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CALCULATION AND DESIGN OF A PIPE-IN-PIPE HEAT EXCHANGER FOR LIQUID COOLING

Xadjibayev Akbarjon Shavkatovich,

Sharipov Kozimjon Komiljonovich,

Shokirov Asliddin Pazlitdin ugli,

Erkinov Ahliyor Komiljon ugli,

Jobborova Maftuna Akrom kizi,

Nosirov Murod Ilkhomovich

Abstract: This paper presents the calculation and design of a pipe-in-pipe heat exchanger intended for cooling liquid media. Thermal and hydraulic calculations were performed, the main design parameters of the apparatus were determined, and appropriate equipment was selected. The obtained results can be used in the design of heat exchange systems in the chemical and petroleum refining industries.

Key words: heat exchanger, pipe-in-pipe, heat transfer, hydraulic calculation, heat exchange, chemical engineering.

Introduction

Heat exchangers are widely used in the chemical, petrochemical, and other industries for transferring thermal energy between heat carriers. Depending on their purpose, they are used as heaters or coolers.

Among various types of heat exchangers, **surface heat exchangers** occupy a special place; in these devices, heat is transferred through a separating wall. One of their representatives is the **pipe-in-pipe heat exchanger**, which is characterized by a simple design and high heat transfer efficiency.

Advantages of pipe-in-pipe heat exchangers:

- high heat transfer coefficient due to high velocities of both heat carriers;
- simplicity of manufacturing.

Disadvantages:

- bulky construction;
- high cost due to significant metal consumption for outer pipes not directly involved in heat transfer;
- difficulty in cleaning the annular (interpipe) space.

Pipe-in-pipe heat exchangers can be used for both heating and cooling processes.
Heating is usually carried out using hot water or saturated steam supplied to the annular

space, where it condenses on the outer surface of the inner pipe.

The use of water steam as a heating agent has the following advantages:

- high heat transfer coefficient;
- large amount of heat released during steam condensation;
- uniform heating, since condensation occurs at a constant temperature;
- easy control of the heating process.

When used for cooling, river or artesian water can serve as a coolant. In cases where temperatures below 5–20 °C are required, refrigeration brines (aqueous solutions of $CaCl_2$, NaCl, etc.) are used.

1. Process Flow Diagram

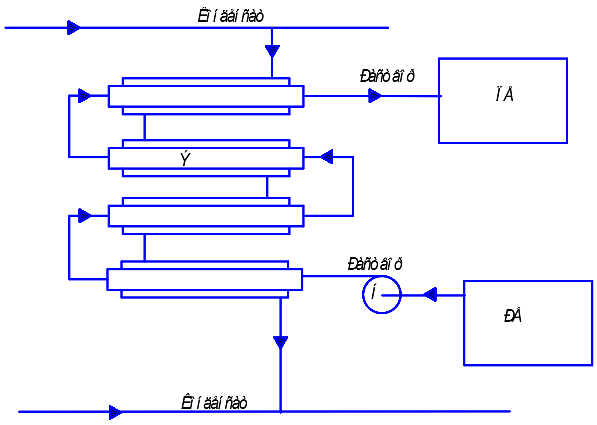

Figure 1. Pipe-in-pipe heat exchanger. Process flow diagram.

Kerosene from the feed tank (FT) is supplied by a centrifugal pump (P) into the tube side of the element (E) of the pipe-in-pipe heat exchanger. Heating steam is introduced into the annular space of the heat exchanger, where it condenses on the outer surface of the inner tubes and flows downward as a condensate film, which is then discharged into the condensate line.

The fluid, heated due to the latent heat of steam condensation, flows by gravity into the receiving tank.

2. Selection of Construction Material

Since the aqueous NaOH solution is a corrosive medium, austenitic stainless steel X18H10T (GOST 5632–72) is selected as the construction material for the main components. This material is resistant to highly aggressive environments and can operate at temperatures up to 600 °C.

3. Operating Temperature Regime of the Apparatus

Принимаем противоточную схему движения теплоносителей

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Пар поступает в межтрубное пространство, а раствор двигается по внутренней трубе

Figure 2. Flow diagram of heat carriers.

3.1. Mean Temperature Difference

$\Delta t_м = t_{1н} – t_{2к} = 105 – 75 = 30$ °C

$\Delta t_б = t_{1к} – t_{2н} = 55 – 15 = 40$ °C

Since the ratio $\Delta t_б/\Delta t_м = 40/30 = 1{,}3 < 2$, то

$\Delta t_{cp} = (\Delta t_б + \Delta t_м)/2 = (40 + 30)/2 = 35$ °C

Average Temperature of the Condensate:

$t_{1cp} = (t_{1н} + t_{1к})/2 = (105+55)/2 = 80$ °C

Average Temperature of the Solution:

$t_{2cp} = t_{1cp} – \Delta t_{cp} = 80 – 35 = 45$ °C.

