Group Assignment on

Double Pipe Heat Exchanger

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CHE312A: Heat Transfer and Its Applications

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Department of Chemical Engineering, IIT Kanpur

Group Number:**13**

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Declaration:

- This report is a collective effort, with equal contribution from all members listed above
- The report is primarily based on our original work. We have provided due credit by citing the source whenever applicable.

Heat Exchangers

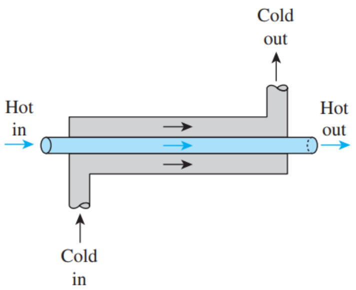
Heat exchangers are devices that facilitate the **exchange of heat** between two fluids at different temperatures while keeping them from mixing. Heat exchangers are commonly used in practice in a wide range of applications, from heating and air-conditioning systems in a household to chemical processing and power production in large plants. Heat exchangers differ from mixing chambers because they do not allow the two fluids to mix.

Types of Heat Exchangers

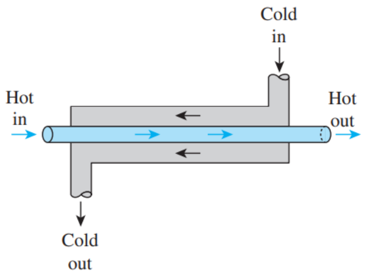
•Double Pipe Heat Exchanger
•Shell and Tube Heat Exchanger
•Plate Heat Exchanger
•Tube in Tube Exchanger

Double Pipe Heat Exchangers

Double-pipe heat exchangers transfer thermal energy between two fluids at different temperatures without mixing them. The primary use of these heat exchangers is the sensible heating or cooling process of fluids where small heat transfer areas are required.
Two types of flow arrangement are possible in a double-pipe heat exchanger: in **parallel flow**, the hot and cold fluids enter the heat exchanger at the same end and move in the same direction. In **counter flow**, on the other hand, the hot and cold fluids enter the heat exchanger at opposite ends and flow in opposite directions.



Parallel flow



Counter Flow

Applications

Double pipe heat exchangers are used in many industrial processes, cooling technology, refrigeration devices, sustainable energy applications, and another field.

Problem Statement

Consider a double pipe heat exchanger used to heat a solution from **25**℃ to **50**℃ at a rate of **300 LPH** using hot water at **75**℃ and **300 LPH.** A **1 m** long, **3 cm** in diameter, and **2 mm** thick copper pipe is used as the inner pipe. The initial temperature of the heat exchanger is **25**℃. Assume that the properties of the solution are reasonably approximated using the properties of water and that the heat exchanger is perfectly insulated. Estimate the **time taken** for the heat exchanger to reach a nearly **steady state**.

Approach:

•We start with defining the steady state for our problem: we define it as the state at which the temperature of the heat exchanger will remain constant with respect to time. All the flux incoming to the heat exchanger will be outputted to the outer fluid.

•Now, assuming the above steady-state condition, we can calculate the outlet temperature of the fluid inside the tube (heat exchanger). This came out to be **50**℃**.**

$$\mathbf{T_o = 50^\circ C}$$

•To get an approximate convective heat transfer coefficient, we first calculated the Reynolds number for the inner tube.

We took the properties of water at its average bulk temperature of **62.5**℃.

$$\mathbf{T_b = (75+50)/2 = 62.5^\circ C}$$

TABLE A.6 Thermophysical Properties of Saturated Water[a]

