$A = \frac{q}{UAF\Delta T_{lm}} \tag{120}$$

This is a minimum transfer area which is needed to convey the heating power on the primary side to the secondary side. Later, side characters can be guessed or iterated, depending on the wanted heat transfer area, pressure lost, size and the shape. There are also mathematics programs and programs designed by the heat exchangers manufactures, which use is recommended to save lots of time and energy if the plan is not to make a totally new model of the heat exchanger or the heat exchanger calculation program.

When calculating the U- value, there are certain values, which have to guess or decide with the experimental knowledge. The values are the wall thickness, thermal resistance of the wall, inner and outer connective heat transfer coefficients and a fouling.

In normal cases it is good to determine and optimize the size of the minimum transfer area with accurate value of U. Because it gives the limit value, which helps to find the minimum size of the PHE. In this work the main variables are the size of the PHE and number of the plates of the PHE. This decision is made, because is desired to show that kind of view in the PHE calculation process. It means that it is not intended to find the minimum or optimal size of the PHE, instead of that it is intended to show the influence of the parameters change correspond in to the pressure loss.

The logarithmic temperature is:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{80-30}{ln\left(\frac{80}{30}\right)} = 51.0°C \tag{121}$$

The U-value is:

$$U \ = \frac{1}{R_t} \sim 706\ W/m^2K \tag{122}$$

Where:

$$R_t = \frac{1}{\alpha_1} + \frac{s}{\lambda} + \frac{1}{\alpha_2} + m_l = \frac{1}{1550W/m^2K} + \frac{0.0006m}{236W/mK} + \frac{1}{1300W/m^2K} + 0 \tag{123}$$

R_t= the total heat transfer resistance
α_1= the inner heat transfer coefficient (assumed to be 1550W/m^{2}K)
s= thickness of the wall (assumed to be 0.6 mm)
α_2= the outer heat transfer coefficient (assumed to be 1300W/m^{2}K)
m_l= the fouling resistance
λ = the thermal conductivity of the material (236 W/mK, aluminum type 304)

The fouling resistance can be assumed as 0 if the system is new. Even in situations, where pressure loss is relatively low, the maximum level of the value of the fouling resistance is up to 0.00012 m^{2}K/W.

The inner and outer heat transfer coefficients can be iterated with knowledge of the Nusselts number (Nu_x), thermal resistance of the fluid (λ_x) and the diameter of the channel (d_x). Values are tied to the subscripts.

$$\alpha_1 = \frac{Nu_1 \lambda}{d_1} \tag{124}$$

$$\alpha_2 = \frac{Nu_2 \lambda}{d_2} \tag{125}$$

To avoid iteration, the inner and outer heat transfer coefficients are assumed to be:

$\alpha_{1,asumption}$= 1550 W/m^{2}K

$\alpha_{2,asumption}$= 1300 W/m^{2}K

Adding all missing values which are required and value A can be solved.

$$A = \frac{q}{UF\Delta T_{lm}} = \frac{436.8\ kW}{\frac{706W}{m^2K}*1*51.0^{\circ}C} \sim 0.012m^2 \tag{126}$$

The heat transfer area per fluid:

$$A_t = 2L_\lambda N_\lambda L_p N_c \tag{127}$$

Where the number of waves:

$$N_\lambda = \frac{W_p}{\lambda} = \frac{1.0m}{0.1m} = 10 \tag{128}$$

This calculation shows that the number of waves depending on the width of the heat exchanger and the length of the wave. The heat exchanger width optimized with knowledge of the space requirements. In this exercise, the value of waves is randomly chosen and not necessary fill re-quirements which is mentioned above in every conditions.

Where:

The length of wave:

$$L_\lambda = \int_0^\lambda \sqrt{1 + \left(\frac{2\pi a}{\lambda}\right)^2 \cos\left(\frac{2\pi x}{\lambda}\right)^2 dx}$$

$$L_\lambda = \int_0^{0,1} \sqrt{1 + \left(\frac{2\pi 0{,}042m}{0{,}1m}\right)^2 \cos\left(\frac{2\pi x}{0{,}1m}\right)^2 dx} \sim 0.27m \quad (129)$$

Where:

a = the amplitude

The amplitude:

$$a = \frac{1}{2}\left(\frac{H_p}{N_{tot}+1} - \delta\right) = \frac{1}{2}\left(\frac{1.0m}{10+1} - 0.006m\right) \sim 0.042m \quad (130)$$

Where:

H_p= the PHE height
δ= the plate thickness
N_{tot}= total number of the plates

The number of channels per fluid:

$$N_c = \frac{N_{tot}+1}{2N_p} = \frac{10+1}{2*1} = 5.5 \quad (131)$$

The area of channel:

$$A_c = 2aW_pN_c = 2 * 0.042m * 1.0m * 5.5 = 0.462m^2 \quad (132)$$

The surface enlargement factor:

$$\Phi = \frac{2L_\lambda N_\lambda L_p N_c}{2W_p L_p N_c} = \frac{L_\lambda N_\lambda}{W_p} = \frac{0.1m*10}{1.0m} = 1.0 \tag{133}$$