3.2. Heat Duty of the Apparatus:

$Q = 1{,}05 G_2 c_2 (t_{2н} - t_{2к})$,

where: c_2= 3,61 kJ/(kg·K)— specific heat capacity of the solution [1, p. 248];

G_2— mass flow rate of the solution;

1.05 — correction factor accounting for heat losses to the surroundings.

$G_2 = 7000/3600 = 1{,}94$ kg/s,

$Q = 1{,}05 \cdot 1{,}94 \cdot 3{,}61(75 - 15) = 441{,}2$ kw.

3.3. Condensate Flow Rate:

$G_1 = Q/c_1(t_{1н} - t_{1к})$,

where: $c_1 = 4{,}19$ kJ/(kg·K)— Specific Heat Capacity of Water at 80 °C [1c.537].

$G_1 = 441{,}2/4{,}19(105\text{-}55) = 2{,}11$ kg/s.

3.4. Selection of Main Design Dimensions of the Apparatus

It is assumed that the apparatus is constructed using pipes of **48 × 4 mm** (inner tube) and **76 × 4 mm** (outer tube) [2, p. 61].

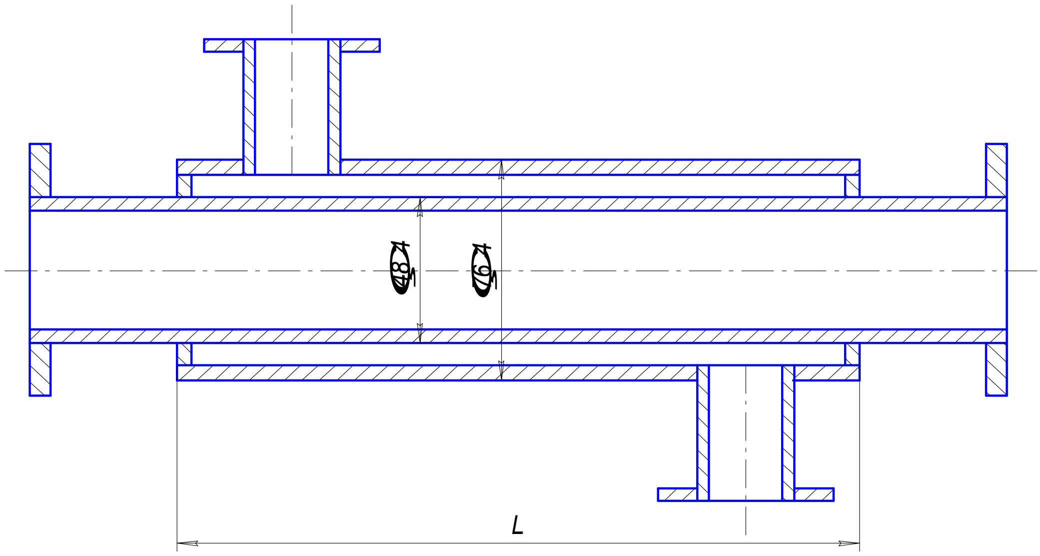

Figure 3. Heat exchange element.

Optimal heat transfer conditions are achieved under a **turbulent flow regime** (Re> 10000). Therefore, the velocity of the solution in the tubes must be greater than w'_2.

$w'_2 = Re_2 \mu_2 / (d_{вн} \rho_2) = 10000 \cdot 2{,}29 \cdot 10^{-3}/(0{,}040 \cdot 1205) = 0{,}48$ м/с

where: $\mu_2 = 2{,}29 \cdot 10^{-3}$ Pa·s — dynamic viscosity of the solution [1 c.516],

$\rho_2 = 1205$ кг/м^3 – Density of the solution [1c.512]

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d_2 = 0,040 – Inner diameter of the tube.

Number of parallel operating pipes (48 × 4):

$n` = G_2/0{,}785 d_{вн}^2 w`_2 \rho_2 = 1{,}94/0{,}785 \cdot 0{,}040^2 \cdot 0{,}48 \cdot 1205 = 2{,}67$

To ensure a stable turbulent flow regime, the number of parallel pipes is taken as n`=2. In this case, the actual velocity of the solution is given by:

$w_2 = G_2/0{,}785 d_{вн}^2 n`_2 \rho_2 = 1{,}94/0{,}785 \cdot 0{,}040^2 \cdot 2 \cdot 1205 = 0{,}64$ m/c.

