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Heat Transfer in Pulsating Laminar Flow in a Pipe: Evaluation of the Reduction in the Heat Exchange Area of Oil Cooler

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Heat Transfer in Pulsating Laminar Flow in a Pipe: Evaluation of the Reduction in the Heat Exchange Area of Oil Cooler

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Abstract. In this study, the heat transfer in a pipe was investigated by a numerical method with the pulsating flow of oil at Reynolds numbers 280, 350, 420. The diameter *D* of the pipe was 0.014 m. The pipe length was 100*D*. The flow pulsations were not harmonical. A numerical experiment was carried out using AnsysFluent. Based on the data obtained from the numerical experiment, an estimate of the reduction in the heat exchange area of the oil cooler was calculated. The reduction in the heat exchange area of the oil cooler ranged from 2.57% to 8.63%, depending on the pulsation mode.

1. Introduction

Oil coolers are widely used in a variety of engineering applications. Shell and tube oil coolers are used to cool oil in bearings of pumping stations, which are used to transport oil products in the petrochemical industry. Due to the high viscosity of the oil, its flow in the oil cooler is laminar. The Reynolds number is below 1000. Consequently, the heat transfer coefficient from the oil side is low, which reduces the cooling efficiency of oil coolers. Usually, low performance is compensated by an increase in the heat exchange surface. To increase heat transfer, various intensification methods are used [1,2]. Passive and active methods are used to intensify heat transfer. The passive method includes pipes with internal spiral finning [3], placement of twisted tapes inside pipes [4,5], double or triple twisted tapes [6,7], helical twisted tapes [8,9]. The active method includes the rotation of the heat exchange surface [10], the use of an electric field [11], flow pulsations [12].

This paper investigates the effect of flow pulsations on heat transfer in the tube side of the oil cooler. The possibility of reducing the heat exchange area of oil coolers due to flow pulsations is also being evaluated. The use of pulsations for intensifying heat transfer in a pipe has been investigated by many authors [13-18]. However, the results obtained in this area are still not sufficient. It should be added that the flow pulsations in the above-mentioned works are close to sinusoidal. In this work, the pulsations are asymmetric.

2. Mathematical model

2.1. Computational domain and boundary conditions

The computational domain was a pipe with a diameter D = 14 mm and a length L = 1400 mm (Figure 1). The velocity at the pipe inlet u_{inlet} m/s was calculated depending on the Reynolds number Re. Re=$u_{inlet}D/\nu$, where ν– is the kinematic viscosity of the oil, m^2/s. At the pipe outlet, the ambient pressure was set to P = 101325 Pa. This study examines heat transfer during oil cooling inside the tube side of a heat exchanger, so the pipe wall temperature was one degree less inlet temperature T_{wall}=49 °C, T_{inlet}=50 °C. The properties of the oil were constant. The density ρ = 859.4 kg/m^3, heat capacity c_p = 1965.4 J/kg · C, dynamic viscosity μ = 0.019 Pa · s, thermal conductivity k = 0.13 W/m · °C. Pulsating velocity was implementing in the inlet of the pipe. The pulsating velocity had the required frequency f =1/T Hz, where T– is the period of the pulsations, s. The asymmetry of the pulsations was characterized by the parameter duty of cycle $\psi = T_1/T$, where T_1– is the time period corresponding to the reverse flow in the pipe, s [19].

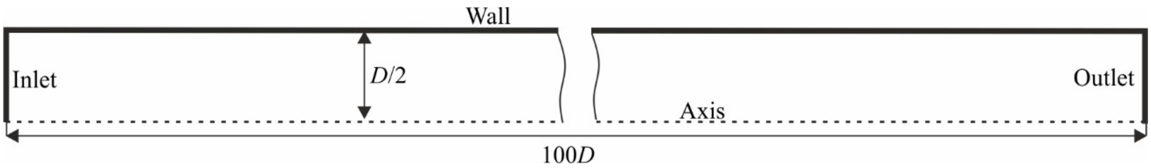

Figure 1. Computational domain.

