Oil Cooler Heat Exchanger

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Oil Cooler Heat Exchanger

Double-Pipe Heat Exchanger Design

Kailash Kumar Jain Munoth
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Nomenclature

\( A \) – heat transfer surface area, \( m^2 \)

\( A_c \) – net freeflow area, \( m^2 \)

\( D_e \) – equivalent diameter for heat transfer, \( m \)

\( D_h \) – hydraulic diameter for pressure drop, \( m \)

\( d \) – diameter of inner tube, \( m \)

\( D \) – diameter of the annulus, \( m \)

\( f \) – fanning friction factor

\( h \) – heat transfer coefficient, \( \frac{W}{m^2 \cdot K} \)

\( H_f \) – height of the fin, \( m \)

\( k \) – thermal conductivity, \( \frac{W}{m \cdot K} \)

\( L \) – nominal length of the exchanger section, also length of longitudinal fin, \( m \)

\( m \) – mass flow rate, \( \frac{kg}{m \cdot s} \)

\( N_f \) – number of fins per tube

\( N_{hp} \) – number of hairpins

\( N_t \) – number of tubes

\( Re \) – reynolds number

\( Q \) – heat load or duty, \( W \)

\( T \) – absolute temperature, \( K \)

\( \Delta T \) – temperature difference, \( K \)

\( U_o \) – overall heat transfer coefficient, \( \frac{W}{m^2 \cdot K} \)

\( u_m \) – mean velocity, \( m/s \)

\( \text{Pr} \) – Prandtl Number

\( \Delta p \) – pressure drop, \( Pa \)

\( P \) – perimeter, \( m \)

\( Nu \) – nusselt number
Greek Symbols

\( \eta_f \) – fin efficiency
\( \eta_o \) – overall fin efficiency
\( \eta_p \) – pump efficiency
\( \mu \) – dynamic viscosity,
\( \rho \) – mass density, \( \frac{Kg}{m^3} \)
\( \delta \) – fin thickness, m

Subscripts

\( b \) – bulk
\( c \) – cross section, cold
\( e \) – equivalent
\( f \) – fins, fin – side, frictional
\( h \) – hot, heat transfer
\( h \) – hydraulic
\( hp \) – hairpins
\( i \) – inside of tubes
\( o \) – outside of bare tubes
\( t \) – total
\( u \) – unfinned
\( w \) – wall, wetted
Summary

Heat exchangers are one of the most important devices in cooling and heating process in factories, buildings, transports and others. The heat exchanger is also used to support cooling processes in power plants. For this project, the oil cooler is designed of double pipe and is constructed to cool hot oil using sea water.

In this report, the best design for a double pipe heat exchangers is chosen based on many factors such as size, cleaning friendly, heat transfer coefficient, economy and so on. For this project, hot oil is coming into the heat exchanger and water is used to cool the oil and as a result heat is exchanged. The inlet and outlet temperatures of both fluids are specified. In this project, the properties of materials are given and therefore there is no needed to experiment with different types of materials. Also, the proper fouling factors are selected to account for the fluid properties. The initial size assumption of the heat exchanger is given but one can start with one arbitrary inner tube and then check geometrical parameters to determine the effects of these changes as parametric studies are done in order to find the best design. Cost analysis of the selected design is also done for better decision making. After choosing the best design, the heat exchanges should be ready to fabricate. In this project, the geometric properties are taken as variables to come up with a best design that produces a high heat transfer coefficient.

Introduction

Background Information

Temperature can be defined as degree of hotness or coldness of an object. Heat exchangers are used to transfer that heat energy from one substance to another. In process units it is necessary to control the temperature of incoming and outgoing streams. These streams can either be gases or liquids. Heat exchangers raise or lower the temperature of these streams by transferring heat to (or) from the stream.

Heat exchangers are devices that exchange the heat between two fluids of different temperatures that are separated by a solid wall. The temperature gradient, or the differences in temperature facilitate this transfer of heat. Transfer of heat happens by three principle means: conduction, convection and radiation. In the use of heat exchangers radiation does take place. However, in comparison to conduction and convection, radiation does not play a major role. Conduction occurs as
the heat from the higher temperature fluid passes through the solid wall. To maximize the heat transfer, the wall should be thin and made of a very conductive material.