Reynolds Number for the Solution:

$Re_2 = w_2 d_2 \rho_2/\mu_2 = 0{,}64 \cdot 0{,}040 \cdot 1205/2{,}29 \cdot 10^{-3} = 13490$,

Flow regime — turbulent

3.5. Heat Transfer Coefficient from the Wall to the Solution

Nusselt Number:

$Nu_2 = 0{,}023 \cdot Re_2^{0,8} \cdot Pr_2^{0,4} \cdot (Pr_2/Pr_{2ст})^{0,25}$

Prandtl Number

$Pr_2 = c\mu/\lambda = 3{,}61 \cdot 2{,}29/0{,}677 = 12{,}2$

λ = 0,677 w/m·K – Thermal conductivity coefficient [3c.55]

In the first approximation, we assume $(Pr_2/Pr_{2ст})^{0,25} = 1$, then

$Nu_2 = 0{,}023 \cdot 13490^{0,8} \cdot 12{,}2^{0,4} = 126{,}0$

$\alpha_2 = Nu_2 \lambda_2/d_{вн} = 126{,}0 \cdot 0{,}677/0{,}040 = 2132$ w/(m²·K)

3.7. Heat Transfer Coefficient from the Condensate to the Wall

Velocity of water in the annular (interpipe) space

$w_1 = G_1/[\rho_1 0{,}785(D_{вн}^2 - d_н^2)n] =$

$= 2{,}11/972 \cdot 0{,}785 \cdot (0{,}068^2 - 0{,}048^2) \cdot 2 = 0{,}60$ m/c,

where: ρ_1 = 972 кг/м³ – Density of water at 80 °C [1c. 537],

$D_{вн}$ = 0,068 м – Inner diameter of the outer (larger) tube,

$d_н$ = 0,048 м – Outer diameter of the inner (smaller) tube.

Reynolds Number for Water:

$Re_1 = w_1 d_э \rho_1/\mu_1$,

where: $\mu_1 = 0{,}355 \cdot 10^{-3}$ – dynamic viscosity of water 80 °C [1c. 537],

$d_э$ – quivalent diameter of the annular space.

$d_э = D_{вн} – d_н = 0{,}068 – 0{,}048 = 0{,}020$ м

$Re_1 = 0{,}60 \cdot 0{,}020 \cdot 972/0{,}355 \cdot 10^{-3} = 32636$

Flow regime — turbulent.

Nusselt Number:

$Nu_1 = 0{,}023 \cdot Re_1^{0,8} \cdot Pr_1^{0,4} \cdot (Pr_1/Pr_{1ст})^{0,25}$

Prandtl Number for Water $Pr_1 = 2{,}21$ [1c. 537]

In the first approximation, we assume $(Pr_1/Pr_{1ст})^{0,25} = 1$, then

$Nu_1 = 0{,}023 \cdot 32636^{0,8} \cdot 2{,}21^{0,4} = 129{,}0$

$\alpha_1 = Nu_1 \lambda_1/d_э = 129{,}0 \cdot 0{,}675/0{,}020 = 4352$ Вт/(м²·К)

where: $\lambda_1 = 0{,}675$ Вт/(м·К) – Thermal conductivity coefficient 80 °C [1c. 537]

3.8. Thermal Resistance of the Wall

$$\sum \frac{\delta}{\lambda} = \frac{\delta_{CT}}{\lambda_{CT}} + r_1 + r_2$$

where $\delta = 0{,}004$ м – Wall thickness

$\lambda_{ст} = 17{,}5$ wt/(м·К) – Thermal conductivity of stainless steel [1c. 529]

$r_1 = r_2 = 1/5800$ м·К/wt – Fouling thermal resistance [1c. 531]

$$\sum \frac{\delta}{\lambda} = (0{,}004/17{,}5) + (1/5800) + (1/5800) = 5{,}73 \cdot 10^{-4} \text{ м·К/wt}$$

3.9. Overall Heat Transfer Coefficient

$$K = \frac{1}{\frac{1}{\alpha_1} + \sum \frac{\delta}{\lambda} + \frac{1}{\alpha_2}}$$

$K = 1/(1/2132 + 5{,}73 \cdot 10^{-4} + 1/4352) = 786$ Вт/(м²·К)

3.10. Wall Temperatures

On the solution side

$t_{ст2} = t_2 + t_{ср}/\alpha_2 = 45{,}0 + 786 \cdot 35{,}0/2132 = 57{,}9°$ C,

$Pr_{ст2}= 9,6 \rightarrow \alpha_{1ут} =2132\cdot(12,2/9,6)^{0,25} =2263$ wt/(м²·К).