2.2. Mesh independence test and verification

A numerical study of heat transfer with a pulsating flow in a tube was carried out in AnsysFluent [20]. The fluid flow was solved in an axisymmetric formulation. To check the convergence of the grid solver, calculations were performed with three grids with different numbers of elements. Mesh A with 50x500 elements, Mesh B with 75x750 elements and Mesh C with 100x1000 elements. The calculation was carried out for a stationary flow at Re 200, 250, and 300. To verify the results obtained, the Nusselt number Nu was compared with the empirical equation for laminar flow in a tube at Re ≤ 2300 [21].

The results obtained are consistent with the empirical equation (1) [21] (Figure 2). The maximum deviation of the Nu number for Mesh A, Mesh B, and Mesh C was no more than 18.5%, 18.9%, 19.2%, respectively. The difference in Nu numbers between Mesh A and Mesh B was below 0.4% between Mesh B and Mesh C below 0.3% (Figure 3). Mesh C was chosen for the numerical experiment.

$$\mathrm{Nu} = 1.86 \cdot (\mathrm{Re} \cdot \mathrm{Pr} \cdot D/L)^{1/3} \qquad (1)$$

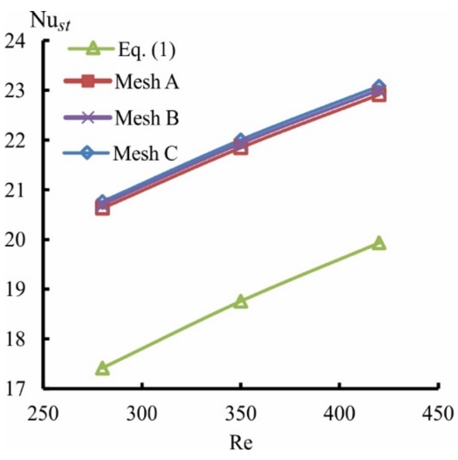

Figure 2. Variation of Nu$_{st}$ with Re for different meshes and equation (1).

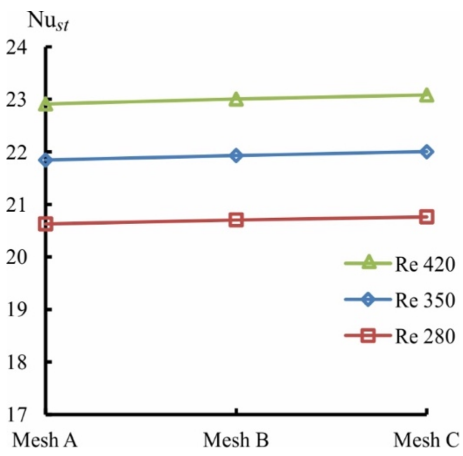

Figure 3. Mesh independence test.

3. Results of numerical simulation

Twelve numerical simulations were calculated for different Reynolds numbers Re, pulsation amplitudes *A*/*D* and Strouhal numbers Sh=fD/u_{inlet}. The parameters of the numerical experiment are shown in Table 1. Figures 4, 5 show the results of numerical simulation. An increase in heat transfer in a pulsating flow is proportional to an increase in the amplitude of pulsations, which is consistent with the data of the previous study [22]. In this study [22] 3*D* pipe with a length of 12*D* was numerically investigated. However, the maximum intensification of heat transfer was 10.5%, which is significantly less than 303% obtained in [22]. The difference in the achieved enhancement of heat transfer can be explained by the fact that in [22] the Strouhal number was 1.13, which is much larger than this study.

Table 1. Parameters of the numerical experiment.

