DESIGN AND ANALYSIS OF PLATE HEAT EXCHANGER WITH CO₂ AND R134a AS WORKING FLUIDS

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ABSTRACT

The purpose of this study is to design a optimal plate type heat exchanger for refrigeration systems. For many industrial applications plate heat exchangers are demonstrating a large superiority over the other types of heat exchangers. In this paper we present both theoretical and simulation analysis for a parallel flow heat exchanger. In this paper we performed analysis using two different working fluids which are i.e., CO₂ and R134a used only for refrigeration systems. We got the results regarding heat transfer and also observed a very low pressure drop. By knowing the inlet conditions like mass flow rate, temperature, pressure of both hot and cold fluid streams, we performed CFD analysis in ANSYS software. The optimal design of multi-pass plate type heat exchanger is designed by optimizing the variables such as number of passes for both streams, no. of plates, plate type and size. In this paper we formulated the equations of heat transfer, pressure drop, overall heat transfer coefficient etc. The selection of suitable material is taken based on thermal aging of materials, stresses induced at high temperatures, failure or fatigue mode from the previous works. After obtaining all these values the model is simulated in ANSYS, where we obtained better and accurate results compared to theoretical analysis.

Key Words: ANSYS, CFD, simulation, PHE, Meshing, Nusselt number, Prandtl number.

INTRODUCTION

In many applications like air conditioning, refrigeration, heat recovery industries like thermal, nuclear power industries, heat exchangers are used to transfer energy from one fluid to another mostly used heat exchangers are boilers, condensers, economizers etc. in thermal and nuclear power industries. Car radiators are also heat exchangers which transfer heat from IC engine to air or water. For efficient heat recovery mostly compact type heat exchangers are used one of them has high heat transfer coefficient which is known as plate type heat exchanger. It is used in many industries because of its efficient heat recovery, light weight, compact design, easy to maintain, less floor space than compared to other heat exchangers. Minimal maintenance, cost effectiveness and especially high efficiency are the main factors to select plate type heat exchanger. To develop a heat exchanger
in this field is a big challenge to many researchers and authors in terms of efficiency and economic considerations [2,3,4]. Many studies were carried out on multichannel PHE’s by authors and presented it in a simple thermal model. The thermal and hydraulic performance of PHE with certain size and type of corrugations can be varied with two ways by adjusting number of passes and proper selection of corrugation pattern. The optimal design of PHE by adjusting corrugation pattern was done by Wang and sunden. Regarding the material selection YorikataMizokami, ToshihideIgari et al [5] done analysis on materials that effect due to thermal aging, stresses and temperatures, which obtained best results. The mathematical modelling of PHE was done by H. Dardour, S. Mazouz et al [6]. The properties of CO\textsubscript{2} and R134a are obtained from HMT data book by C.P.Kothandaraman [7].

The main approach of this paper is done through by calculating temperatures at each channels by fixing the both inlet conditions of hot and cold fluid streams, calculated theoretically and compared to simulated results which obtained similar values. Increasing or decreasing the number of plates has been done according to the expected simulation results.

PLATE HEAT EXCHANGER DESCRIPTION

Plate heat exchanger consists of stack of metal plates which are made up certain material and gap between successive plates is the channels for liquid to flow along the plates. Plate separates both hot and cold fluids which allows heat transfer to be carried out. The first and last plates have fluid only on one side. The heat transfer is carried out in parallel flow process through the channels where cold fluid becomes warmer and hot fluid becomes cooler.

HEAT EXCHANGER BASIC FORMULATION

For prediction of performance of PHE, when inlet and outlet conditions and overall heat transfer coefficients are known it is better to use NTU- effectiveness method.

\[ NTU = \frac{UA}{C_{min}} \]

Where \( C_{min} \) is the minimum value of \((m \times c_p)\). A= (no.of plates \times area per plate), U is the overall heat transfer coefficient. The dimensionless effectiveness of PHE is defined as ratio of actual heat transfer to the maximum heat transfer.

\[ \varepsilon = \frac{C_c (T_{c,out} - T_{c,in})}{C_{min}(T_{h,in} - T_{c,in})} \]

Where \( h_{in} \) and \( C_{in} \) denotes hot and cold fluid inlet data.
SOLIDWORKS MODELLING

To analyse the temperature variations in the plate type heat exchanger model is modelled in SOLIDWORKS according to the new design and the required length, width, height, No. of plates, and thickness of the plates. The exploded view of different parts concerning the heat exchanger are shown in the figure.

COMPUTATIONAL MODEL

To analyse the temperature variations in the heat exchanger a computational model of only the fluid part is modelled with required no of plates and the part is modelled only of the fluid path inside the plate and the directional losses are neglected in the analysis.
MATHEMATICAL MODELLING

Hypothesis
A set of assumptions must be introduced to develop a mathematical model of PHE.

The set of assumptions are

- The plate heat exchanger operates under steady state conditions.
- No phase change occurs: both fluids are single phase and are unmixed.
- Heat losses are negligible.
- The temperature in the fluid streams is uniform over the flow cross section.
- There is no thermal energy source or sink in the heat exchanger.
- The fluids have constant specific heats.
- The fouling resistance is negligible.

Problem formulation and Governing equations

In PHE the two fluids exchange heat energy through the separating plates. Applying steady flow energy conservation equation between the two plates gives:

For the first channel:

\[
\text{Change in } H + \dot{q}_2 = 0
\]

where \( \dot{q}_2 \) is the heat flux per unit length, \( H \) is the enthalpy of the fluid.

\[
\dot{q}_2 = UA(T_2 - T_1)
\]

where \( U \) is the overall heat transfer coefficient, \( A \) is the surface area of the plate. Similar equations are also established for the others channels.