On the water side:

$t_{ст1} = t_1 - K\Delta t_{ср}/\alpha_1 = 80,0 - 786\cdot35,0/4352 = 73,7°$ C,

$\alpha_1 = 4352(2,21/2,43)^{0,25} = 4250$ Вт/(м²·К).

3.11. Refined Calculation of the Overall Heat Transfer Coefficient

$K = 1/(1/2263 + 5,73\cdot10^{-4}+1/4250) = 800$ Вт/(м²·К)

Check of the wall temperature

$t_{ст1} = t_1 - K\Delta t_{ср}/\alpha_1 = 80,0 - 800\cdot35,0/4250 = 73,4°$ C

$t_{ст2} = t_2 - K\Delta t_{ср}/\alpha_2 = 45,0+ 800\cdot35,0/2263 = 57,4°$ C

The obtained values are close to the previously assumed ones, and no further refinement is required

3.12. Heat Transfer Surface Area

$F = Q/(K\Delta t_{ср}) =441,2\cdot10^3/(800\cdot35,0) =15,75$ м²

3.13. Selection of Standard Equipment

According to **GOST 8930–78** [2, p. 61], standard non-detachable elements with a length of **6.0 m** are selected, each having a heat transfer surface area of **0.90 m²**.
Thus, the number of elements in one row is determined as:

$N = F/(nF_1) =15,75/(2\cdot0,90) = 8,75$ assume $N = 9$

4. Hydraulic Calculation

4.1. Friction Factor of the Solution in the Tubes

The velocity of the solution in the tubes is: $w_2 = 0,64$ m/s

Relative Roughness:

$e_2 = \Delta/d_{вн} = 0,0002/0,040 = 0,0050$

where $\Delta = 0,0002$ м – relative roughness [2c. 14]

Friction factor. Since the following condition is satisfied:

$10/e_2 = 10/0,005 =2000 < Re_2 < 560/e_2 = 560/0,005 = 112000$

then the friction factor is given by:

$\lambda_2 = 0,11(e_2 + 68/Re_2)^{0,25} = 0,11(0,005 + 68/13490)^{0,25} = 0,035$

4.2. Sum of Local Resistances

$\sum\xi = \xi_1 + \xi_2 + 4\xi_3 = 0{,}5 + 1{,}0 + 8\cdot 0{,}154 = 2{,}73$

where $\xi_1 = 0{,}5$ – Inlet to the pipe [2c.14]

$\xi_2 = 1{,}0$ – Outlet from the pipe

$\xi_3 = AB = 1{,}4\cdot 0{,}11 = 0{,}154$ – Elbow of circular cross-section

4.3. Hydraulic Resistance of the Tube Side

$$\Delta P_2 = \left(\lambda_2 \frac{lN}{d_{вн}} + \sum\xi\right)\frac{\rho_2 w_2^2}{2} = (0{,}035\cdot 6\cdot 9/0{,}040 + 2{,}73)1205\cdot 0{,}64^2/2 = 12334 \text{ Pa}$$

4.4. Pump Selection

Required Pump Head

$H = \Delta P/(\rho g) = 12334/(1205\cdot 9{,}8) = 1{,}04$ m

Volumetric Flow Rate

$Q = G/\rho = 1{,}94/1205 = 0{,}0016$ м^3/c

Based on these two parameters, a centrifugal pump X8/30 is selected, for which the capacity is $Q = 2{,}4\cdot 10^{-3}$ м^3/c, head $H = 17$ м [2c. 38].

4.5. Friction Factor for Water in the Annular Space

Velocity of water in the annular (interpipe) space $w_1 = 0{,}60$ m/s

Relative roughness:

$e_1 = \Delta/d_э = 0{,}0002/0{,}020 = 0{,}0100.$

Since the condition is satisfied:

$10/e_1 = 10/0{,}010 = 1000 < Re_1 < 560/e_1 = 560/0{,}0100 = 56000,$

then the friction factor is equal to:

$\lambda_1 = 0{,}11(e_1 + 68/Re_1)^{0{,}25} = 0{,}11\cdot(0{,}0100 + 68/32636)^{0{,}25} = 0{,}036$

4.6. Sum of Local Resistances

$\sum\xi = 9(\xi_1 + \xi_2) = 13{,}5$

where $\xi_1 = 0{,}5$ – Inlet to the pipe [2c.14]

