Massey University

School of Food and Advanced Technology

280.371 PROCESS ENGINEERING OPERATIONS

Heat Exchanger Design

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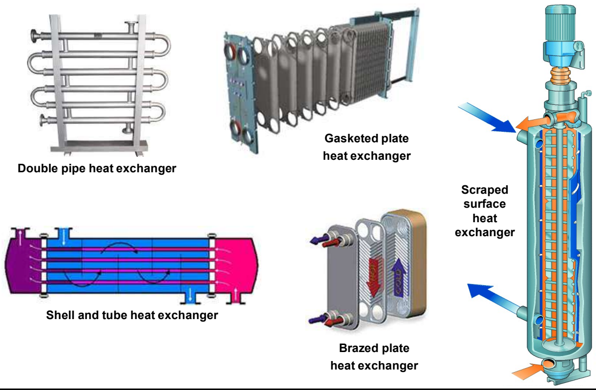
Overview

- Introduction
  - Classification of HEs
  - Revision thermal design of HEs (280.271)
  - Revision fluid flow friction losses (280.272)
  - Double pipe heat exchanger design examples
- Shell and tube heat exchangers (STHE)
  - F_T - LMTD & ε - NTU methods
  - STHE design example
- Plate heat exchangers (PHE)
  - Design features
  - Approximate design method
  - PHE design example

Introduction

- Lectures deal with continuous indirect HEs
  - Constant flow of product and service fluids
  - Steady state, i.e. temperature profiles are constant
  - Indirect => HE surface separating fluids
- Optimal design meets both
  - Heat transfer duties
  - Minimises the capital and operating costs
- Need to also consider fluid mechanics
  - High ΔP can increase pumping costs
  - Low ΔP can decrease HTCs => larger HE
- Hence, need integrated thermal and hydraulic design

Common Types of HEs



Classification of HEs

- By fluid
  - Liquid - Liquid
  - Liquid - Condensing/evaporating fluid
  - Gas - Liquid
- By flow arrangement
  - Cocurrent or parallel flow
  - Countercurrent flow
  - Cross flow
  - Mixed flow

  } liquid - liquid, liquid - gas, gas - gas

  - Isothermal — liquid - condensing/evaporating fluid, gas - condensing/evaporating fluid

