# 280.371 PROCESS ENGINEERING OPERATIONS

# Integrated Thermal and Hydraulic Design of Continuous Indirect Heat Exchangers

## Worked Examples and Tutorial Questions

**2023**

# WORKED EXAMPLES

The following 3 examples will be worked through during the lectures.

### *Q1a Double Pipe Heat Exchanger*

Product at a flow rate of 5000 kg/h will be heated from 20°C to 40°C by hot water at 140°C. A 15°C hot water temperature drop is targeted. A number of double pipe HE modules will be connected in series to provide fully countercurrent flow, hot water flowing in the inner tube. Each module is 2.5 m long, the inner pipe is 50 mm in diameter, the outer pipe is 75 mm and the wall is 1.5 mm thick stainless steel (k = 16.3 W $m^{-1}$ $K^{-1}$). Assume that the product has the same properties as water, that fouling is negligible and that HE is insulated against heat losses.

1. Calculate the number of modules required.
2. Calculate the pressure drops.
3. If a manual valve is inserted in the hot water supply to control the product outlet temperature, what will be the effect on the HW outlet temperature?

### *Q1b Double Pipe Heat Exchanger*

As in example 1a, a product at a flow rate of 5000 kg/h will be heated from 20°C to 40°C by hot water at 140°C. Now there is no restriction on the hot water temperature drop, but rather the pressure differential between the hot water supply and return mains is specified as 0.5 kPa.

- Will the heat exchanger from example 1a still be suitable for the application?

### *Q2 Shell and Tube Heat Exchanger Rating and Design Modification*

A 1 shell pass/2 tube passes shell and tube exchanger is to be used for cooling hot oil. The oil will be pumped through the shell side and the cooling medium, cold water, through the tubes. (In this case, the water is more likely than the oil to foul the heat transfer surface; it is therefore made to flow through the tubes, which are easier to clean than the shell side.)

The desired oil outlet temperature is 94°C, the expected fouling factor ($R$) is 0.00066 $m^2$ K $W^{-1}$, and the maximum allowable frictional pressure drops on the tube side and the shell side are respectively 30 kPa and 40 kPa. The baffle spacing can be altered if necessary.

1. Will the heat exchanger meet the required heat transfer duty without exceeding the allowable pressure drops?

2. Can it make full use of these pressure drops?

**Data available**

The following data (given, and immediately derivable) are available.

**Oil:** $T_{oil,\,i}$ = 118.5°C

Required $T_{oil,\,o}$ = 94°C

Oil physical properties at 106.25°C:

$\mu$ = $1.64 \times 10^{-3}$ Pa s
$\rho$ = 820 kg $m^{-3}$
$\lambda$ = 0.26 W $m^{-1}$ $K^{-1}$
c = 2650 J $kg^{-1}$ $K^{-1}$
Oil flow rate = 57,000 L $h^{-1}$

**Water:** $T_{w,\,i}$ = 30°C

Water flow rate = 78 000 L $h^{-1}$

Water physical properties at 34.7°C:

$\mu$ = $0.723 \times 10^{-3}$ Pa s
$\rho$ = 994 kg $m^{-3}$ (0.4% higher than assumed value)
$\lambda$ = 0.625 W $m^{-1}$ $K^{-1}$
c = 4180 J $kg^{-1}$ $K^{-1}$ (= assumed value)

**Heat exchanger specifications:**

| | |
|---|---|
| Tube length (= shell length) | = 3.6 m |
| Tube ID | = 0.0206 m |
| Tube OD | = 0.0254 m |
| Tube pitch (triangular) | = 0.0318 m |
| Total number of tubes | = 68 |
| Number of tube passes | = 2 |
| Number of baffles | = 8 |
| Shell diameter | = 0.337 m |
| Tube thermal conductivity | = 20 W $m^{-1}$ $K^{-1}$ |

## *Q3 Plate Heat Exchanger Sizing Procedure*

The required duty is to heat 22.7 kg $s^{-1}$ of water (the process or primary liquid) from 10°C to 88°C using 34 kg $s^{-1}$ of hot water (the service or secondary liquid) at an inlet temperature of 95°C. Flow is countercurrent. The allowable frictional pressure drops on the process and service sides are 100 kPa and 150 kPa respectively.

