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Design & Thermal Analysis of Double Pipe Heat Exchanger by Changing the Mass Flow Rate

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Abstract- The project is based on Voice Personal Assistant (VPA) it is a digital assistant that uses voice recognition, natural language processing and speech synthesis to provide aid to users through voice recognition applications. One of the most studied and popular was the direction of interaction, based on the understanding of the machine by the machine of the natural human language. It is no longer a human who learns to communicate with a machine, but a machine learns to communicate with a human, exploring his actions, habits, behavior and trying to become his personalized assistant.

Keywords- Pyttsh, Python, speech recognition.

I. INTRODUCTION

Heat exchangers are employed in a variety of applications, included power plants, and nuclear reactors in energy production, RAC systems, self-propelled industries, food industries, heat retrieval systems, & chemical handling. The techniques of upgrading can be divided into two categories: active and passive ways. The active approach necessitates the use of peripheral forces. Discrete surface geometries are required for passive approaches. These strategies are commonly utilized to increase heat exchanger performance. Helical tubes have already been designated as among the passive heat transfer enhancement materials. Due the short construction and high heat transfer coefficient, and they will be widely employed in various industrial applications.

The development of high-performance thermal systems has sparked interest in heat-transfer technologies. In heat exchangers, raising the convection heat transfer coefficient or expanding the convection surface area improves heat transmission. Inserts inside these pipes/tubes are one means of increasing the convection coefficient within such a heat exchanger.

Heat exchangers are devices that allow energy to be transferred between two fluids of different temperatures. A heat exchanger takes the use of fact that energy flows when there has been a difference in temperature. As a result, heat will be transferred from the higher-temperature heat reservoir to the lower-temperature heat reservoir.

The temperature differential created by the circulating fluids forces the energy to move between both. The energy passing through a heat exchanger might be sensible or latent heat of flowing fluids.The fluid that provides the energy is referred to as hot fluid. Cold fluid is a type of fluid that receives energy."

In such a heat exchanger, the temperature of the hot fluid should drop whereas the temperature of the cold would increase. The goal of a heat exchanger would be to either heat or cool the fluid.

Whenever one of the fluids experiences a phase shift, the temperature of the other fluid remains unaltered. Condensers and evaporators are two different types of heat exchangers. Heat exchangers featuring convective fluid heat transfer inside tubes are commonly used in a variety of technical applications. Heat transfer improvement strategies to accommodate high heat flux, i.e., to minimise the size and expense of heat exchangers, has gotten a lot of attention in recent years. Heat transfer improvement the rate of all sorts of thermos-technical apparatus is extremely important to the industry. It results in reduced in size and weight in addition to conserving basic energy.

Many heat transfer improvement techniques have been developed up to this point. Twisted-tape is among the most essential members of a improvement techniques used in heat exchangers.

II. METHODOLOGY

The experiment is carried out in a two-pipe heat exchanger with the characteristics listed in the table below:-

Table 1: Structural parameters (Maakoul et al. 2017).

Fin material	Aluminium
Fin number, N	10
Fin width, b	0.6 mm
Inner tube internal diameter Di	8 mm
Inner tube external diameter Do	10 mm
annulus internal diameter Ds	20 mm
Tube length, L	100 mm

1. Heat Transfer Calculations:
Firstly, it should be noted that for all calculations, the thermos-physicalproperties of the water in the annulus and the internal tubewere calculated atT_a^{out} and T_b^{out} respectively.

The Reynolds number determined by:

$$\mathbf{Re} = \frac{\rho u D_h}{\mu}$$

With μ the dynamic viscosity in the annulus side, D_h equivalenthydraulic diameter, ρ water or air density.

For the inner tube $D_h= D_i$, while for the finned annulus, thehydraulic diameter is:

$$\mathbf{Dh} = \frac{4[\frac{\pi(D_s^2-D_a^2)-NBH}{4}}{N(2H+b)+\pi D_o-Nb}$$

Ds are the annulus internal diameter, Do the inner tube external diameter, N the number of fins, b the fins width and H the finsheight.

2. Heat transfer rate:
Heat transfer rate of the annulus side fluid (air):

$$\mathbf{Q_a = m_a C_{pa} (T_a^{out} \text{ and } T_a^{in})}$$

Heat transfer rate of the tube side fluid (water):

$$\mathbf{Q_t = m_{tb} C_{p,tb} (T_{tb}^{in} \text{ and } T_{tb}^{out})}$$

m_ T is the temperature, while cp is indeed the specific heat. The subscripts a; tb denote the annulus or inner tube sides, etc; the superscripts 'in'and'out' denote the values at the inlet and outflow, respectively.