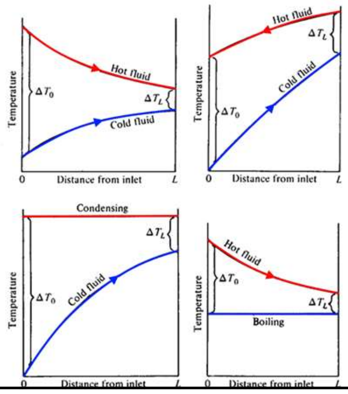
Pure Flow Arrangements

Cocurrent

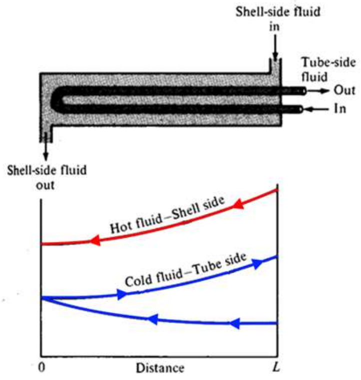
Countercurrent

Condensing vapours

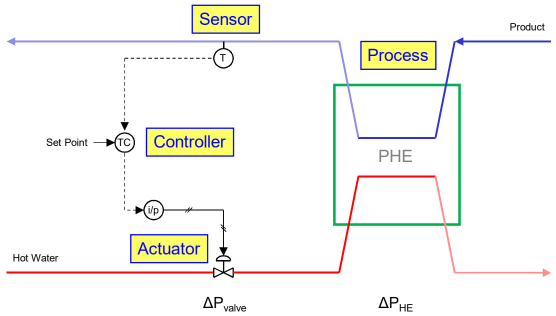
Boiling



Mixed Flow Arrangement



Feedback Control of a PHE



Temperature Control of a HE

- Underlying concept
  - Control valve modulates the flow rate of one of the streams, usually the working fluid
    - If increase valve open % -> reduces ΔP_{valve} -> increases ΔP_{HE} -> increases flow rate
  - Changes the temperature profile along HE
    - If increase flow rate -> temperature profile for that stream moves closer to horizontal -> increases ∇T_{lm} -> increases q
- Heat exchanger must be sized for worst case, i.e. highest required q
  - For good control, valve should be partially open under normal operating conditions, say ⅔ open
- Split available ΔP between valve and HE

General Heat Transfer Equations

- Energy balance over HE (assuming no losses):

$$q = m_a c_a \Delta T_a = m_b c_b \Delta T_b$$

or

$$q = C_a \Delta T_a = C_b \Delta T_b$$

where ΔT represents a temperature change
C represents a capacity rate, i.e. mc

- Heat transferred (q) in a HE is given by:

$$q = UA\nabla T_{tm}$$

where ∇T_{tm} is the true temperature difference

True Mean Temperature Difference

- For pure countercurrent, cocurrent or isothermal:

$$\nabla T_{tm} = \nabla T_{lm} = \frac{\nabla T_1 - \nabla T_2}{\ln\left(\frac{\nabla T_1}{\nabla T_2}\right)}$$

where ∇T_{lm} is the log mean temperature difference (LMTD)

- All other flow arrangements, e.g. mixed, crossflow

$$\nabla T_{tm} = F_T \nabla T_{lm}$$

where F_T = correction factor, always $F_T < 1$

Overall Heat Transfer Coefficient

- For a plane heat transfer surface (a slab) or for a thin walled cylindrical heat transfer surface:

$$\frac{1}{U} = \frac{1}{h_a} + \frac{x_w}{k_w} + \frac{1}{h_b} + R_a + R_b$$

  - Units of U, h_a, h_b (W m^{-2} K^{-1}) and R_a, R_b (m^2 K W^{-1})
- Assume U to be constant
  - May vary a bit along the HE if ΔTs are large
- For thin walled cylindrical surfaces ($r_o/r_i < 1.05$)
  - Use arithmetic average of r_o & r_i to calculate A
- For thick walled cylindrical surfaces
  - Need to allow for $A_a \neq A_b$

HTC Correlations

$$Nu = a Re^b Pr^c \left(\frac{\mu}{\mu_w}\right)^d$$

$$\frac{hD}{\lambda} = a\left(\frac{Dv\rho}{\mu}\right)^b \left(\frac{c_p\mu}{\lambda}\right)^c \left(\frac{\mu}{\mu_w}\right)^d$$

$$1 \geq b \geq c$$

a, b, c and *d* found from fitting experiment data

Pressure Drop Along a Pipe

- Pressure drop along a circular pipe is given by:

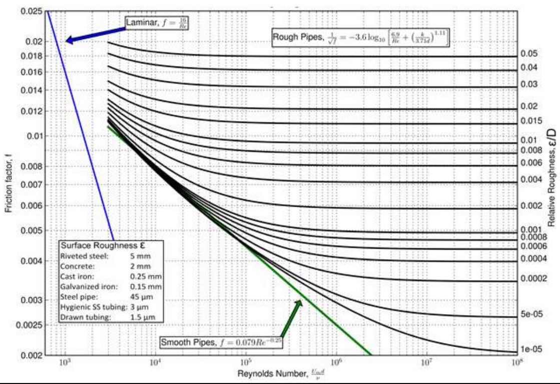
$$\Delta P = 4 f_F \frac{L}{D} \frac{\rho v^2}{2}$$

where f_F is the Fanning friction factor, obtained from Moody diagram or equations

- For turbulent flow in non-circular ducts, replace D with the hydraulic diameter:

$$D_h = \frac{4 \times \text{flow area}}{\text{wetted perimeter}}$$

Moody Diagram



Example 1a: Double Pipe HE

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?