# TUTORIAL PROBLEMS

The following ten problems provide the student with representative questions and can be used for individual study or in a tutorial situation. Model answers for the questions are available.

1. A proposed counterflow double-pipe heat exchanger design has a stainless steel (k = 16.3 W $m^{-1}$ $K^{-1}$) inner tube with an outside diameter of 69 mm and an inside diameter of 60 mm. This is surrounded by a shell of 117 mm inside diameter. Ethylene glycol flowing in the tube at a rate of 2.0 kg/s is heated from 40°C to 60°C by hot water in the annulus, cooling from 95°C to 85°C. The tube side heat transfer coefficient is given as 650 W $m^{-2}$ $K^{-1}$. Assume fouling resistances are negligible.

    (a) What is the length of the heat exchanger required to achieve the given temperatures? ***Answer: 28.4 m.***

    (b) What will be the pressure drop on the annulus side? ***Answer: 830 Pa.***

    (c) If the pressure drop on the annulus side is 10% greater than that available, suggest a design modification that will bring it within bounds. No calculations are required but you need to justify your answer by discussing the effect of the change on all relevant parameters. ***Answer: Increase the shell diameter and marginally increase the length of the HE. This works because the pressure drop will be more sensitive than the annulus side HTC to this change. Also point out that most of the HT resistance is on the tube side anyway.***

    **Data**

    **Ethylene glycol**

    | | |
    |---|---|
    | density | 1087 kg $m^{-3}$ |
    | viscosity | 5.15 x $10^{-3}$ Pa s |
    | heat capacity | 2562 J $kg^{-1}$ $K^{-1}$ |
    | thermal conductivity | 0.259 W $m^{-1}$ $K^{-1}$ |

    **Water**

    | | |
    |---|---|
    | density | 967 kg $m^{-3}$ |
    | viscosity | 0.318 x $10^{-3}$ Pa s |
    | heat capacity | 4205 J $kg^{-1}$ $K^{-1}$ |
    | thermal conductivity | 0.675 W $m^{-1}$ $K^{-1}$ |

2. A liquid foodstuff is being heated in a horizontal shell and tube heat exchanger with one shell pass and two tube passes. The foodstuff flows in the tubes and the heating medium, hot water, flows in the shell. The inside and outside surfaces of the tubes are clean and free of scale.

The tube side heat transfer coefficient = 9000 W $m^{-2}$ $K^{-1}$.

The required overall heat transfer coefficient ($U_o$) based on the outside area of the tubes is 1300 W $m^2$ $K^{-1}$.

(a) Determine the hot water volumetric flow rate necessary to achieve the required overall heat transfer coefficient. ***Answer: 0.00217 $m^3$ $s^{-1}$.***

(b) Determine the shell side pressure loss of the heat exchanger. ***Answer: 0.647 kPa.***

**Data**

**Heat exchanger**

| | | |
|---|---|---|
| Length of tube bundle | = | 5.2 m |
| Shell diameter | = | 0.35 m |
| Baffle spacing | = | 0.5 m |
| Tube I.D. | = | 0.023 m |
| Tube O.D. | = | 0.027 m |
| Tube pitch (square) | = | 0.03 m |
| Thermal conductivity of tube metal | = | 16.3 W $m^{-1}$ $K^{-1}$ |

**Service liquid**

| | | |
|---|---|---|
| $\mu$ | = | 0.00047 Pa.s |
| $\mu_w$ | = | 0.00079 Pa.s |
| $\rho$ | = | 1000 kg $m^{-3}$ |
| $\lambda$ | = | 0.653 W $m^{-1}$ $K^{-1}$ |
| c | = | 4180 J $kg^{-1}$ $K^{-1}$ |

3. A horizontal shell and tube heat exchanger (with one shell pass and two tube passes of 30 tubes per pass) is to be used to cool a glucose syrup. The syrup flows through the tubes and the service liquid, cold water, flows through the shell.

The required overall heat transfer coefficient ($U_o$) based on the outside area of the tubes is 1350 W $m^{-2}$ $K^{-1}$. The tube-side heat transfer coefficient is fixed by process conditions at 8000 W $m^{-2}$ $K^{-1}$.