For every side of the compressor, the physical parameters were assessed just at average temperature of the input and outflow.

The annular side's Nusselt number is:

$$\mathbf{Nu} = \frac{h_a D_h}{\gamma_a}$$

The annulus side heat transfer coefficient equals ha, while the gas thermal conductivity equals ka.

For the tube side, its Nusselt number equals:

$$\mathbf{Nu} = \frac{h_i D_i}{\gamma_i}$$

Where, h_i is the tube side heat transfer coefficient, and k_i the fluidthermal conductivity.

III. CFD MODELLING

CFD is indeed a strong tool for predicting fluid motion in diverse conditions, allowing for correct design, thanks to developments in computer technology and accessible numerical algorithms. CFD is just a sophisticated method of analysing not just too fluid flow behaviour and yet also heat or mass transport processes.

The below governing incompressible fluid flow equations can also be used to explain the steady-state fluid flow characteristic in the three-dimensional computational domain.

Continuity equation:

$$\frac{\partial u_i}{\partial x_i} = 0$$

Momentum balance without gravity force:

$$\rho u_i \frac{\partial u_i}{\partial x_i} = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j}$$

Energy equation:

$$\rho c_p \frac{\partial (u_i T)}{\partial x_i} = \frac{\partial}{\partial x}\left(\lambda_{\text{eff}} \frac{\partial T}{\partial x_i}\right)$$

The partial derivatives again for RNG k model were turbulent kinetic equations given conservative version:

$$\rho u_j \frac{\partial k}{\partial x_i} = \frac{\partial}{\partial x_j}\left[\mu_{\text{eff}}\, \alpha_k \frac{\partial k}{\partial x_j}\right] + \tau_{ij}\frac{\partial u_i}{\partial x_j} - \rho\varepsilon + G_k + G_b$$

Turbulent kinetic dissipation equation

$$\rho u_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\mu_{\text{eff}}\, \alpha_\varepsilon \frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon}^{*}\rho\frac{\varepsilon^2}{k}$$

The above Reynolds Averaged Navier-Stokes (RANS) turbulence models offer the most economic approach for computing complex turbulent industrial flows.

The Navier-Stokes equations, in general, describe the turbulent flow motion. Solving these equations for complex flow issues, on the other hand, is prohibitively expensive and time-consuming [26].

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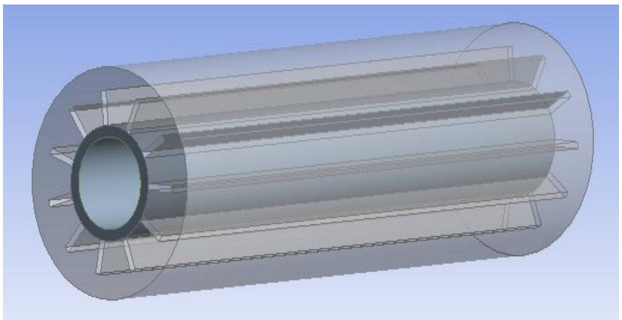
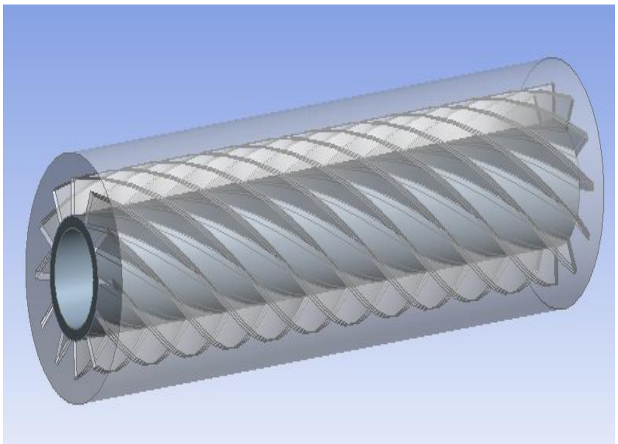
Alternatively, two strategies have already been proposed: (i) Big Eddy Simulation (LES), which simulates large energy-containing, eddy directly whilst accounting for tiny eddies by averaging.

The following steps are required for the separation of big and small eddies: (ii) Reynolds averaging (RANS), in which all eddies were accounted for it by Reynolds stresses produced by averaging the Navier-Stokes equations. (time averaging for statistically steady flows, ensemble averaging for unsteady flows).

1. Computational Domain:
The aim of this analysis is to quantitatively examine and evaluate alternative straight and helical fin designs in a twin pipe heat exchanger. Air and water are indeed the working fluids, with cold air flowing with in annulus & hot water flowing in the inner tube in such a counter-current arrangement. Three double pipe heat exchangers with longitudinal fins were utilised to validate this numerical model, and three having helical fins having variable fins were explored.