Suggested Calculation Strategy

Are all inlet and outlet temperatures specified or available from energy balance?

- NO –> Use ε-NTU method
- YES –> Is the flow regime pure countercurrent, cocurrent or isothermal?
  - YES –> F_T = 1, use ∇T_{lm}
  - NO ->Use F_T - LMTD method
  - Note F_T - LMTD and ε - NTU approaches are different ways of grouping the variables involved in HT. The equations are related.

Example 1b: Double Pipe HE

As before, 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.

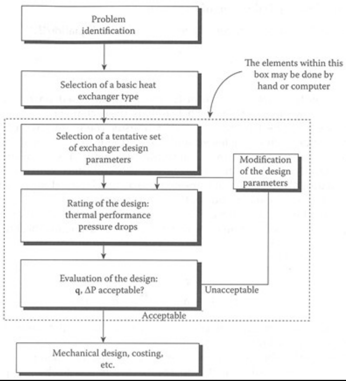
- Will the heat exchanger from example 1a still be suitable for the application?

(NB: this example better demonstrates the integration of thermal design and fluid mechanics)

HE Design vs Rating

- Examples 1a & 1b demonstrate the two main different kinds of HE problems:
  - **1a Design/Sizing** – sizing of a **new** heat exchanger, which afterwards is built. Usually includes calculating the required area (A)
  - **1b Rating** – performance determination of an **existing** or fully-specified heat exchanger. Commonly requires either:
    - Yes/no decision on whether it will meet a particular duty, e.g. comparing required q with predicted q
    - Calculating the outlet temperatures
- Rating and Design are often iterative processes

HE Design Process



- Complex relationships between design parameters
  - Often, not possible to express unknown parameters explicitly
- In practise, design procedure may consist of multiple rating steps

Design Modification

- After the first calculation it is often necessary to adjust design parameters and recalculate, this phase is termed: Design Modification, e.g.
  - If q is too low can
    - Increase A by lengthening flow passages
    - Increase film HTC on controlling side, say by increasing fluid velocity
  - If ∇T_{tm} is very low can
    - Change to be more fully countercurrent flow
  - if the ΔPs are not satisfactory can
    - Change number of parallel flow passages per pass
    - Change the duct equivalent diameter

ΔP and HT Performance

- Heat transfer rate per unit of surface area per unit of temperature difference increases by increasing the fluid velocity.

$$h = \frac{q}{A \nabla T_{tm}} \propto v^{0.8}$$

- Friction power expenditure per unit of surface area also increases with velocity.

$$\frac{\Delta P}{A} Q = \frac{2 f_F \rho v^2 Q}{\pi D^2}$$

Fanning friction factor is a function of Re which is a function of velocity → Friction power expenditure per unit surface area is proportional to $v^{2 \to 3}$

ΔP and HT Performance

- Can make use to relative exponents (0.8 vs 2-3) during design modification phase (i.e. iteration)
- If friction power (ΔP) expenditure is <u>too high</u>:
  - Can reduce fluid velocities (v ↓) by increasing the number of parallel flow passages in the exchanger
  - This will also reduce heat transfer rate (q), but by much less due to relative exponents
- However, if heat transfer rate is now <u>too low</u>:
  - Can increase the surface area (A ↑) by lengthening the flow passages
  - Overall, still results in a lower ΔP