Temperature, T (K)	Pressure, p (bars)[b]	Specific Volume (m³/kg) $v_f \cdot 10^3$	v_g	Heat of Vaporization, h_{fg} (kJ/kg)	Specific Heat (kJ/kg · K) $c_{p,f}$	$c_{p,g}$	Viscosity (N · s/m²) $\mu_f \cdot 10^6$	$\mu_g \cdot 10^6$	Thermal Conductivity (W/m · K) $k_f \cdot 10^3$	$k_g \cdot 10^3$	Prandtl Number Pr_f	Pr_g	Surface Tension, $\sigma_f \cdot 10^3$ (N/m)	Expansion Coefficient, $\beta_f \cdot 10^6$ (K⁻¹)	Temperature, T (K)
273.15	0.00611	1.000	206.3	2502	4.217	1.854	1750	8.02	569	18.2	12.99	0.815	75.5	−68.05	273.15
275	0.00697	1.000	181.7	2497	4.211	1.855	1652	8.09	574	18.3	12.22	0.817	75.3	−32.74	275
280	0.00990	1.000	130.4	2485	4.198	1.858	1422	8.29	582	18.6	10.26	0.825	74.8	46.04	280
285	0.01387	1.000	99.4	2473	4.189	1.861	1225	8.49	590	18.9	8.81	0.833	74.3	114.1	285
290	0.01917	1.001	69.7	2461	4.184	1.864	1080	8.69	598	19.3	7.56	0.841	73.7	174.0	290
295	0.02617	1.002	51.94	2449	4.181	1.868	959	8.89	606	19.5	6.62	0.849	72.7	227.5	295
300	0.03531	1.003	39.13	2438	4.179	1.872	855	9.09	613	19.6	5.83	0.857	71.7	276.1	300
305	0.04712	1.005	29.74	2426	4.178	1.877	769	9.29	620	20.1	5.20	0.865	70.9	320.6	305
310	0.06221	1.007	22.93	2414	4.178	1.882	695	9.49	628	20.4	4.62	0.873	70.0	361.9	310
315	0.08132	1.009	17.82	2402	4.179	1.888	631	9.69	634	20.7	4.16	0.883	69.2	400.4	315
320	0.1053	1.011	13.98	2390	4.180	1.895	577	9.89	640	21.0	3.77	0.894	68.3	436.7	320
325	0.1351	1.013	11.06	2378	4.182	1.903	528	10.09	645	21.3	3.42	0.901	67.5	471.2	325
330	0.1719	1.016	8.82	2366	4.184	1.911	489	10.29	650	21.7	3.15	0.908	66.6	504.0	330
335	0.2167	1.018	7.09	2354	4.186	1.920	453	10.49	656	22.0	2.88	0.916	65.8	535.5	335
340	0.2713	1.021	5.74	2342	4.188	1.930	420	10.69	660	22.3	2.66	0.925	64.9	566.0	340
345	0.3372	1.024	4.683	2329	4.191	1.941	389	10.89	664	22.6	2.45	0.933	64.1	595.4	345
350	0.4163	1.027	3.846	2317	4.195	1.954	365	11.09	668	23.0	2.29	0.942	63.2	624.2	350
355	0.5100	1.030	3.180	2304	4.199	1.968	343	11.29	671	23.3	2.14	0.951	62.3	652.3	355
360	0.6209	1.034	2.645	2291	4.203	1.983	324	11.49	674	23.7	2.02	0.960	61.4	697.9	360

- At **335K(82℃)** , the Reynolds number of fluid comes out to be **7751.41.**

R_e~7751.41 <**10^4** laminar flow

- Since the Reynolds number comes under the region of laminar flow, We will use the formula of **Average Nusselt number** for the duct flow, which is given by the expression:

For *laminar, fully developed conditions* with a *constant surface temperature,* the assumption of negligible axial conduction is often reasonable. Substituting for the velocity profile from Equation 8.15 and for the axial temperature gradient from Equation 8.33, the energy equation becomes

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) = \frac{2u_m}{\alpha}\left(\frac{dT_m}{dx}\right)\left[1-\left(\frac{r}{r_o}\right)^2\right]\frac{T_s - T}{T_s - T_m} \qquad T_s = \text{constant} \qquad (8.54)$$

A solution to this equation may be obtained by an iterative procedure, which involves making successive approximations to the temperature profile. The resulting profile is not described by a simple algebraic expression, but the resulting Nusselt number may be shown to be [2]

$$Nu_D = 3.66 \qquad T_s = \text{constant} \qquad (8.55)$$

Note that in using Equation 8.53 or 8.55 to determine h, the thermal conductivity should be evaluated at T_m.[1]

- Once we get a nusselt number, we will equate it with **(hD/k)** to get a convective heat transfer coefficient, which comes out to be 80.03.

