International Journal of Scientific Research & Engineering Trends
Volume 9, Issue 5, Sep-Oct-2023, ISSN (Online): 2395-566X

CFD Analysis of Double Pipe Heat Exchanger

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Abstract- Heat exchangers are employed in a variety of applications, included power plants, 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.

Keywords- Thermal performance, ANSYS, CFD, double pipe heat exchanger.

I. INTRODUCTION

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. DOUBLE PIPE HEAT EXCHANGER

Double pipe heat exchangers are widely employed in a variety of industrial processes & research fields, including air coolers, waste heat recovery and conversion systems, including chemical process heating. Different heat transfer enhancement techniques had already recently been included in the aforementioned applications; these techniques can indeed be beneficial from such a practical standpoint, as well as about their implementation could result in energy savings, time savings, an increase throughout thermal ratings, and an extension of both the equipment's working life [1].

A double pipe heat exchanger (also known as a 'pipe-in-pipe' heat exchanger) is a form of heat exchanger with a 'tube in tube' configuration. It comprises of 2 pipes, one within the other, even as name implies. One fluid flows via the inner pipe (which would be equivalent towards the tube-side in some kind of a shell and tube type exchanger) and the other thru the outer pipe that also surrounds the inner pipe (analogous to the shell-side in a shell and tube exchanger). This is what a cross-section of a twin pipe exchanger might look like:

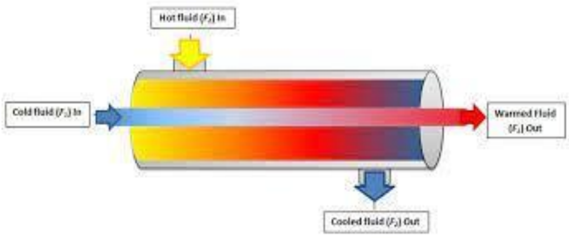


Fig 1. Double pipe heat exchanger.

International Journal of Scientific Research & Engineering Trends
Volume 9, Issue 5, Sep-Oct-2023, ISSN (Online): 2395-566X

They usually feature a U-tube construction to support tube thermal expansion without any need for expansion joints, as seen below:

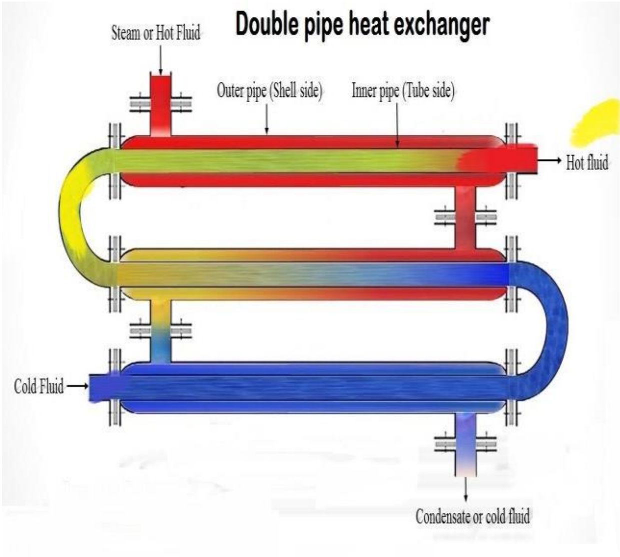


Fig 2. U-typedouble pipe heat exchanger.

These are amongst the most basic and inexpensive heat exchangers. They're suitable for high-temperature, high-pressure, or viscous applications.

The double pipe heat exchanger (DPHE) is amongst the most basic and often used heat exchangers (Fig. 1.2). The chemical, food, oil, or gas sectors all employ this type of heat exchanger. Because of the comparatively tiny diameter, numerous detailed studies have indicated that this process of heat exchanger is employed in high-pressure applications. They're also crucial when a wide variety of temperatures is required. That type of heat exchanger had been shown to contribute significantly to pasteurisation, reheating, preheating, digester heating, or effluent heating operations. Because of its inexpensive design and maintenance costs, DPHEs are also used by small businesses. As a consequence, they reached the conclusion the prior research on just this type of heat exchanger must be classified in order to eliminate the confusion of selecting the most suited techniques of interest.

A basic gas-to-liquid double-pipe heat exchanger is made up of two pairs of concentric pipes, with fluids flowing in opposing directions thru the exchanger (counter-current flow). The heat transfer coefficient upon that liquid side of gas-to-liquid heat exchangers would be typically one awful lot higher than on the gas side, limiting the heat transfer rate. Here on gas side, expanded surfaces known as fins have been used to improve heat transfer surfacearea and hence boost heat transfer capacity; That enhances convective heat transfer through lowering the gas side's thermal resistance, though at the cost of a higher pressure drop [3]. “Fin type and shape are important elements in improving the thermohydraulic performance of gas-to-liquid heat exchangers, because fins may improve the heat transfer surface area by a factor of 5–30 depending on the required [4].”

III. METHODOLOGY

1. Specifications of Double Pipe Heat Exchanger Used:

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

2. Heat Transfer Calculations:

Firstly, it should be noted that for all calculations, the thermos-physical properties of the water in the annulus and the internal tube were calculated at T_a^{out} and T_b^{out} respectively.