ΔP and HT Performance

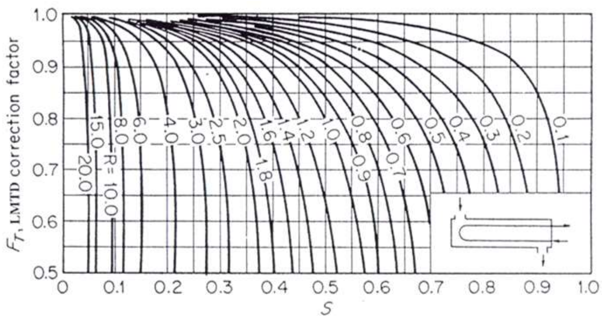
- Power expended to overcome fluid friction is the non-thermal price paid for heat transfer and is the main **running cost**.
- To keep the running cost low:
  - Need a large **surface area** which is expensive (size contributes most to the exchangers **capital cost**).
- The higher the friction power expenditure, the smaller and cheaper the exchanger can be.
  - Important to utilise any **available pressure drop** in the processing system that the exchanger is part of → size and cost can be minimised.

F_T - LMTD Approach

- F_T – LMTD correction factor
  - Depends on flow arrangement of HE (degree of counter-current flow)
  - Depends on inlet and outlet fluid temperatures
  - Find from chart
  - Calculate R and S using definitions given on chart.

$$R = \frac{\Delta T_{min}}{\Delta T_{max}} \qquad S = \frac{\Delta T_{max}}{\Delta T_{max\,possible}}$$

F_T - LMTD Approach



ε - NTU Approach

- Effectiveness (ε)

  Compares the actual heat transfer rate to the thermodynamically limited maximum possible heat transfer rate.

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_a \Delta T_a}{C_{min}\left(T_{a,i} - T_{b,i}\right)} = \frac{C_b \Delta T_b}{C_{min}\left(T_{a,i} - T_{b,i}\right)}$$

C_{min} is the smaller of C_a and C_b

$(T_{a,i} - T_{b,i})$ is the maximum possible temperature change.

$$\varepsilon = \frac{C_{min} \Delta T_{max}}{C_{min}\left(T_{a,i} - T_{b,i}\right)} = \frac{\Delta T_{max}}{\Delta T_{max\,possible}}$$

ε - NTU Approach

- Capacity-rate ratio (C_R)

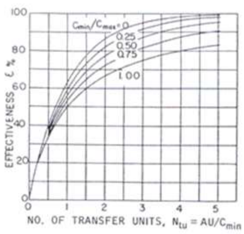
$$C_R = \frac{C_{min}}{C_{max}} \quad \left(= \frac{\Delta T_{min}}{\Delta T_{max}} \right)$$

- Number of transfer units (NTU)

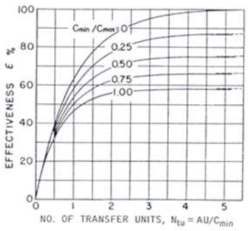
$$NTU = \frac{UA}{C_{min}} = \frac{\Delta T_{max}}{\Delta T_{lm}}$$

Dimensionless expression of heat transfer size of the HE. For a given C_{min}, if NTU is small (small U and/or A), ε is low. If NTU is large, ε approaches the limit imposed by the flow arrangement and thermodynamic considerations.

ε – NTU Charts



Pure
Countercurrent



1 Shell and
2 Tube Passes

ε - NTU Approach

- Number of transfer units (NTU)
  - A large NTU and high ε are attained at cost
    - Large A → expensive HE.
    - Large U → high velocities in the HE flow passages and therefore high friction power requirements and high running costs.
- There is a penalty paid for any departure from pure counter-current flow.
  - For a given C_R and ε, a higher NTU is required.
- Relationships between ε, C_R and NTU
  - REMEMBER: $C_R = 0$ for a condensing/evaporating fluid.