In a heat exchanger forced convection allows for the transfer of heat of one moving stream to another moving stream. With convection as heat is transferred through the pipe wall it is mixed into the stream and the flow of the stream removes the transferred heat. This maintains a temperature gradient between the two fluids.

**Figure 1: Double Pipe Heat Exchanger**

**Problem Statement**

A double pipe heat exchanger is used in the industry as a condenser for chemical processes and cooling fluid processes. A double pipe heat exchanger is designed in a decently large size for large applications in the industry. For this project, the oil cooler is a double pipe heat exchanger that needs to be designed with sea water. The decision was made to use a hairpin heat exchanger for the oil cooler. The mass flow rate, inlet and outlet temperatures of fluids, fluid properties, length of heat exchanger, and material have already been decided for this project. Freedom is still available in the selection of proper fouling factors. The geometrical information that is provided is to initiate the analysis. The geometrical properties will be taken as variable parameters to come up with a suitable design. We start with one inner tube and completing the hand calculation, we study the variation of critical qualitative parameters by changing geometrical properties. Cost analysis of the selected design has to be made for the best design chosen. The report will contain material selection, mechanical design parameters and cost analysis.
Assumptions

- The Inner and Outer tube have constant fouling resistance.
- Sea water fouling factor will be approximated by the city water fouling coefficient.
- Sea water fluid properties will be approximated from the saturated water table.
- Heat transfer coefficient is calculated based on the outside surface area.
- Cost is not an important factor during design stages.
- Corrosion does not have a huge impact on heat transfer coefficient.

Design Methodology

Basic Formulas

Heat Duty:

\[ Q = (m_C)_c \Delta T_c = (m_C)_h \Delta T_h \]

Inner Tube – Sea Water:

\[ u_m = \frac{m_c}{\rho_c \pi \left( \frac{d_i^2}{4} \right)} \]
\[ Re = \frac{\rho_c u_m d_i}{\mu} \]
\[ f = (1.58 \ln Re - 3.28)^{-2} \]
\[ Nu_b = \frac{(f/2)(Re_b)(Pr_b)}{1 + 8.7 \left( \frac{f}{2} \right)^2 (Pr_b - 1)} \]
\[ h_i = \frac{Nu_b \cdot k}{d_i} \]

Annulus – Oil:

\[ u_m = \frac{\dot{m}_h}{\rho_h A_c} \text{ where } A_c = \frac{\pi}{4} (D_i^2 - d_o^2) \]
\[ Re = \frac{\rho_h u_m D_h}{\mu} \text{ where } D_h = \frac{4A_c}{P_w} = D_i - d_o \]
\[ Nu_b = \frac{\left(\frac{f}{2}\right)(Re_b)(Pr_b)}{1 + 8.7\left(\frac{f}{2}\right)^2(Pr_b - 1)} \]

\[ h_o = \frac{Nu_b \times k}{D_e} \text{ where } D_e = \frac{D_i^2 - d_o^2}{d_o} \]

Overall heat transfer coefficient (with fouling):
\[ U_f = \frac{d_o}{d_i h_i} + \frac{d_o}{d_i} + \frac{d_o \ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o} \]

Overall heat transfer coefficient (without fouling):
\[ U_c = \frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o} \]

Cleanliness factor:
\[ CF = \frac{U_f}{U_c} \]

Total heat transfer area:
\[ A_o = \frac{Q}{U_o \Delta T_m} \]

Total heat transfer area (without fouling):
\[ A_{oc} = \frac{Q}{U_c \Delta T_m} \]

Total heat transfer area (with fouling):
\[ A_{of} = \frac{Q}{U_f \Delta T_m} \]

Number of hairpins:
\[ N_{hp} = \frac{A_o}{A_{hp}} \text{ where } A_{hp} = 2\pi d_o L \]

Percentage Over Surface:
\[ OS = 100 \times U_c \times \frac{1 - CF}{U_c \times CF} \]

Pressure Drop:

**Inner Tube**

\[ \Delta p_t = 4f \frac{2L}{d_i^2} \rho \frac{u_m^2}{2} N_{hp} \]

**Annulus**

\[ \Delta p_a = 4f \frac{2L}{D_i^2} \rho \frac{u_m^2}{2} N_{hp} \]

Pumping Power:

**Inner Tube**

\[ p_t = \frac{\Delta p_t m_c}{\eta_p \rho_c} \]