The hydraulic diameter:

$$D_h = \frac{4A_c L_p}{P_{wet} L_p} = \frac{4A_c L_p}{A_t} = \frac{4(2aW_p N_c)L_p}{2L_\lambda N_\lambda N_p N_c} = \frac{4a}{\Phi} = \frac{4*0.042m}{1.00} = 0.168m \tag{134}$$

The fanning friction factor:

$$f = \left[\frac{\cos\beta}{(0.045\tan\beta + 0.09\sin\beta + f_0/\cos\beta)^{0.5}} + \frac{1-\cos\beta}{\sqrt{3.8f_1}}\right]^{-0.5}$$

$$f = \left[\frac{\cos 60^\circ}{(0.045\tan 60^\circ + 0.09\sin 60^\circ + 2.52*10^{-3}/\cos 60^\circ)^{0.5}} + \frac{1-\cos 60^\circ}{\sqrt{3.8*0.14}}\right]^{-0.5} = 0.71 \tag{135}$$

Where:

$$f_0 = (1.56\ln Re - 3.0)^\wedge - 2 = (1.56\ln 261306 - 3.0)^\wedge - 2 = 2.52*10^{-3} \tag{136}$$

$$f_1 = \frac{9.75}{Re^{0.289}} = \frac{9.75}{261306^{0.289}} = 0.14 \tag{137}$$

Reynolds's number is defined as:

$$Re_x = \frac{\rho_x D_h}{\mu_x} = \frac{\dot{m}_x D_h}{A_{cannel}\mu_x} = \frac{G_x D_h}{\mu_x} = \frac{\frac{1.3kg}{s}*0.168m}{0.462m^2*1.99*10^{-7}} = 2375514 \tag{138}$$

Nusselts' number:

$$Nu = \frac{hD_h}{k_f} = 0.025Pr^{\frac{1}{3}}(fRe^2 sin2\beta)^{0.374}(\frac{\mu}{\mu_s})^{1/6} \tag{139}$$

Where:

10˚<β<80˚,

k_f= the thermal conductivity of fluid

μ_s= dynamic viscosity of fluid at the wall temperature.

μ/μ_smay been used as 1 if assumed that μ moderate changes whit temperature.

The pressure loss of the secondary side:

$$\Delta p_f = \frac{2fLG^2}{Dh*\rho} N_p \frac{2*0.71*1*\left(\frac{4*1.3kg/s}{\pi*0.169^2}\right)^2}{0.169m^2*1000kg/m^3} * 1 = 27\ kPa \qquad (140)$$

The *DIAGRAM* 8 below shows, that hydraulic diameter depends on of the number of the plates if size stays constant.

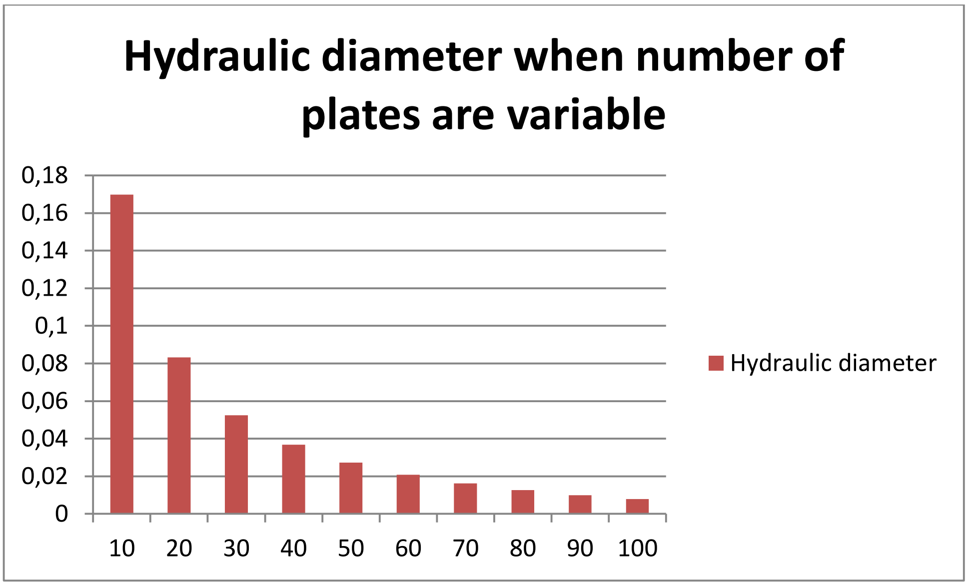


Diagram 8. Hydraulic Diameter when Number of Plates are Variable.