(a) Calculate the required cold-water flow rate in $m^3$ $s^{-1}$. ***Answer: 7.76 x $10^{-3}$$m^3$.$s^{-1}$***

(b) The pressure limits of the available pumps are 50 kPa for the cold water pump and 10 kPa for the glucose syrup pump. Determine in each case whether or not the pump will cope with the proposed duty. ***Answer: Shell side: 0.666 kPa; Tube side: 9.6 kPa.***

**Data**

**Heat exchanger**

| | | |
|---|---|---|
| Length of tube bundle | = | 4.8 m |
| Shell diameter | = | 0.25m |
| Baffle spacing | = | 0.6 m |
| Tube I.D. | = | 0.020 m |
| Tube O.D. | = | 0.024 m |
| Tube pitch (triangular) | = | 0.032 m |
| Thermal conductivity of tube metal | = | 18.0 W $m^{-1}$ $K^{-1}$ |

**Glucose syrup**

| | | |
|---|---|---|
| Throughput | = | $8.70 \times 10^{-3}$ $m^3$ $s^{-1}$ |
| Viscosity of bulk liquid | = | $8.60 \times 10^{-4}$ Pa.s |
| Viscosity of liquid at tube wall | = | $3.87 \times 10^{-3}$ Pa.s |
| Density | = | 1130 kg $m^{-3}$ |

**Service liquid**

| | | |
|---|---|---|
| Viscosity of bulk liquid | = | 0.0012 Pa.s |
| Viscosity of liquid at tube wall | = | 0.00033 Pa.s |
| Density | = | 1000 kg $m^{-3}$ |
| Thermal conductivity | = | 0.58 W $m^{-1}$ $K^{-1}$ |
| Specific heat | = | 4180 J $kg^{-1}$ $K^{-1}$ |

4. A shell and tube heat exchanger (with one shell pass and two tube passes) is to be used to heat a fruit juice to pasteurising temperature. The fruit juice will flow through the tubes, and the service liquid, hot water, will flow through the shell. The maximum available hot water flow rate is 46770 L $h^{-1}$.

Appropriate calculations have shown that at the proposed juice throughput the tube side film heat transfer coefficient will be 9000 W $m^{-2}$ $K^{-1}$, and that the overall heat transfer coefficient ($U_o$) based on the outside area of the tubes must be 2050 W $m^{-2}$ $K^{-1}$ if the heat exchanger is to fulfil the required thermal duty. Further data are given below.

(a) Determine quantitatively if the required overall heat transfer coefficient can be achieved. ***Answer: No.***

(b) If it cannot, how can the heat exchanger be modified to ensure that it will be achieved? A quantitative answer is required. ***Answer: See model answer.***

(c) What will be the pressure drop owing to viscous friction on the shell side of the modified exchanger? ***Answer: 3.0 kPa.***

**Data**

**Heat exchanger**

| | | |
|---|---|---|
| Length of tube bundle | = | 4.8 m |
| Shell diameter | = | 0.3 m |
| Baffle spacing | = | 0.6 m |
| Tube I.D. | = | 0.02 m |
| Tube O.D. | = | 0.023 m |
| Tube pitch (square) | = | 0.03 m |
| Thermal conductivity of tube metal | = | 16.8 W $m^{-1}$ $K^{-1}$ |

**Hot water properties**

| | | |
|---|---|---|
| Density | = | 950 kg $m^{-3}$ |
| Specific heat | = | 4230 J $kg^{-1}$ $K^{-1}$ |
| Thermal conductivity | = | 0.688 W $m^{-1}$ $K^{-1}$ |
| Bulk viscosity | = | $2.5 \times 10^{-4}$ Pa.s |
| Viscosity at tube surfaces | = | $3.6 \times 10^{-4}$ Pa.s |

5. A shell and tube heat exchanger (with one shell pass and two tube passes) is being designed to preheat a flavoured milk product (prior to the product being heated to sterilizing temperature by direct steam injection). The milk product will flow through the tubes, and hot water, the service liquid, through the shell. The hot water flow rate will be fixed at 79 000 L $h^{-1}$.

(a) A preliminary estimate of heat exchanger specifications, and hot water physical properties, are given below (see 'Further Data'). Rate this preliminary design by calculating:

- the shell side film heat transfer coefficient, ***Ans: 3041 W $m^{-2}$ $K^{-1}$***
- the overall heat transfer coefficient (given a tube side film heat transfer coefficient of 2000 W $m^{-2}$ $K^{-1}$), and ***Ans: 918 W $m^{-2}$ $K^{-1}$***
- the shell side frictional pressure drop. ***Ans: 1.7 kPa***

(b) Assuming the heat transfer characteristics as calculated in (i) are satisfactory, and assuming that the available pressure drop on the shell side is 3 kPa, should this preliminary design be accepted as satisfactory, or should some change be made to the provisional specifications, and further rating calculations be carried out to see if an improved overall performance can be achieved?