ε - NTU Approach

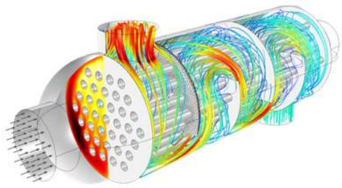
- For a Rating problem where the outlet temperatures are sought for a given HE:
  1. Calculate $C_R = C_{min}/C_{max}$ and $NTU = UA/C_{min}$
  2. Determine ε from the appropriate chart
  3. Calculate q from:

$$q = \varepsilon\, C_{min}\left(T_{a,i} - T_{b,i}\right) = \varepsilon\, C_{min}\, \Delta T_{max\ possible}$$

  4. Calculate outlet temperatures from energy balance equations

STHE Rating or Design

- Tube side analysis straight forward
  - Similar to dual pipe HE example
- Shell side has a mixed flow arrangement
  - True mean ∇T_{tm} less than log mean ∇T_{lm}
  - Need to use F_T or ε-NTU methods



Heat Transfer for Thin-Walled Tubes

- Overall HTC for thin-walled tubes ($r_2/r_1 < 1.05$)

$$\frac{1}{U} = \frac{1}{h_t} + \frac{r_o - r_i}{k_w} + \frac{1}{h_{ss}} + R$$

- Average surface area of contacting liquid

$$A = 2\pi\left(\frac{r_o + r_i}{2}\right)LN$$

where L is the length of the tube bundle
N is the total number of tubes, i.e. all passes

Heat Transfer for Thick-Walled Tubes

- Overall HTC (U_o) based on the outside area for thick-walled tubes:

$$\frac{1}{A_o U_o} = \frac{1}{A_i h_t} + \frac{\left(r_o - r_i\right)}{A_{lm} k_w} + \frac{1}{A_o h_{ss}}$$

- Which can be simplified to:

$$\frac{1}{U_o} = \frac{r_o}{r_i h_t} + \frac{r_o \ln\left(r_o / r_i\right)}{k_w} + \frac{1}{h_{ss}}$$

- This is used in conjunction with the outside area

$$A_o = 2\pi\, r_o\, L\, N$$

STHE Tube Side HTC (h_t)

- Use previous correlation or best available:

$$Nu_t = \frac{\left(\frac{f_{F,t}}{2}\right)\left(Re_t - 1000\right)Pr_t}{1 + 12.7\sqrt{\frac{f_{F,t}}{2}}\left(Pr_t^{2/3} - 1\right)}\left(1 + \left(\frac{D}{L}\right)^{2/3}\right)$$

where

$$f_{F,t} = 0.25\left(1.82\log_{10}\left(Re_t\right) - 1.64\right)^{-2} \qquad Re_t = \frac{D v_t \rho}{\mu} \qquad Pr_t = \frac{c\mu}{\lambda}$$

$$v_t = \frac{Q_t}{\frac{\pi D^2}{4} N_{t/pass}} = \frac{4Q_t}{\pi D^2 N_{t/pass}} \qquad Nu_t = \frac{h_t D}{\lambda}$$

STHE Shell Side HTC (h_{ss})

- Select appropriate correlation, e.g. $Re_{ss} > 2000$:

$$Nu_{ss} = 0.36 Re_{ss}^{0.55} Pr_{ss}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

where

$$Re_{ss} = \frac{D_e v_{ss} \rho}{\mu} \qquad Pr_{ss} = \frac{c\mu}{\lambda} \qquad Nu_{ss} = \frac{h_{ss} D_e}{\lambda}$$

still need some flow parameters to solve, i.e. v_{ss} & D_e

STHE Shell Side Flow Parameters

- Shell side equivalent diameter (D_e), i.e. the hydraulic diameter of the gap between the tubes:

$$D_e = \frac{4 \times \text{cross-section area for flow}}{\text{wetted perimeter}}$$

- For a triangular pitch:

$$D_e = \frac{8\left(0.433\,P^2 - \frac{\pi}{8} D_o^{\,2}\right)}{\pi D_o}$$

- For a square pitch:

$$D_e = \frac{4\left(P^2 - \frac{\pi}{4} D_o^{\,2}\right)}{\pi D_o}$$

STHE Shell Side Flow Parameters

- Shell side velocity (v_{ss}) calculated from volumetric flow rate (Q_{ss}):

$$v_{ss} = \frac{Q_{ss}}{a_{ss}} \qquad a_{ss} = \frac{D_{ss}(P - D_o)B}{P}$$

where a_{ss} is the cross-sectional area for flow
D_{ss} is the shell diameter
P is the tube pitch (measured between centres)
D_o is the outside tube diameter
B is the baffle spacing