The Reynolds number determined by:

$$\mathbf{Re = \rho u Dh}$$

With μ the dynamic viscosity in the annulus side, D_h equivalent hydraulic diameter, ρ water or air density. For the inner tube $D_h = D_i$, while for the finned annulus, the hydraulic diameter is:

$$\mathbf{Dh = 4[Ds2-Da2-NBH4N2H+b+\pi Do-Nb}$$

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

$$\mathbf{Nu = haDha}$$

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

For the tube side, its Nusselt number equals:

Nu = hiDii

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

IV. CFD MODELLING

1. Numerical Consideration:
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.

2. Governing Equation:
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

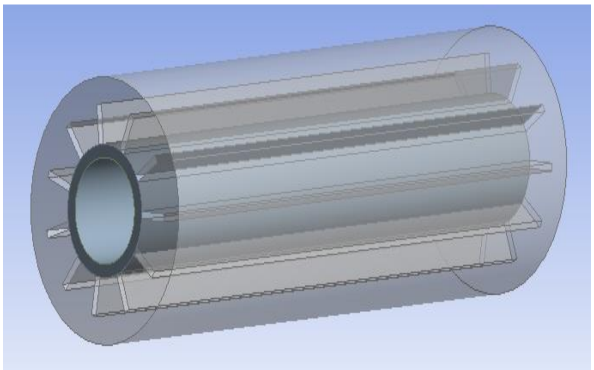
$$\rho u_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\mu_{\text{eff}}\, \alpha_\varepsilon \frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}^{*}\rho\frac{\varepsilon^2}{k}$$

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

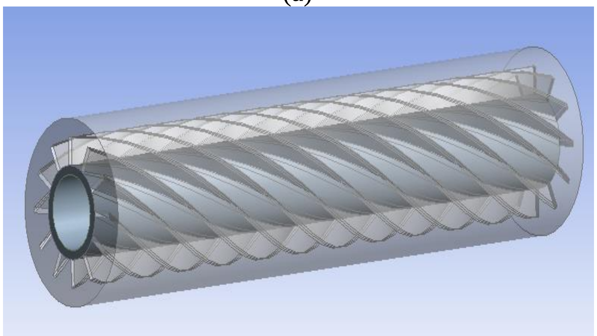
The Navier-Stokes equations, in general, describe the turbulent flow motion. Solving these equations for complex flow issues, on the other hand, is prohibitively expensive and time-consuming [26].

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).

3. 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 3. CFD domain.

4. 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.

5. Meshing of Domain

The computational domain was discretized it in to a finite number of control volumes using a generalized curve linear coordinate grid generating technique based on body–fitted coordinates in just this work. The issues' geometries are meticulously created. The GAMBIT [12] was used to model or mesh all of the situations. FLUENT also has a CFD application that gives the user entire control over how to accommodate suitable complicated geometry.

The grid system's refinement and generation are critical for predicting heat transport in complicated geometries. To put it another way, the density and dispersion of grid lines are critical in generating accuracy. Because mean flow and turbulence interact so strongly, numerical results of turbulent flows are more reliant on grid optimisation than those for laminar flows [11].

V. RESULTS AND DISCUSSION

1. Tube with Straight Fins:

Table 2: Simulation runs corresponding to tube with straight fins

Reynolds number	no. of fins	Heat transfer coefficient (W/m^2-k)	Nusselt number (Nu)	heat transfer rate (Q_{HF}/Q_{HL})	Pressure drop (Pa)	Thermal performance enhancement
12700	10	198.81	308.78	1.105	102.03	1.166
12700	11	235.36	366.81	1.02	151.24	1.270
12700	12	265.32	784.32	1.034	192.22	1.292
14700	10	239.18	440	1.085	168.3	1.228
14700	11	249.11	581.15	1.09	257.17	1.293
14700	12	379.21	880.18	1.02	382.4	1.405
16700	10	276.22	643	1.035	235.35	1.247
16700	11	280.6	686.22	1.054	335.15	1.453
16700	12	389.6	946.6	1.013	539.96	1.487

2. Tube with Helical Fins:

This section presents the simulation results for double pipe heat exchanger with helical fins.

Table 3. Simulation runs corresponding to tube with helical fins.

Reynolds number	no. of fins	Heat transfer coefficient (W/m^2-k)	Nusselt number (Nu)	Heat transfer rate (Q_{HF}/Q_{HL})	Pressure drop (Pa)	Thermal performance enhancement
12700	10	220.34	334.33	1.105	216.9	1.271
12700	11	240.51	471.98	1.02	323.93	1.536
12700	12	275	871.93	1.034	433.46	1.338
14700	10	220.34	519.17	1.085	411.56	1.468
14700	11	240.51	581.2	1.09	455.24	1.093
14700	12	275	932.44	1.02	537.57	1.071
16700	10	286.39	667.22	1.035	601.13	1.298
16700	11	296.55	736.11	1.054	688.44	1.245
16700	12	395.34	956.22	1.013	723.76	0.998

VI. CONCLUSION

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. 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. A DPHE helical 12 fin provides the highest thermal enhancement factor. Thus, the helical fins designs increase thermohydraulic performance, but the 10 fin arrangement reduces it. Despite its low rank, the DPHE with helical 12 fin design gives the best thermal improvement, and should be used where heat transfer capability was critical but pressure drop is minimal.

Furthermore, as even the helical route lengthens, the swirl and secondary flows expand as a result of increased centrifugal force. They disrupt the water boundary barrier and produce turbulence along the outer surface of something like the internal tube. As both a result, there seems to be a significant increase in heat transmission as well as a significant rise in pressure drop.

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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