**Annulus**

\[ p_a = \frac{\Delta p_a m_h}{\eta_p \rho_h} \]

**Design Calculations**

**Typical Design**

<table>
<thead>
<tr>
<th>Fluid</th>
<th>SAE-30 Oil</th>
<th>Sea Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Temperature, C</td>
<td>65</td>
<td>20</td>
</tr>
<tr>
<td>Outlet Temperature, C</td>
<td>55</td>
<td>30</td>
</tr>
<tr>
<td>Pressure drop limitations, kPa</td>
<td>140</td>
<td>5</td>
</tr>
<tr>
<td>Total mass flow rate, kg/s</td>
<td>2.5</td>
<td>1.2</td>
</tr>
<tr>
<td>Density, kg/m³</td>
<td>912</td>
<td>1013.4</td>
</tr>
<tr>
<td>Specific Heat, KJ/kg-k</td>
<td>1.901</td>
<td>4.004</td>
</tr>
<tr>
<td>Viscosity, kg/m-s</td>
<td>0.075</td>
<td>9.64 \times 10^{-4}</td>
</tr>
<tr>
<td>Prandtl Number</td>
<td>1050</td>
<td>6.29</td>
</tr>
<tr>
<td>Thermal Conductivity, W/m-K</td>
<td>0.1442</td>
<td>0.639</td>
</tr>
</tbody>
</table>

\[ d_i = 2.067 \text{ in}, d_o = 2.375 \text{ in}, D_i = 3.548 \text{ in}, L = 3 \text{ m}, \]
Inner Tube – Sea Water:

\[ u_m = \frac{\dot{m}_c}{\rho_c \pi \frac{d_i^2}{4}} = 0.547 \text{ m/s} \]

\[ Re = \frac{\rho_c u_m d_i}{\mu} = 30188 \]

\[ f = (1.58 \ln Re - 3.28)^{-2} = 0.006 \]

\[ Nu_b = \frac{f}{(2)(Re_b)(Pr_b)} = 160.074 \]

\[ h_i = \frac{Nu_b \ast k}{d_i} = 1948.259 \frac{W}{m^2 \cdot K} \]

Annulus – Oil:

\[ u_m = \frac{\dot{m}_h}{\rho_h A_c} = 0.779 \frac{m}{s} \text{ where } A_c = \frac{\pi}{4} (D_i^2 - d_o^2) = 0.004 \text{ m}^2 \]

\[ Re = \frac{\rho_h u_m D_h}{\mu} = 282.107 \text{ where } D_h = \frac{4A_c}{P_w} = D_i - d_o = 0.030 \text{ m} \]

\[ Nu_b = 1.61 \left( Re \ast Pr \ast \frac{D_h^{1/3}}{L} \right) = 23.069 \]

\[ h_o = \frac{Nu_b \ast k}{D_e} = 44.770 \frac{W}{m^2 \cdot K} \text{ where } D_e = \frac{D_i^2 - d_o^2}{d_o} = 0.074 \text{ m} \]

Overall Heat Transfer Coefficient (with Fouling):

\[ U_f = \frac{d_o}{d_i h_i} + \frac{d_o R_f i}{d_i} + \frac{d_o \ln (\frac{d_o}{d_i})}{2k} + R_f o + \frac{1}{h_o} \left( \frac{1}{h_o} \right)^{-1/2} = 41.333 \frac{W}{m^2 \cdot K} \]

Overall Heat Transfer Coefficient (without Fouling):

\[ U_c = \frac{d_o}{d_i h_i} + \frac{d_o \ln (\frac{d_o}{d_i})}{2k} + \frac{1}{h_o} \left( \frac{1}{h_o} \right)^{-1/2} = 43.465 \frac{W}{m^2 \cdot K} \]
Cleanliness Factor:

\[ CF = \frac{U_f}{U_c} = 0.951 \]

Total Heat Transfer Area:

\[ A_o = \frac{Q}{U_o \Delta T_m} = 114.982 \, m^2 \]

Number of Hairpins:

\[ N_{hp} = \frac{A_o}{A_{hp}} = 101 \text{ where } A_{hp} = 2\pi d_o L = 1.137 \, m^2 \]

Percentage Over-Surface:

\[ OS = 100 \times U_c \times \frac{1 - CF}{U_c \times CF} = 5.160\% \]

Pressure Drop:

Inner Tube

\[ \Delta p_t = 4f \frac{2L}{d_i} \rho \frac{u_m^2}{2} N_{hp} = 41347.662 \, Pa \]

Annulus

\[ \Delta p_a = 4f \frac{2L}{D_h} \rho \frac{u_m^2}{2} N_{hp} = 709270.196 \, Pa \]

Pumping Power:

Inner Tube

\[ P_t = \frac{\Delta p_t m_c}{\eta_p \rho_c} = 61.201 \, W \]

Annulus

\[ P_a = \frac{\Delta p_a m_h}{\eta_p \rho_h} = 2430.339 \, W \]
Best Design

\[ d_i = 4.026 \text{ in}, d_o = 4.500 \text{ in}, D_t = 6.065 \text{ in}, L = 2m \]

**Inner Tube – SAE-30 Oil:**

\[ u_m = \frac{\dot{m}_c}{\rho_c \pi \frac{d_i^2}{4}} = 0.334 \text{ m/s} \]

\[ Re = \frac{\rho_c u_m d_i}{\mu} = 415.032 \]

\[ f = (1.58 \ln Re - 3.28)^{-2} = 0.026 \]

\[ Nu_b = \frac{\left(\frac{f}{2}\right)(Re_b)(Pr_b)}{1 + 8.7 \left(\frac{f}{2}\right)^{1/2} (Pr_b - 1)} = 33.064 \]

\[ h_i = \frac{Nu_b * k}{d_i} = 46.624 \frac{W}{m^2 * K} \]

**Annulus – Sea Water:**

\[ u_m = \frac{\dot{m}_h}{\rho_h A_c} = 0.141 \frac{m}{s} \text{ where } A_c = \frac{\pi}{4} (D_i^2 - d_o^2) = 0.008 m^2 \]

\[ Re = \frac{\rho_h u_m D_h}{\mu} = 5906.2 \text{ where } D_h = \frac{4A_c}{P_w} = D_i - d_o = 0.040 m \]

\[ Nu_b = \frac{\left(\frac{f}{2}\right)(Re_b)(Pr_b)}{1 + 8.7 \left(\frac{f}{2}\right)^{1/2} (Pr_b - 1)} = 41.392 \]

\[ h_o = \frac{Nu_b * k}{D_e} = 283.407 \frac{W}{m^2 * K} \text{ where } D_e = \frac{D_i^2 - d_o^2}{d_o} = 0.093 m \]

**Overall heat transfer coefficient (with fouling):**

\[ U_f = \frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k} + R_{fo} + \frac{1}{h_o} \approx 34.74 \frac{W}{m^2 * K} \]
Overall heat transfer coefficient (without fouling):

\[ U_c = \frac{d_o}{d_i h_i} + \frac{d_o \ln\left(\frac{d_o}{d_i}\right)}{2k} + \frac{1}{h_o} \approx 36.2 \frac{W}{m^2 K} \]

Cleanliness factor:

\[ CF = \frac{U_f}{U_c} = 0.96 \]

Total heat transfer area:

\[ A_o = \frac{Q}{U_o \Delta T_m} = 136.79 m^2 \]

Number of hairpins:

\[ N_{hp} = \frac{A_o}{A_{hp}} = 95 \text{ where } A_{hp} = 2\pi d_o L = 1.436 m^2 \]

Percentage Over Surface:

\[ OS = 100 \times U_c \times \frac{1 - CF}{U_c \times CF} = 4.198\% \]

Pressure Drop:

**Inner Tube**

\[ \Delta p_t = 4f \frac{2L}{d_i} \rho \frac{u_m^2}{2} N_{hp} = 19410.2 Pa \]

**Annulus**

\[ \Delta p_a = 4f \frac{2L}{D_h} \rho \frac{u_m^2}{2} N_{hp} = 3559.7 Pa \]

Pumping Power:

**Inner Tube**

\[ P_t = \frac{\Delta p_t m_c}{\eta_p \rho c} = 66.51 W \]
Annulus

\[ P_a = \frac{\Delta p_a m_h}{\eta_p \rho_h} = 5.269 \text{ W} \]

Conclusion and Recommendations

The parametric analysis yielded many different designs for us to choose from; we had approximately around 200 iterations which gave us 200 different designs for us to choose from. Stage 1 of the parametric study involved changing the geometric properties of the inner and outer pipe. We also changed the length of the heat exchanger with varying geometric properties to come up with 100 different designs. Stage 2 of the parametric study we decided to switch the liquid in the annulus into the inner tube and switch the liquid in the inner tube into the annulus. We ran this design for all the varying geometric properties from stage 1 to give us another 100 extra designs to choose from. Detailed stage 1 & 2 parametric study calculations can be found in the appendix. From Stage 1 parametric study results we were able to discard many designs due to pressure drop limitations and were left with a select few that can be seen in the parametric studies section above. From these selected few we were able to select a design that best encapsulates all the considerations given by the design statement.

