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PERFORMANCE ANALYSIS OF DOUBLE-PIPE FINED HEAT EXCHANGER

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ABSTRACT

Double-pipe heat exchanger as simplest solution of this heat transfer devices has found application in chemical industry for almost one century.There exists a variety of constructions of such heat exchangers such as double-pipe straight heat exchanger, double-pipe U-tube heat exchangers and multitube units. All this constructions might be equiped with fins in order to increase heat transfer. Fins applied for these purpose might be of longitudinal, circular, helical or other forms. Current paper tends to bring new insight regarding the optimization of double-pipe heat exchanger with longitudinal fins. Apart of analysis of thermal performance and fluid dynamic characteristics, an analysis from view point of thermodynamical irreversibilities and the overall heat exchanger performance is provided. The optimization aspects are discussed from view point of different equivalent diameter of annulus space, different flow length and different number of fins employed.
Keywords: Double-pipe heat exchanger, fins, performance

1. INTRODUCTION

The double-pipe heat exchanger, consisting of two coaxial pipes, is simple to fabricate and relatively easy for maintenance and cleaning. However, in order to increase the performance of such heat exchanger, commonly longitudinal straight fins have been applied. In that way, increase of heat transfer area is achieved, without excessive increase of pressure drop. Nevertheless, final overall performance depends on number of fins, thickness of fins and cross section area of annular flow space. This because, the increase of number of fins results on larger heat transfer area- means larger heat transfer rate, but it may end in extremely large pressure drop which may overcome benefits related to heat transfer. Hence, in designing process, there exists a kind of trade off between benefits related to heat transfer and drawback related to pressure drop. In order to properly address this issues, one needs to handle with corresponding equation related to heat transfer and pressure drop, but also with thermodynamics of heat transfer process such as analysis of related irreversibility’s.

2. PREDICTION OF HEAT TRANSFER AND PRESSURE DROP

Heat transfer and pressure drop of heat exchangers, is generally presented in terms of dimensionless variables such as Nusselt number (*Nu*) and friction factor (*f*). For the purpose of current paper, following expression were used to calculate heat transfer and pressure drop in double-pipe heat exchanger. For the tube side Nusselt number is calculated based on Gnilienski equation preferred in the literature [1].

$$Nu = \frac{h \cdot d}{k} = \frac{(\xi/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1.0+12.7(\xi/8)^{1/2}\left(\mathrm{Pr}^{2/3}-1\right)},\ 1< \mathrm{Pr} <10^6 \text{ and } 3000< \mathrm{Re} <5\cdot 10^6 \quad (1)$$

Where d= tube diameter, Pr= Prandtl number
Friction factor is calculated based on following Petukhov equation:

$$\xi = \frac{1}{(0.79 \cdot \ln \mathrm{Re} - 1.64)^2} \quad (2)$$

For the finned annular space (shell side) following equation was used for Nusselt number [2]:

$$Nu = 0.01783 \cdot \mathrm{Re}^{0.835} \cdot \mathrm{Pr}^{0.33},\ \mathrm{Re} > 10000 \quad (3)$$

Where d_h= hydraulic diameter of finned annular space, Pr= Prandtl number
Friction factor is estimated based on following equation[3]:

$$f = 0.109 \cdot \mathrm{Re}^{-0.255},\ \mathrm{Re} > 2100 \quad (4)$$

Equivalent diameter (alternatively known as hydraulic diameter) of finned annular space is calculated based on geometry of finned annular space (fig. 2).

Equivalent diameter for calculation of heat transfer is derived based on assumption that the tip of the fin is adiabatic:

$$d_{eh} = \frac{4A_c}{P_{wh}} = \frac{4\left(\pi D_i^2/4 - \pi d_0^2/4 - n_f bh\right)}{\pi d_0 - n_f + 2hn_f} \quad (5)$$

Figure 1. Double-pipe heat exchanger with longitudinal fins

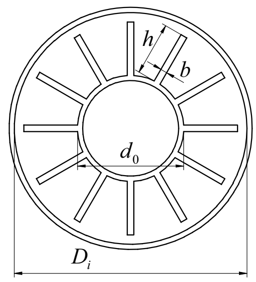

Figure 2.Basic parameters of annular finned space

Equivalent diameter for calculation of pressure drop:

$$d_{ef} = \frac{4A_c}{P_{wh}} = \frac{4\left(\pi D_i^2/4 - \pi d_0^2/4 - n_f bh\right)}{\pi d_0 - n_f + 2hn_f + \pi D_i} \quad (6)$$

Minimum cross section flow area, relevant for calculation of fluid flow velocity which appears in Re is calculated based on following expression:

$$A_{cm} = \pi D_i^2/4 - \pi d_0^2/4 - n_f bh \qquad (7)$$

Heat exchanger is considered to be single pass, hence no pressure drop due to change of the flow direction is considered.