№	1	2	3	4	5	6	7	8	9	10	11	12
Re	280	350	420	280	350	420	280	350	420	280	350	420
A*/*D	11	11	11	15	15	15	19	19	19	23	23	23
ψ	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
***f*, Hz**	1	1	1	1	1	1	1	1	1	1	1	1
Sh	0.032	0.025	0.021	0.032	0.025	0.021	0.032	0.025	0.021	0.032	0.025	0.021

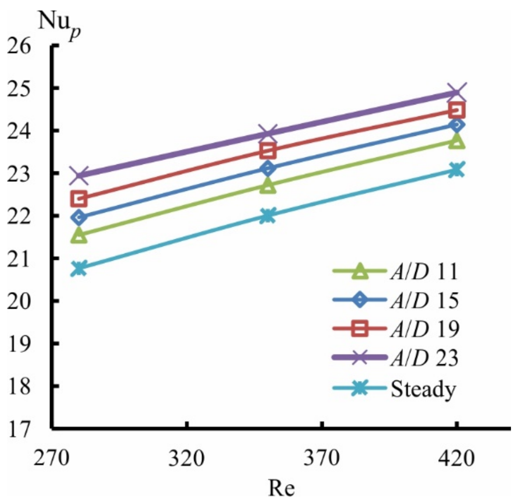

Figure 4. Variation of Nu_p number with Reynolds number Re for steady and pulsating flow.

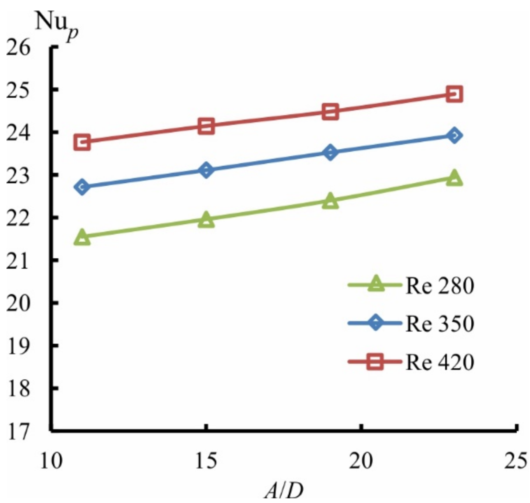

Figure 5. Variation of Nu_p number with amplitude of pulsation *A*/*D*.

4. Evaluation of the reduction in the heat exchange area of oil cooler

The estimation of the decrease in the heat exchange surface of the oil cooler with flow pulsations was carried out on a shell-and-tube heat exchanger with a staggered arrangement of a tube bundle. The coolant on the tube side was oil, on the shell side was water. The characteristics of the heat exchanger are shown in Table 2. The method for calculating the oil cooler with flow pulsations is given below. The required heating capacity of the oil cooler was calculated using the formula (2). The area of the oil cooler for stationary flow was calculated by the formula (3), for pulsating flow by formula (4). The overall heat transfer coefficient for a stationary flow was calculated using formula (5), for a pulsating flow using formula (6).

Figures 6,7 show the results calculated by formulas (2)–(6). The required heat exchange area to maintain heating capacity according to formula (2) decreases with an increase in the amplitude of pulsations and increases with an increase in the Reynolds number. A decrease in the heat exchange area with an increase in the Reynolds number is associated with a decrease in the Strouhal number (see table 1). The overall heat transfer coefficient (6) significantly depends on the heat transfer of the oil, which is laminar flow. Therefore, the percentage decrease in the heat exchange area with a pulsating flow is consistent with the percentage of heat transfer intensification but is not equal to it. For example, at Re 280 and A/D 15, the increase in heat transfer and the decrease in the heat transfer area were 5.75% and 4.95%, respectively.

Table 2. Parameters of the heat exchanger.

Volume flow rate of oil, m^3/h	30
Heat carrier in shell side	water
Heat carrier in tube side	oil
Oil inlet temperature T_1, °C	55
Oil outlet temperature, T_2°C	45
Logarithmic mean temperature difference ΔT_{lm}, °C	14

$$Q = c_p \dot{m}(T_1 - T_2) \tag{2}$$

where $\dot{m}$ – mass flow rate of oil, kg/s.