N_u=hD/k=80.03

- Due to the unavailability of outer diameter, we couldn't precisely calculate the convective heat transfer for the outer fluid. However, using our steady-state assumption, we assumed it to be the same as the inner fluid's **convective heat transfer coefficient.**

•As we know, the width of the copper tube is significantly less than the length. We can use plane wall approximation to calculate the change in its internal temperature with respect to time

•We will first calculate the Biot number for our arrangement, which comes out to be **0.0008**. Which is less than **0.1,** so we can employ a lumped capacitance method in our problem with

Copper					
Pure	1358	8933	385	401	117

minor modification. Properties of copper at **300 K:**

•The final expression for our problem will look something like this, plus an additional convection term, as convection is happening from both sides.

$$-\dot{E}_{\text{out}} = \dot{E}_{\text{st}} \tag{5.1}$$

or

$$-hA_s(T - T_\infty) = \rho V c \frac{dT}{dt} \tag{5.2}$$

•Now, we are looking for an average temperature that we can use to represent the out inner and out fluids temperature. For that, we used the expression derived in [1].

$$\begin{cases} \dfrac{dT_h^*}{dx^*} + N_h(T_w^* - T_h^*) = 0 \\ -\dfrac{dT_c^*}{dx^*} + N_c(T_w^* - T_c^*) = 0 \\ V^*C_h^*N_h(T_h^* - T_w^*) + C_c^*N_c(T_c^* - T_w^*) = 0 \end{cases}$$

$$T_h^*(x^*) = \frac{e^{\alpha x^*} - V^*C^*}{e^{\alpha} - V^*C^*}$$

$$T_c^*(x^*) = V^*C^*\left(\frac{e^{\alpha x^*} - 1}{e^{\alpha} - V^*C^*}\right)$$

$$V^* = \frac{V_h}{V_c} \quad \text{mean velocities ratio of fluids}$$

$$C^*_{h,c} = \frac{C_{h,c}}{C_w} \quad \text{heat capacity ratio of fluids and wall}$$

N_h and N_c are defined as

$$N_{h,c} = \frac{h_{h,c} A_{h,c}}{\dot{m}_{h,c} C_{p_{h,c}}} \tag{9}$$

$$\alpha = \frac{1 - V^* C^*}{(1/N_h) + (1/N_c) V^* C^*}$$

•After all the calculations, we got an average temperature of **60.72**℃ for hot fluid and **35.23**℃ for cold fluid.

•Once the expression for the copper tube is done, we need to integrate the temperature from the initial temperature of **298 K** to its steady-state temperature.

•Steady state temperature of the copper tube is approximated with the temperature of the fluid in its vicinity. Since the fluid on both sides has different temperatures, it will give rise to a temperature gradient.

•We need to estimate its average temperature. For that, we took the mean temperature of the fluid around its vicinity, which we calculated earlier.

•After solving the integration, we obtained a time of approximately **43.5 seconds**, which is satisfactory as the copper has very high thermal conductivity. It won't take much time to reach its steady state.

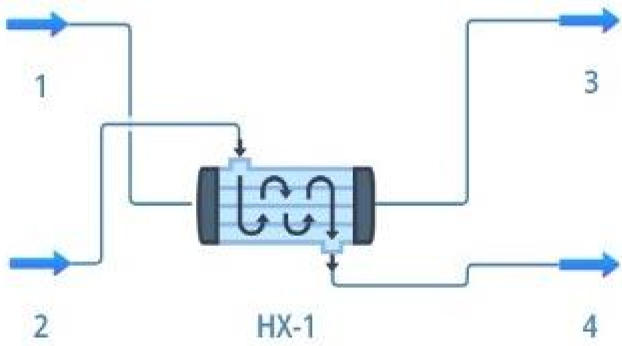
T_{req}=43.5 s

Conclusions:

Henceforth, we have analyzed our problem using several concepts like flow in a duct, lumped heat capacitance method, and calculated time required to reach a steady state. Since our problem involves a lot of approximations, like assuming the fluids inside and outside to have the

same convective heat transfer coefficient, the constant surface temperature throughout the copper tube.
We may have accumulated a few **error percentages** from the answer, which involves rigorous calculations and **numerical analysis.**

We tried to be as **accurate** as possible at each step and calculated the time required to reach a steady state with all our resources.
Additionally, we have simulated our problem in software like **DWSIM** to extract some useful information.



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[1] Moulay Abdelghani-Idrissi, Farid Bagui Countercurrent Double-Pipe Heat Exchanger Subjected to Flow-Rate Step Change, Part I: New Steady-State Formulation, 2002.
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