What change or changes would you suggest should be made, **and why**, and what new calculations should be performed? A **qualitative** answer only is required here, but it must be a carefully considered and complete one. ***Answer: See model answer.***

**Further Data**

**Preliminary estimate of heat exchanger specifications**

| | | |
|---|---|---|
| Length of tube bundle | = | 4.5 m |
| Shell diameter | = | 0.45 m |
| Baffle spacing | = | 0.5 m |
| Tube I.D. | = | 0.025 m |
| Tube O.D. | = | 0.03 m |
| Tube pitch (square) | = | 0.04 m |
| Thermal conductivity of tube metal | = | 17 W $m^{-1}$ $K^{-1}$ |

**Hot Water Physical Properties**

| | | |
|---|---|---|
| Density | = | 960 kg $m^{-3}$ |
| Specific heat | = | 4200 J $kg^{-1}$ $K^{-1}$ |
| Thermal conductivity | = | 0.678 W $m^{-1}$ $K^{-1}$ |
| Bulk viscosity | = | $2.8 \times 10^{-4}$ Pa.s |
| Viscosity at tube surface | = | $2.6 \times 10^{-4}$ Pa.s |

6. A shell and tube heat exchanger is being designed to indirectly heat a flow of potable water from 15 to 81°C using saturated steam at 140°C. The water will flow at 43 200 L h$^{-1}$ through the tube side of the exchanger, and the steam will be supplied to the shell side.

As the first step in the design procedure, you (the designer) decide to try (on paper) 30 tubes per pass with a tube length of 5.3 m.

(a) Using the data given above, and the further data given below, you must now complete the following tasks:

(i) Determine how many tube passes the heat exchanger must have if it is to meet the required heat transfer duty. Assume a total fouling resistance of 0.0002 m$^2$ K W$^{-1}$, and a film heat transfer coefficient on the steam side of 7000 W m$^{-2}$ K$^{-1}$. ***Answer: 2 tube passes.***

(ii) Rate the exchanger configuration determined in (a) (i) in terms of the expected pressure loss owing to viscous friction on the tube side. ***Answer: 10.2 kPa.***

(b) The available pressure drop for the tube side of the heat exchanger is 20 kPa. If the pressure loss calculated in (a) (ii) differs from this figure by more than 15%, suggest briefly, **with reasons, specific** modifications that might be made to the proposed exchanger configuration (prior to re-rating it) in an attempt to match actual tube side pressure loss to the available pressure drop, **while preserving the required heat transfer performance**. What would be the advantage of achieving a match **in this case**? ***Answer: See model answer.***

**Further Data**

**Heat exchanger tube specifications**

| | | |
|---|---|---|
| Inside diameter | = | 0.022 m |
| Outside diameter | = | 0.026 m |
| Thermal conductivity of tube metal | = | 17.5 W m$^{-1}$ K$^{-1}$ |

**Physical properties of water at mean water temperature (48°C)**

| | | |
|---|---|---|
| Specific heat | = | 4180 J kg$^{-1}$ K$^{-1}$ |
| Density | = | 989.1 kg m$^{-3}$ |
| Viscosity | = | $5.67 \times 10^{-4}$ Pa.s |
| Thermal conductivity | = | 0.641 W m$^{-1}$ K$^{-1}$ |

**Viscosity of water at tube wall temperature (94°C)** = $3.02 \times 10^{-4}$ Pa.s

7. Contaminated wastewater leaving a process at 90°C and 100 kg $s^{-1}$ is to be used to heat process water flowing at 120 kg $s^{-1}$ from 40°C to 70°C. Factory space is limited, so it has been decided to use a plate heat exchanger (PHE).

If the maximum allowable total frictional pressure drops on the wastewater and process water sides of the proposed PHE are 200 and 350 kPa respectively, determine what design the PHE must have, i.e. what plate area, how many plates and how many passes for each liquid. ***Answer: 0.958 m$^2$; 187 plates; 2 passes are satisfactory.***

Assume that there will be countercurrent flow in each pass. Assume that fouling is absent.