<table>
<thead>
<tr>
<th>OPERATING CONDITIONS</th>
<th>Hot side</th>
<th>Cold side</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate [kg/s]</td>
<td>0.2</td>
<td>0.2</td>
</tr>
<tr>
<td>Inlet temperatures [K]</td>
<td>288</td>
<td>273</td>
</tr>
</tbody>
</table>

**Heat exchanger data**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate length[m]</td>
<td>1</td>
</tr>
<tr>
<td>Plate width[m]</td>
<td>0.3</td>
</tr>
<tr>
<td>Plate thickness[m]</td>
<td>0.002</td>
</tr>
<tr>
<td>Distance between the plates[m]</td>
<td>0.01</td>
</tr>
<tr>
<td>Plate material</td>
<td>Aluminium</td>
</tr>
<tr>
<td>Gasket material</td>
<td>NBR</td>
</tr>
</tbody>
</table>
Using the above boundary conditions the temperatures in different channels at different lengths are calculated.
In this paper CO\textsubscript{2} and R134a are used as working fluids.

**HEAT TRANSFER COEFFICIENT CALCULATIONS**

Reynolds number \( R_e = \frac{4* m}{\mu* p} \)

\( m \)-mass flow rate of refrigerant=0.2 kg/s
\( p \)-perimeter of plate=0.62

By substituting the values in the Reynolds number equation
\( R_e = \frac{4*0.2}{(1.08 \times 10^{-5} \times 0.62)} \)
\( R_e = 119474.313 \)

Substitute the \( R_e \) in the nusselt number equation.

Nusselt number \( \text{Nu} = 0.023 * R_e^{0.8} * Pr^{n} \)

\( Pr = \frac{\mu*c_p}{k} = 0.7215 \)
\( n = 0.4 \) for heating
\( n = 0.3 \) for cooling

By substituting the Reynolds and prandtl number Nusselt number derived
\( \text{Nu} = 240.44 \)
\( h_1 = 195.01275 \text{ w/m}^2\text{k.} \)

Using the same procedure the heat transfer coefficient is found

For CO\textsubscript{2} at 273k and 1atm
\( R_e = \frac{4* m}{\mu* p} \)

\( m \)-mass flow rate of coolant=0.2 kg/s
\( p \)-perimeter of plate=0.62

\( R_e = \frac{4*0.2}{(1.657 \times 10^{-5} \times 0.62)} \)
\( R_e = 77871 \)

Nusselt number \( \text{Nu} = 0.023 * R_e^n * Pr^{0.4} \)

\( Pr = 0.74 \)
\( \text{Nu} = 166.925 \)
\( h_2 = 233.89 \text{ w/m}^2\text{k.} \)

For calculating overall heat transfer coefficient \( U \)

\[ U = \frac{1}{(1/h_1)-(1/h_2)} \]

By substituting the values of \( h_1 \) and \( h_2 \)

\( U = 90 \text{ w/m}^2\text{k.} \)

From these calculations NTU and Effectiveness is found to be 3.4 and 0.42.

Using the \( h_1 \) and \( h_2 \) as input values for the simulation the temperature variations are obtained from one plate to another from the top to bottom. Temperature variations of the body to body interface is neglected as we are mainly concern of the inlet and outlet temperatures of the fluids passing.
RESULTS AND DISCUSSIONS

The physical properties of the fluids used are

<table>
<thead>
<tr>
<th>PROPERTIES OF THE FLUID</th>
<th>CO2 AT 0 Degrees</th>
<th>R134 a AT 15 Degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>THERMAL CONDUCTIVITY (w/mk)</td>
<td>0.013512</td>
<td>0.02326</td>
</tr>
<tr>
<td>SPECIFIC HEAT CAPACITY (J/Kg.K)</td>
<td>816</td>
<td>901</td>
</tr>
<tr>
<td>DYNAMIC VISCOSITY (Ns/m³)</td>
<td>16.57×10⁻⁶</td>
<td>1.802×10⁻⁶</td>
</tr>
</tbody>
</table>

CONVERGENCE CRITERIA

GRID INDEPENDENCE is achieved in all the simulations

Contours Obtained for the Last plate are:

![Fig Temperature contour on top surface](image1)

![Fig Pressure contour of first channel](image2)
From the above figure it is observed that Hot fluid temperature is decreased and cold fluid temperature is increased.

From the above figure it is observed that there is a slightly pressure drop at the entry of the inlet of plate remaining area it is very low.

To achieve very slight variations in the inlet and outlet temperatures of the both fluids mainly to cool the R134a we used eight plates in this design and it is observed that the temperature changes are very high in the starting plates and goes on decreasing to the ending plates of the heat exchanger as the temperature differences decreases.

CONCLUSION

The plate type heat exchanger is modelled in solid works and the fluid flow analysis is done on the modelled fluid part. The analysis stated that when the thickness of the plates decreases then the heat flow is higher and if the number of plates increases then the outlet temperature difference of the fluids increased and the pressure contour stated that, there is little pressure drop in the entry and outlet of the fluid, From the turbulent contour it is interfered that there is very high turbulence in the entry and outlets due to sudden change in cross section along the plates. In future analysis, the heat exchanger will be modelled and the temperature difference will be optimised by variations in the design. We will try to employ this PHE as intercooler (which is the main aim) in multistage refrigeration system which attains low work input, better C.O.P, and efficiency.
REFERENCES