(a)



(b)
Fig 1. CFD domain.

2. Boundary Conditions:
Several basic conditions of both the physical model must be established adequately in order to analyse the heat & momentum transfer of DPHEs. The boundary conditions including all borders of both the computing domain must be established for numerical stance on the issue. The water entering this computational domain there at upstream boundary conditions was considered to be have uniform velocity U_{in}, temperature T_{in} (333.15 ºC) with a high level of turbulence I (1 percent). In the y and z axes, the velocity components were assumed to be zero.

The entry, outflow, and bundle zone make up the fluid area. The fin is included solid section. No-slip criteria for velocity were defined at solid surfaces. Heat convection to the fin and heat conduction inside the fin is both taken into account.

Assume a zero heatflux in the symmetry planes. There is also no convective flux over the symmetry plane because the normal velocity component equals zero. Temperature gradients or tangential components of velocity gradients within normal direction are therefore set to zero.

2.1 Inlet:

$$U = U_{in} = \text{constant}$$
$$T = T_{in} = \text{constant}$$
$$I = 1\%$$

2.2 Outlet:

Static pressure

2.3 Tube:

no-slip condition

$$T = T_w = \text{constant}$$

2.4 Fin:

no-slip condition

Coupling of conduction and convection

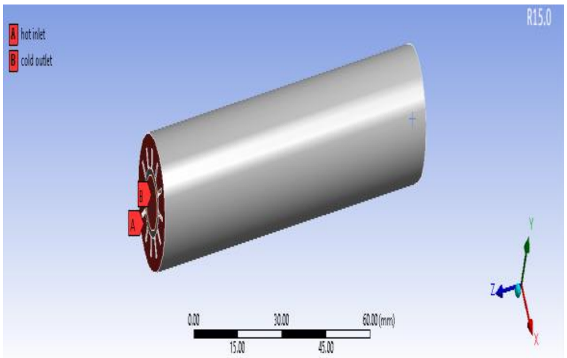


Fig 2. Boundary condition in CFD.

IV. RESULTS AND DISCUSSION

1. CFD Code Validation:
The heat transfer and pressure drop properties of a double-pipe heat exchanger with helical fins were evaluated using a computer model. The findings of this numerical model were compared to experimental data collected using empirical correlations under turbulent flow conditions for a double-pipe heat exchanger without longitudinal fins.

The Reynolds number Re inside the finned annulus (air) were altered from 12700 to 17700, while it was held constant at Re = 12000 in the inner tube. The heat transmission and pressure drop parameters calculated numerically are compared to experimental data for such an annulus having longitudinal fins previously published [28]. Further, the correlations derived from some other investigation [29] were compared to the numerically calculated inner tube parameters. The validation was carried out utilising the following correlations with appropriate geometric parameters or hydraulic diameter.

For an annulus with longitudinal fins:

$$j_h = (0.0263Re^{0:9145} + 4.9 - 7Re^{2.618})^{1/3}$$

$$f = 576 \exp [0:08172(\ln Re)^2 – 1.7434 \ln Re – 0.6806]$$

j_i is the Colburn factor and f is the friction factor. It is worth noting that under turbulent flow conditions, and within the considered Re range, jh, and consequently h, is independent of the number of fins.

2. Inner tube [29]:

$$Nu = 0.023Re^{0.8}Pr^{1/3}$$

$$f =(0:79 \ln(Re) - 1.64)^{-2}$$

Fig. 3 shows a comparison of the annulus side heat transfer coefficients obtained by the CFD model and the empirical correlation as a function of Re for longitudinal fins. The average deviation of the finned-annulus-side heat transfer coefficient is 20%.

Similarly, Fig. 4 shows a comparison of the pressure drops obtained by the CFD model and the empirical correlation as a function ofRe. The pressure decrease has a maximum relative variation of 2%. In addition, the inner-tube heat transfer coefficient & pressure drop both have relative variations of less than 3% and 1%, respectively.

Based on the comparisons above, it could be determined that now the numerical model produces accurate results with an acceptable degree of precision. The model is then used to forecast the heat transfer & thermos-hydraulic performance of a double-pipe heat exchanger with helical fins for various configurations.