$$A_{st} = \frac{Q}{\Delta T_{lm} U_{st}}, \text{ m}^2. \tag{3}$$

$$A_p = \frac{Q}{\Delta T_{lm} U_p}, \text{ m}^2. \tag{4}$$

$$U_{st} = \frac{1}{\frac{1}{h_w} + \frac{L}{k_{wall}} + \frac{1}{h_{oil(st)}}}, \text{ W/m}^2\text{°C}, \tag{5}$$

where L – tube wall thickness, m; k – thermal conductivity of the tube material, W/m°C; h_w – heat transfer of water in shell side, W/m^2°C; $h_{oil(st)}$ – heat transfer of oil in tube side for steady flow, W/m^2°C. The $h_{oil(st)}$ in equation (5) was calculated as follows

$$h_{oil(st)} = \frac{Nu_{st} k_{oil}}{D},$$

where k_{oil} – thermal conductivity of oil, W/m°C; Nu_{st} – the Nusselt number for a stationary oil flow was calculated by the equation (1). The h_w in equation (5) was calculated by the formula for the flow in a staggered tube bundle at Re > 1000 [23]

$$h_w = 0.6 \mathrm{Re}_w^{0.5} \mathrm{Pr}_w^{0.36} \frac{k_w}{D_o},$$

where k_w – thermal conductivity of water, W/m°C; D_o – outlet diameter of tube, m; Pr_w – Prandtl number of water (Pr_w was 4.76); Re_w – Reynolds number of water (Re_w was 15584 at water velocity at water gap velocity in tube bundle 0.7 m/s).

$$U_p = \frac{1}{\frac{1}{h_w} + \frac{L}{k} + \frac{1}{h_{oil(p)}}}, \text{W/m}^2\text{°C}, \quad (6)$$

where $h_{oil(p)}$ – heat transfer of oil in tube side for pulsating flow was calculated as follows

$$h_{oil(p)} = h_{oil(st)} \frac{\text{Nu}_p}{\text{Nu}_{st}},$$

where Nu_p – Nusselt number in pulsating flow obtained from numerical simulation in this study.

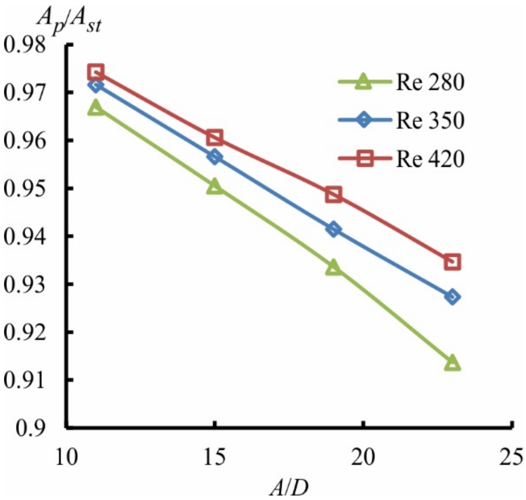

Figure 6. Change of heat exchange area A_p/A_{st} with Reynolds number Re in pulsating flow.

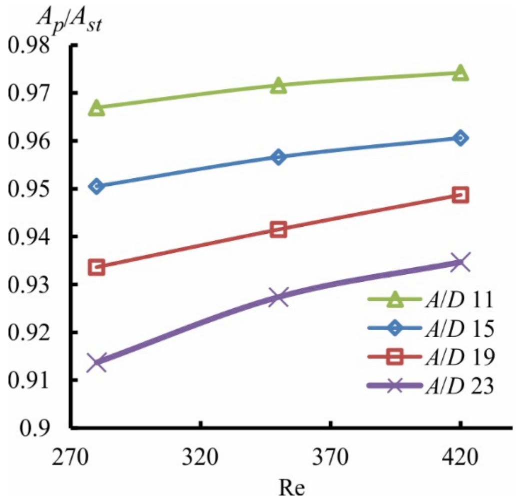

Figure 7. Variation of heat exchange area A_p/A_{st} with amplitude of pulsation A/D.

5. Conclusion

The use of not harmonical flow pulsations leads to an increase in heat transfer up to 10.5% in a pipe with a laminar oil flow. The resulting intensification can be used to reduce the area of tubular heat exchangers in the case of oil flow in the tube side. A decrease in the heat exchange area due to flow pulsations can range from 2.57% to 8.63%, depending on the pulsation mode.

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