$\xi_2 = 1{,}0$ – Outlet from the pipe

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4.7. Hydraulic Resistance of the Annular Space

$$\Delta P_1 = \left(\lambda_1 \frac{lN}{d_э} + \sum \xi\right)\frac{\rho_1 w_1^2}{2} = (0{,}036 \cdot 6 \cdot 9/0{,}020 + 13{,}5) \cdot 972 \cdot 0{,}60^2/2 = 19368 \text{ Pa}$$

4.5. 4.8. Pump Selection

Required Pump Head

H = ΔP/(ρg) = 19368/(972·9,8) = 2,0 м

Volumetric flow rate per second

Q = G / ρ = 2,11/972 = $2{,}17 \cdot 10^{-3}$ м^3/с

Based on these two parameters, we select the centrifugal pump X8/30, for which the capacity is Q Q = $2{,}4 \cdot 10^{-3}$ m^3/s and the head is H =17m. [2c. 38]

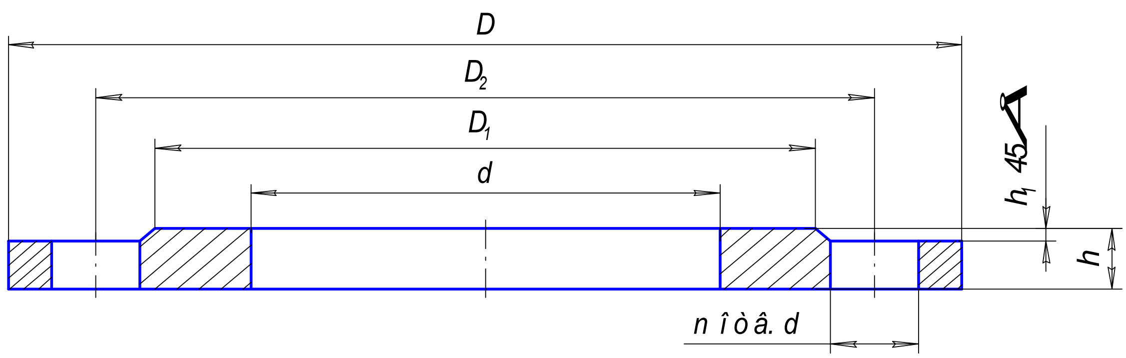
5. Structural design calculation

5.1. Connection of elements

The elements are connected to each other by means of bends with a radius of 100 mm curved at 180°

5.2. Flanges

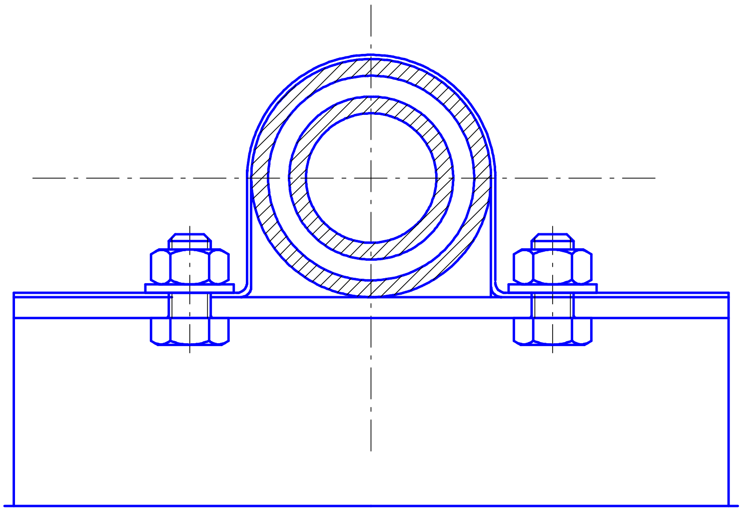
The bends and inner tubes are equipped with flat welded flanges according to GOST 12820-80, the design and dimensions of which are given below.



d	D	D₂	D₁	h	n	D
50	140	110	90	13	4	14

5.3. Support structures

The heat exchanger is mounted on a welded frame made of No. 5 angle steel. The heat-exchange elements are attached to the support with bolts using clamps.



6. Results and Discussion

As a result of the performed calculations, a “pipe-in-pipe” type heat exchanger was designed, providing efficient cooling of the liquid under the specified parameters.

The obtained values of the heat transfer and overall heat-transfer coefficients confirm the efficiency of the selected design. The turbulent flow regime of the heat carriers promotes the intensification of heat exchange.

7. Conclusion

The developed heat-exchange apparatus meets the specified technological requirements. The performed thermal and hydraulic calculations showed that the selected design ensures efficient heat transfer with relatively compact dimensions.

“Pipe-in-pipe” type heat exchangers are promising for application in the chemical industry due to their simple design and high efficiency.

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