Try a 2 pass/2 pass arrangement. Will it be satisfactory - i.e. will the rate of heat transfer be within, say, 15% of that required? ***Answer: Yes.***

8. A PHE is to be used as a regenerator in a soft drink pasteurizing plant. Hot pasteurized drink leaving the plant at 94°C is to be used to preheat cold, unpasteurized drink entering the plant from 48°C to 88°C. The drink flow rate is 7.78 kg $s^{-1}$.

Determine the plate area, the number of passes required and the number of plates required for this duty given that the available total frictional pressure drops on the hot and cold sides of the regenerator are each 130 kPa. Assume that flow will be countercurrent in each pass and assume that the soft drink has the same physical properties as water. Assume also that fouling is absent.

Try a 5 pass/5 pass arrangement first. Will it be satisfactory, or will the number of passes on each side need to be increased or decreased? If an increase is required, try 7 pass/7pass. If a decrease is required, try 2 pass/2 pass. ***Answer: plate area = 0.254 $m^2$; 5/5 is unsatisfactory (124 plates); 7/7 is satisfactory (212 plates).***

9. You approached a plate heat exchanger (PHE) supplier and asked her for a technical quote for a PHE to perform the following duty:

To cool 36 kg $s^{-1}$ of beer from 70°C to 17.5°C using chilled water as the service (secondary) liquid.

The chilled water is available at 2°C and 30 kg $s^{-1}$.

Allowable frictional pressure losses on the beer and water sides of the PHE are 60 kPa and 55 kPa respectively.

Fouling of plate surfaces is not expected to be a problem.

The PHE supplier has come back to you and suggested a PHE with 340 plates (individual plate area = 0.53 $m^2$) installed to give a 3 pass/3 pass mainly counterflow arrangement. She claims that this set-up will meet both the heat transfer and pressure drop requirements of the duty.

(a) Carry out appropriate calculations to confirm or refute her claims. Is plate size satisfactory? Will the pressure drop constraints be met? Will the beer be cooled to the correct temperature (or will it be under- or over-cooled)? ***Answer: plate size OK (0.53 $m^2$); pressure drops OK ; but beer would be undercooled.***

Assume that the specific heat of both the beer and the water can be taken as 4.2 kJ $kg^{-1}$ $K^{-1}$. Assume, similarly, that the density of each liquid can be taken as 1000 kg $m^{-3}$.

(b) Will you accept the PHE supplier's technical quote, or will you reject it? If you accept it, say why you think it is satisfactory. If you reject it, what change or changes would you suggest should be made to her specification before further calculations are carried out? ***Answer: See model answer.***

10. Water used to spray-cool peas after blanching (a heat process for destroying enzymes) is circulated through one side of a plate heat exchanger (PHE) for recooling. A second flow of water, used to spray-preheat the peas entering the blancher, is circulated counter-currently through the other side of the PHE for re-warming. (In this way, heat removed from the outgoing peas during cooling is re-used for preheating the incoming peas, thus saving energy).

The cooling water and the preheating water each have the same flow rate of 7.5 kg $s^{-1}$. The cooling water enters the PHE at 60°C and is required to leave at 38°C. The preheating water enters the PHE at 48°C.

The existing PHE has to be taken out of service suddenly owing to sever gasket failure (resulting from a maintenance oversight). Fortunately, a spare PHE is available. This PHE is already set up with 198 plates (of individual plate area = 0.25 $m^2$), arranged to give 3 passes on each side.

Making use of the data given below, carry out suitable calculations to show whether or not the spare PHE will cope with the heat recovery task described, while making full use of the available pressure drops of 14.8 kPa on each side. ***Answer: the PHE <u>will</u> cope with the required duty.***

**USEFUL DATA**

1. The surfaces of the plates in the PHE become fouled by substances originating in the peas. Previous experience with the old PHE suggests that 0.000125 $m^2$ K $W^{-1}$ is a typical value for the individual fouling resistance. Fouling conditions are the same on each side of the PHE.

2. The mean water temperatures in the PHE are sufficiently close to 40°C to assume the following values for density and specific heat: 1000kg $m^{-3}$ and 4200 J $kg^{-1}$ $K^{-1}$, respectively.