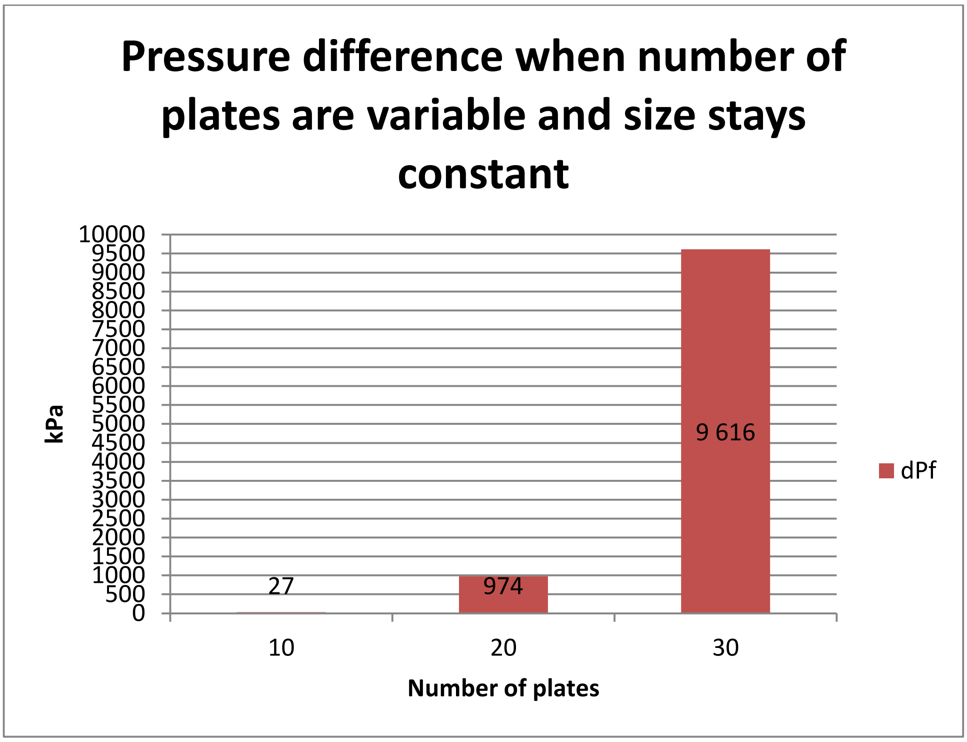


Diagram 9. Pressure Difference, when Number of Plates are Variable and Size stays Constant.

The *DIGRAMS 8 and 9* shows that, there is limited range from the hydraulic diameter with specific mass flow, when they are wanted to stay inside of a reasonable pressure difference if size of the plates stays constant. Another option is to use size of the heat exchanger as a variable. It is notable, that the pressure difference, which is calculated in (*DIAGRAM* 9) is only considering the pressure loss of the passes.There is also pressure loss calculated from the ports of the heat exchanger, which are usually lover than 10 present of the total pressure loss of the heat exchanger.However it can be almost 25 or 30 percent in some design variations (12, p. 291.)

The pressure loss of the port:

$$\Delta p_p = \frac{1{,}5NpG^2p}{2*\rho} \quad (141)$$

If assumed that the total pressure loss of the heat exchanger is approximately 32.4 kPa and the pressure loss of the share of the passes is 27 kPa, means that the pressure loss which are left over to ports are 5.4 kPa. With that information, the size of the ports can be determined with iteration or some mathematic calculation program.

This leads:

$$\Delta p_p = \frac{1.5NpG^2p}{2*\rho} \rightarrow 5.4kpa = \frac{1.5*10*\left(\frac{4*\dot{m}}{\pi*Dh(X)^2}\right)^2}{2*\rho} \rightarrow Dh\sim 0.16m \qquad (142)$$

There are multiple different ways to calculate and measure the "Chevon" plate heat exchanger and this is just one method. Those calculations above are to show, that there are many variables, which are good to considered when designing a totally new model of the heat exchanger or just make pre calculations with purpose of choosing the heat exchanger from models what are already exist. Relevant is to give needed information, which can be used in progress where the hydro-ejector system is replaced with the heat exchanger system.

9. Final Results and Conclusions

As a result of this work it can be seen that it is possible to make a rehabilitation in the Russian Federation district heating system in a big area or just an individual building. It gives clearance from the possibility to make the rehabilitation in a singular building without disturbing fluid flow at the mainline. Work gives a pre knowledge of the Russian tariff systems and development plans of the energy law, which help investors make decisions in order to invest in the Russian heating sector.

This work also gives proof, that there are no theoretical obstacles to combine the Russian Heating norms and the Northern European technology, and result in more energy effective and more customer satisfaction enhancing products. It can also be said that the ejector system is workable and a great heating system. Its simplicity and service assurance can be mentioned as good features.

The physical theory of the ejector is relevant in many other situations of engineering, because the ejector system can be confronted with different variations in multiple systems, including space rockets, fountains, laser technology and much more.

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