STHE Tube Side ΔP

- Pressure drop along pipe + 4 velocity heads per pass to allow for sudden expansions & contractions

$$\Delta P_t = 4 f_{F,t} \frac{L N_p}{D} \frac{\rho v_t^2}{2} + 4 N_p \frac{\rho v_t^2}{2}$$

where use Moody chart to find $f_{F,t}$ or:

$$f_{F,t} = 0.25\left(1.82 \log_{10}(Re_t) - 1.64\right)^{-2}$$

$$Re_t = \frac{D v_t \rho}{\mu}$$

STHE Shell Side ΔP

- Pressure drop along shell

$$\Delta P_{ss} = 4 f_{F,ss} \frac{L D_{ss}}{B D_e} \frac{\rho v_{ss}^2}{2}$$

where

$$f_{F,ss} = \frac{0.45}{(Re_{ss})^{0.195}}$$

$$Re_{ss} = \frac{D_e v_{ss} \rho}{\mu}$$

Design Modification for STHEs

- Various design options are available in the design modification step
  - If HE is limited by the heat it can transfer, need to increase U or A.
    - If h_t controlling U, increase number of tube passes (for the same number of tubes) → increasing velocity
    - If h_{ss} controlling U, decrease baffle spacing or decrease baffle cut → increasing shell side velocity
    - To increase A, increase length of HE, increase shell diameter and total number of tubes, or go to multiple shells in series or parallel.

Design Modification for STHEs

- If HE is limited by a large frictional pressure drop on tube side:
  - Decrease number of tube passes or increase tube diameter.
  - Decrease tube length and increase shell diameter and total number of tubes.
- If HE is limited by a large frictional pressure drop on shell side:
  - Increase baffle spacing or baffle cut.
  - Increase tube pitch.
- An existing shell and tube HE can only be rated. There is some scope for modification, e.g., altering the shell side baffle spacing.

Example 2: STHE Rating and Modification

1 shell/2 tube passes STHE is to be used for cooling hot oil using cold water. The oil, pumped at 57,000 L/h on the shell side, needs to be cooled from 118.5°C to 94°C. The 30°C cold water comes through the tubes at 78,000 L/h. 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. STHE specifications:

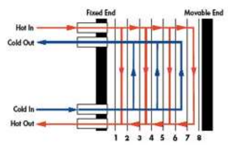
Tube length (= shell length)	= 3.6 m
Tube ID	= 0.0206 m
Tube OD	= 0.0254 m
Tube pitch (triangular)	= 0.0318 m
Total number of tubes	= 68
Number of tube passes	= 2
Number of baffles	= 8
Shell diameter	= 0.337 m
Tube thermal conductivity	= 20 W m^{-1} K^{-1}

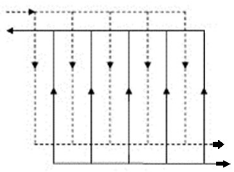
Will the heat exchanger meet the required heat transfer duty without exceeding the allowable pressure drops and make full use of the available ΔP's? The baffle spacing can be altered if necessary.

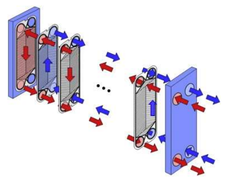
PHE Construction & Flow Arrangements

- Single pass
  - Each liquid flows through a set of parallel passages that make up a single pass.
  - Flow is countercurrent → high effectiveness and F_T close to 1.
  - *Z-configuration*
    - Inlet and outlet connections are made to both the head and the follower.
    - Gives more even distribution of fluid among parallel flow passages.
  - *U-configuration*
    - Connections made to the head only.
    - Plate pack can easily be disassembled without disturbing pipework.
    - Only possible for single pass configuration.