3. IRREVERSIBILITIES OF HEAT EXCHANGER AND OVERALL PERFORMANCE

In general thermodynamic irreversibility's related to operation of a heat exchanger may be attributed to finite temperature differences between heat exchanger working fluids and to finite pressure drop of corresponding fluid streams. Such irreversibility's results in entropy generation which may be used to quantitatively describe the quality of the heat transfer process within a given heat exchanger.
For current analysis, heat exchanger is analyzed under steady state conditions and is considered to be an open adiabatic system with respect to outside boundaries (normally heat exchangers are thermally insulated with respect to surrounding). By considering that both working fluids are incompressible liquids, flowing in opposite directions, following equation may be used for calculation of entropy generation rate on both fluid sides [3,4]:

$$\dot{S}_{gen} = m_h c_{ph} \ln\left(\frac{T_{ho}}{T_{hi}}\right) + m_c c_{pc} \ln\left(\frac{T_{co}}{T_{ci}}\right) + m_h \frac{\Delta p_h}{\rho_h} \frac{\ln\left(\frac{T_{ho}}{T_{hi}}\right)}{T_{ho} - T_{hi}} + m_c \frac{\Delta p_c}{\rho_c} \frac{\ln\left(\frac{T_{co}}{T_{ci}}\right)}{T_{co} - T_{ci}} \qquad (8)$$

Where subscript *h* stays for the hot fluid and *c* for the cold fluid.
Assessing of heat exchanger performance based on level of irreversibility present, usually is performed by comparison of dimensionsless entropy generation rate under different operating conditions. Dimensionless entropy generation known as entropy generation number within current paper is defined as follows:

$$Ns = \frac{\dot{S}_{gen}}{\left(m_c c_{pc}\right)} \qquad (9)$$

In order to demonstrate the value of Ns for evaluation a heat exchanger, it is assumed that current double-pipe heat exchanger is made from steel pipes in counterflow arrangement and is used to heat Dowtherm (a heat transfer liquid) with an inlet temperature of 15 °C using waste hot water with inlet temperature of 95 °C. Further it has been assumed that colder fluid (Dowtherm) is flowing through annulus part of heat exchangers, employing longitudinal fins made from steel (fig. 2) whereas the hoter fluid (water) is flowing in opposite direction through inner pipe. Fluid flow velocity of dowtherm has been assumed to vary from 3 m/s to 14 m/s for which Re>10000 are obtained it means eq. 3 might be used to derive Nu on annulus side. Fluid flow velocity of water has been assumed to vary from 0.5 m/s to 5 m/s, which are common in such systems and for which Re number within the range of eq. 1 are obtained. Fluid flow properties were considered constant. The values of such properties used for calculations were derived based on mean temperatures of fluids (table 1), [4]:

Table 1. Physical properties of heat exchanger fluids

	Cold fluid (dowtherm)	Hot fluid (water)
Mean temperature (°C)	40	85
Density (kg/m^3)	1044	969
Specific heat (J/kgK)	1622	4197
Thermal conductivity (W/mK)	0.138	0.676
Kinematic viscosity (m^2/s)	$2.59 \cdot 10^{-6}$	$3.21 \cdot 10^{-7}$
Prandtl number (-)	31.7	1.93

Calculations were performed for different number of fins and different flow length. It should be noted that based on eq. 5 and 6, different number of fins results in different equivalent diameter. In fig. 3 are presented only results for 16 and 32 fins for flow length of 5 m, 10 m, 15 m and 20 m:

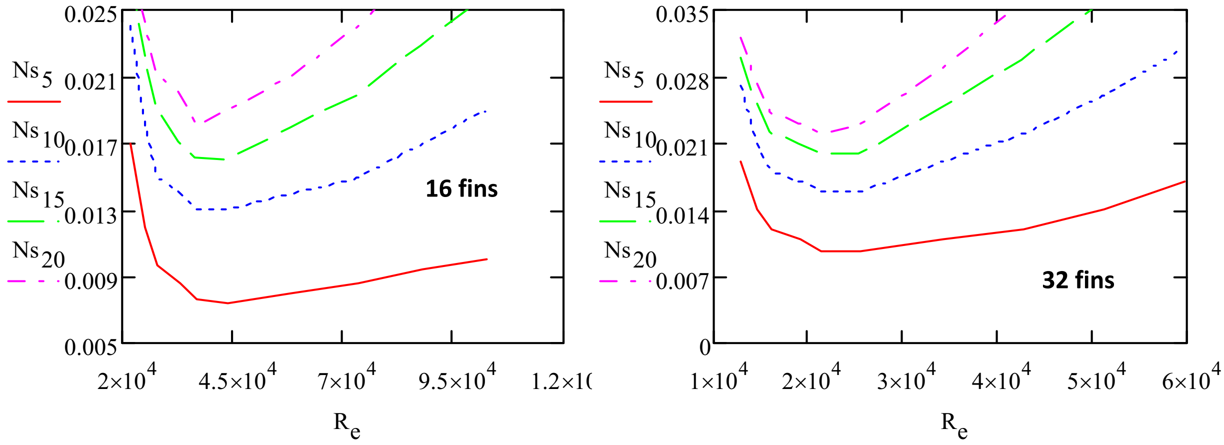

Figure 3. Variation of Ns over Re for different flow length and different number of fins

Overall performance of current double-pipe heat exchanger is assessed by direct plot of the heat transfer rate per unit volume $\dot{q}_v$ versus the required pumping power per unit heat exchanger volume e_v [5]

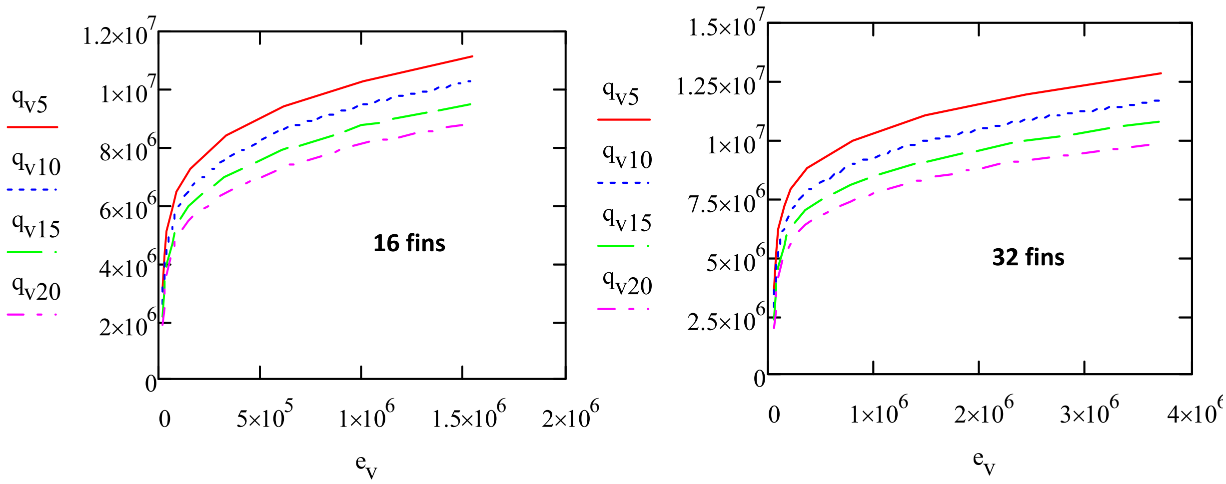

Figure 4. Comparison of double-pipe heat exchanger performance based on variation of heat transfer rate over pumping power

4. CONCLUSIONS

Current work demonstrates that a minimum entropy generation number Ns can be found for each flow length of double pipe-heat exchanger and for each number of fins employed. Such minimum indicates the flow regime for which the heat exchanger perfumes optimally from view point of thermodynamics. In addition, fig. 4 demonstrates that heat exchanger configurations with lowest entropy generation number results in a better overall performance, means higher heat transfer rate for same pumping power.

5. REFERENCES

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