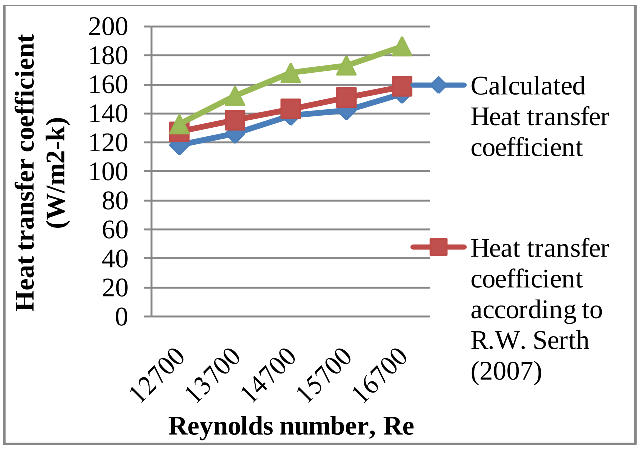


Fig 3. “Plot of heat transfer coefficient versus Re for longitudinal fins”.

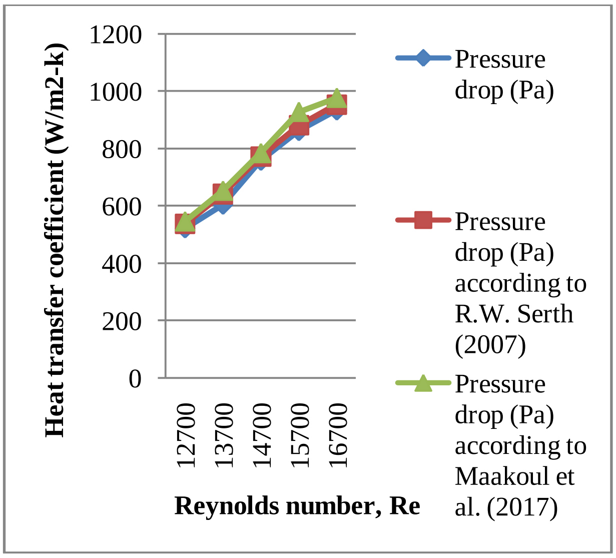


Fig 4. “Plot of pressure drop versus Re for longitudinal fins.

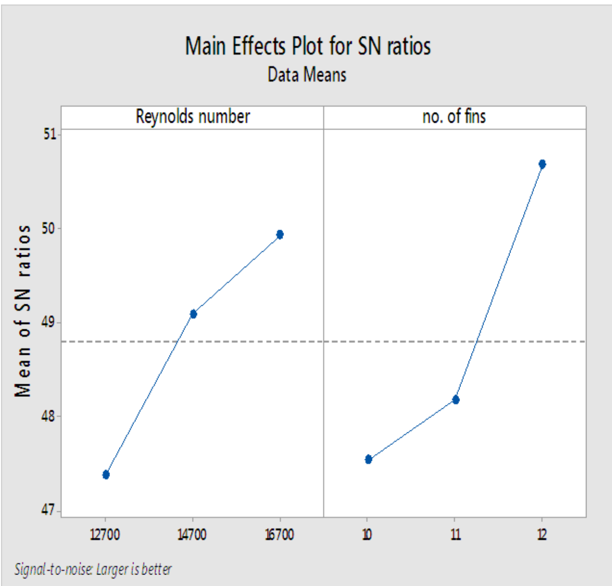


Fig 5. Main-effect plots for Heat transfer coefficient.

It is evident from the above graph that Reynolds number has comparatively less effect on **Heat transfer coefficient**. Although **Heat transfer coefficient** is associated with both Reynolds number and fin number. Fin Reynolds number. and fin number have dominant influence with contribution ratios of 49%, and 50% respectively From Fig. 5.2, the optimal combination for SNR- ηf is determined as A3B3.

V. CONCLUSION

The thermo-hydraulic performance of various configurations of gas-to-liquid double-pipe heat exchangers featuring helical fins was investigated that used a computational model based on CFD. The heat transmission, pressure drop, unit weight, and overall performance of helical or longitudinal fin configurations studied numerically simulated and the results. The effects of increasing the number of fins and Reynolds number on thermos-hydraulic performance also were investigated.

The following conclusions are reached as a result of the findings.

- It is clear that now the Reynolds number has a smaller impact across all parameters, including heat transfer coefficient, Nusselt number, heat transfer rate, pressure drop, and thermal performance enhancement, than that of the number of fins. The ideal factor combination for SNR-ηf is determined to be A3B3.
- Helical fins have a larger surface area for heat transmission than longitudinal fins.
- Overall, helical fins outperform longitudinal fins for thermohydraulic performance of the double heat exchangers, indicating that now the pressure loss of helical fins was countered either by increased heat transfer rate.
- Finally, the helical fin arrangement for double-pipe heat exchangers has been shown to be helpful in boosting heat transmission and thermohydraulic performance.

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