Single Pass PHE

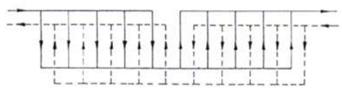






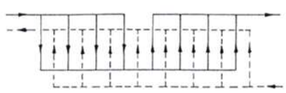
PHE Construction & Flow Arrangements

- Multipass with equal passes
  - Used when greater NTU values are required compared to a single pass and when there is sufficient pressure drop on each side.
  - Plate flow ports are blanked off to achieve this.
  - On each side of the PHE the flow channels within any given pass are in parallel, but passes themselves, when there are more than one, are arranged in series.
  - Only Z-configuration possible.



PHE Construction & Flow Arrangements

- Multipass with unequal passes
  - When the flow ratio Q_{max}/Q_{min} is high, or there is some other reason for minimising the pressure drop on one side, unequal passes can be used, with fewer passes on the low pressure drop side.
  - Can cause a high degree of parallel flow of the two liquids → reduced effectiveness for a given NTU and C_R, or decreases F_T.
  - Only Z-configuration possible.



PHE Plate Design

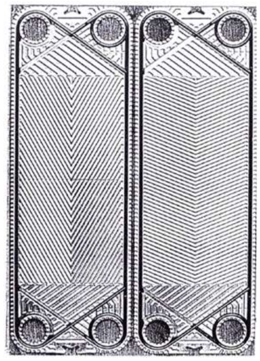
- HT and pressure drop characteristics depends on the plate aspect ratio and on the shape and pattern of profilations.
- Plate dimensions depend on the port size, which depends on throughput. The port size governs the plate width.

PHE Plate Design

- "Hard" plates
  - Long and narrow and has profilation patterns causing high flow disturbance. Gives high film heat transfer coefficient → high U and high NTU and also relatively high frictional pressure drop.
  - Used for difficult duties like regenerative heat transfer where ∇T_{tm} is low due to close temperatures at each end.
- "Soft" plates are the opposite and are used for easier duties.

Soft v Hard

SOFT plate
60° angle



HARD plate
120° angle

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PHE Selection

- Assessment of PHE suitability can be done by comparing actual flow rate and duty NTU required with that of the available range of plates:

– Individual plate area	up to ~ 2.5 m²
– NTU per pass	~ 0.3 to 3.5
– Max operating pressure	~ 2000 kPa
– Max viscosity	5 Pa.s
– Port diameter	up to 400 mm
– Max total flow rate	2500 m³/h

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PHE Sizing

- Accurate design data is proprietary
  - Supply firms will want accurate physical property data, flow rates and temperatures
- Approximate method
  - Suitable for:
    - Multipass with equal passes
    - Liquids with properties similar to water at 40°C
  - Underlying concept:
    - Fully utilise the available ΔP
  - Can also be used to determine whether a PHE is a feasible selection for a particular HT duty

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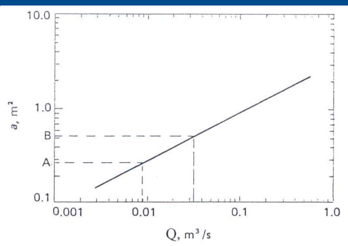
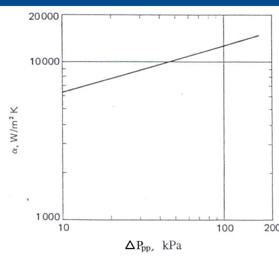
PHE Sizing – Underlying Curve 1



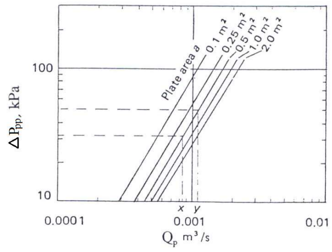
- Individual plate area (a) is related to total PHE volumetric flow rate (Q)
  - As total volumetric flow rate increases, the plate port diameter and plate size has to increase.