From the parametric analysis we recommend choosing the design with an outer diameter of 6” and an inner diameter of 4” with the heat exchanger having a length of 2 meters. We choose this design as the best design for multiple reasons. The cross sectional area for this design is relatively smaller than for the other designs we compared but this also allows us to no dissipate too much heat through conduction of the material. The sea water will be place inside the annulus and this gave us a Reynolds number of 415.03 which is highly laminar with a friction factor of 0.026. The heat transfer coefficient on the inner tube was 46.624 W/m^2*K which is fairly higher than most of the other designs. The SAE-30 oil is placed inside the inner tube and we calculated a Reynolds number of 5906.2 which is turbulent flow. The heat transfer coefficient in the annulus is 283.4 W/m^2*K which is very high compared to the designs compared in the parametric study. We have found that this design has a clean overall heat transfer coefficient of 36 W/m^2*K and fouled heat transfer coefficient of 34.7 W/m^2*K with a cleaning factor of 0.0.96. This cleaning factor is very high so we don’t have to maintain a regular cleaning schedule to clean the heat exchanger. While maintain high levels of cleanliness factors we can assume that the overall heat transfer coefficient will be closer to the clean side than the fouled giving the heat exchanger a very good heat transfer coefficient. The pressure drop from the tube side and the annulus are very similar to the other designs and so there really is not a big number to deal with. Lastly, the pumping power is small enough to power required to pump the liquids and we are not consuming a lot of power to make it affordable. We have chosen to recommend this design because it has a good heat transfer coefficient for the pumping power that is required.
Appendix

Parametric Studies

Parametric Studies (Switching SAE-30 Oil to Inner Tube and Sea Water to Annulus)

MATLAB PROGRAM :

di=0.021;do=0.027;Di=0.041;thermalconductivity_pipe=52;MassFlowCOLD_FLUID=1.2 ;MassFlowHOT_FLUID=2.5;Efficiency_Pump=.8;L=2;

temperature_HOT_FLUIDi = 65;
temperature_HOT_FLUIDo = 55;
temperature_COLD_FLUIDi = 20;
temperature_COLD_FLUIDo = 30;

density_HOT_FLUID = 912;density_COLD_FLUID=1013.4;
shHOT_FLUID = 1901;shCOLD_FLUID=4004;
Prandtl_HOT_FLUID =1050;Prandtl_COLD_FLUID=6.29;
tcHOT_FLUID=0.1442;tcCOLD_FLUID=0.639;
R_fo = 0.000176;
R_fi=0.000088;
Mass_Flow_Rate_HOT_FLUID = 2.5;
Mass_Flow_Rate_COLD_FLUID = 1.2;
viscosity_COLD_FLUID = 0.964*10^-3;
viscosity_HOT_FLUID= 0.075;
Q = 47525;
Tm = 10;

UmC = Mass_Flow_Rate_COLD_FLUID/(density_COLD_FLUID*(pi/4)*(Di^2-do^2))
HydDia=pi*(Di^2-do^2)/(pi*(Di+do))
ReC = density_COLD_FLUID*UmC*HydDia/(viscosity_COLD_FLUID)
de=(Di^2-do^2)/do
Heat_Transfer_Coefficient_Cold_Fluid=0;

if ReC > 2300
    fC = double(((1.58*log(ReC))-3.28)^(-2))
    NubCOLD_FLUID =
    ((fC/2)*ReC*Prandtl_COLD_FLUID)/((1+8.7*(Prandtl_COLD_FLUID-1)*(fC/2)^0.5))
    Heat_Transfer_Coefficient_Cold_Fluid = NubCOLD_FLUID*tcCOLD_FLUID/de
end

UmH = Mass_Flow_Rate_HOT_FLUID/((density_HOT_FLUID*(pi/4)*(di^2)))
ReH = density_HOT_FLUID*UmH*di/(viscosity_HOT_FLUID)
Heat_Transfer_Coefficient_Hot_Fluid=0;

if ReH > 2300
    fH = double(((1.58*log(ReH))-3.28)^(-2))
    NubHOT_FLUID =
    (fH/2)*ReH*Prandtl_HOT_FLUID/(1+8.7*(Prandtl_HOT_FLUID1)*(fH/2)^0.5)
Heat_Transfer_Coefficient_Hot_Fluid = NubHOT_FLUID*tcHOT_FLUID/di
end

if ReH < 2300
    fH = double((1.58*log(ReH)-3.28)^(-2))
    NubHOT_FLUID = 1.61*(ReH*Prandtl_HOT_FLUID*HydDia/L)^(1/3)
    Heat_Transfer_Coefficient_Hot_Fluid = NubHOT_FLUID*tcHOT_FLUID/di
end

OverallHTC_Fouling_Inv = do/(di*Heat_Transfer_Coefficient_Hot_Fluid) +
    do*R_fi/di + do*log(do/di)/(2*thermalconductivity_pipe) + R_fo +
    1/Heat_Transfer_Coefficient_Cold_Fluid

OverallHTC_Fouling = 1/OverallHTC_Fouling_Inv
OverallHTSArea = Q*OverallHTC_Fouling_Inv/Tm

HeatTranAreaForHairPin = 2*pi*do*L
NoOfHairPins = OverallHTSArea/HeatTranAreaForHairPin
NoOfHairPins = ceil(NoOfHairPins)

CleanOverallHTCof_inv = do/(di*Heat_Transfer_Coefficient_Hot_Fluid) +
    do*log(do/di)/(2*thermalconductivity_pipe) +
    1/(Heat_Transfer_Coefficient_Cold_Fluid)
CleanOverallHTCof = 1/CleanOverallHTCof_inv

CleanlinessFactor = OverallHTC_Fouling/CleanOverallHTCof

OverSurface = 100 * (1-CleanlinessFactor)/CleanlinessFactor
PressureDropCOLD_FLUID = 4*fC*2*L*NoOfHairPins*density_HOT_FLUID*UmC^2/(2*di)
PressureDropHOT_FLUID = 4*fH*2*L*NoOfHairPins*density_COLD_FLUID*UmH^2/(2*HydDia)
PumpingPowerCOLD_FLUID = PressureDropCOLD_FLUID *
    MassFlowCOLD_FLUID/(Efficiency_Pump*density_COLD_FLUID)
PumpingPowerHOT_FLUID = PressureDropHOT_FLUID *
    MassFlowHOT_FLUID/(Efficiency_Pump*density_HOT_FLUID)

if PressureDropCOLD_FLUID > 5000 || PressureDropHOT_FLUID > 140000
    fprintf('Pressure drop exceeded')
    return;
end

Parametric Studies (Sea Water-Inner Tube and SAE 30-Oil Outer Tube Changing Geometric Properties: Di, do, di, and L)

[Excel Sheet Calculations are attached]
### TABLE 9.2
Heat Exchanger and Condenser Tube Data

<table>
<thead>
<tr>
<th>Nominal Pipe Size (in.)</th>
<th>Outside Diameter (in.)</th>
<th>Schedule Number or Weight</th>
<th>Wall Thickness (in.)</th>
<th>Inside Diameter (in.)</th>
<th>Surface Area (ft²/ft₄)</th>
<th>Inside Area (ft²/ft₄)</th>
<th>Metal Area (in²)</th>
<th>Flow Area (in²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/4</td>
<td>1.05</td>
<td>40</td>
<td>0.113</td>
<td>0.824</td>
<td>0.275</td>
<td>0.216</td>
<td>0.333</td>
<td>0.533</td>
</tr>
<tr>
<td></td>
<td></td>
<td>80</td>
<td>0.154</td>
<td>0.742</td>
<td>0.275</td>
<td>0.194</td>
<td>0.434</td>
<td>0.432</td>
</tr>
<tr>
<td>1</td>
<td>1.315</td>
<td>40</td>
<td>0.133</td>
<td>1.049</td>
<td>0.344</td>
<td>0.275</td>
<td>0.494</td>
<td>0.864</td>
</tr>
<tr>
<td></td>
<td></td>
<td>80</td>
<td>0.179</td>
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