PHE Sizing – Underlying Curve 2



- Film heat transfer coefficient (α) related to the pressure drop for a single flow pass (ΔP_{pp})

PHE Sizing – Underlying Curve 3



- Pressure drop for a single flow pass (ΔP_{pp}) is related to the volumetric flow rate through a passage (Q_p) and the plate size

PHE Sizing – Approximate Method

- Select a plate size (a) based on volumetric flow rate
- Iterative procedure:
  - Guess number of passes (H)
  - Calculate the number of parallel channels per pass that fully utilise the ΔP available for each pass $\frac{n}{2H} = \frac{Q_{total}}{Q_p}$
  - Calculate total number of plates (n) and compare trial heat transfer rate (q_T) with the required heat rate (q_R)
    - If $q_T \approx q_R$ then stop
    - If $q_T << q_R$ then increase H and try again
    - If $q_T >> q_R$ then decrease H and try again

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PHE Sizing Procedure

Step 1

- Use energy balance equation to find flow rate Q_a and Q_b and find out the ΔP available on each side of the PHE.

$$q_R = m_a c_a \Delta T_a = m_b c_b \Delta T_b$$

Step 2

- Heat transfer area of an individual plate (a) vs total volumetric flow rate that the PHE can handle.

$$a = 2.68\, Q_{max}^{0.485}$$

where Q_{max} = highest of Q_a or Q_b

PHE Sizing Procedure

Step 3

- Guess number of passes 'H'. For example guess 3 passes, H = 3

$$\Delta P_{pp} = \frac{\Delta P_{available}}{H}$$

Step 4

- Calculate the heat transfer coefficient of fluid 'a' and 'b'

$$\alpha_{a/b} = 3057(\Delta P_{pp})^{0.308}$$

Pressure drop per pass

Units in kPa

PHE Sizing Procedure

Step 5

- Calculate U_T from:

$$\frac{1}{U_T} = \frac{1}{\alpha_a} + \frac{1}{\alpha_b} + R$$

PHE Sizing Procedure

Step 6

- Eqn (54) from study notes give the flow rate per passage Q_p based on the plate area and the pressure drop per pass.

$$Q_{p\,a/b} = 0.000127\ (a)^{0.224}\ (\Delta P_{pp\,a/b})^{0.584}$$

PHE Sizing Procedure

Step 7

– Using the higher of the Q_p values calculated in step 6 and the higher of the two total flow rates (Q_{max}), calculate the number of plates required 'n'.

$$n = \frac{2HQ_{max}}{Q_{p\ max}} \qquad Q_{p\ max} = \text{higher of } Q_{p_a} \text{ or } Q_{p_b}$$

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PHE Sizing Procedure

Step 8

$$\nabla T_{lm} = \frac{\nabla T_1 - \nabla T_2}{\ln\left(\frac{\nabla T_1}{\nabla T_2}\right)}$$

Step 9

– Calculate q_T and compare to the required q_R

$$q_T = U_T n a \nabla T_{lm}$$

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PHE Sizing Procedure

Step 10

– Repeat steps 3 to 9 with different H values until agreement is reached (q_T and q_R within ± 10%).

Then,

Total heat transfer surface area = n a

NOTE

Equations from Step 2, 4, 6 are valid for water at 40°C with the following properties:

$\rho = 1000 \text{ kg m}^{-3}$

$c = 4200 \text{ J kg}^{-1}\text{K}^{-1}$

$\lambda = 0.63 \text{ W m}^{-1}\text{K}^{-1}$

$\mu = 0.65 \times 10^{-3} \text{ Pa s}$

If fluid properties deviate too much…

Liquid Other Than Water at 40°C

- If liquids other than water at 40°C or physical properties of liquid differs more than ±10% the procedure needs to be modified
- Conversion factors are available in the literature but these will not be covered in this course

Example 3: PHE Sizing

Design a PHE capable of heating 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.