HEAT TRANSFER AND HEAT EXCHANGE EQUIPMENT

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IMPERIAL FLOWSHEET

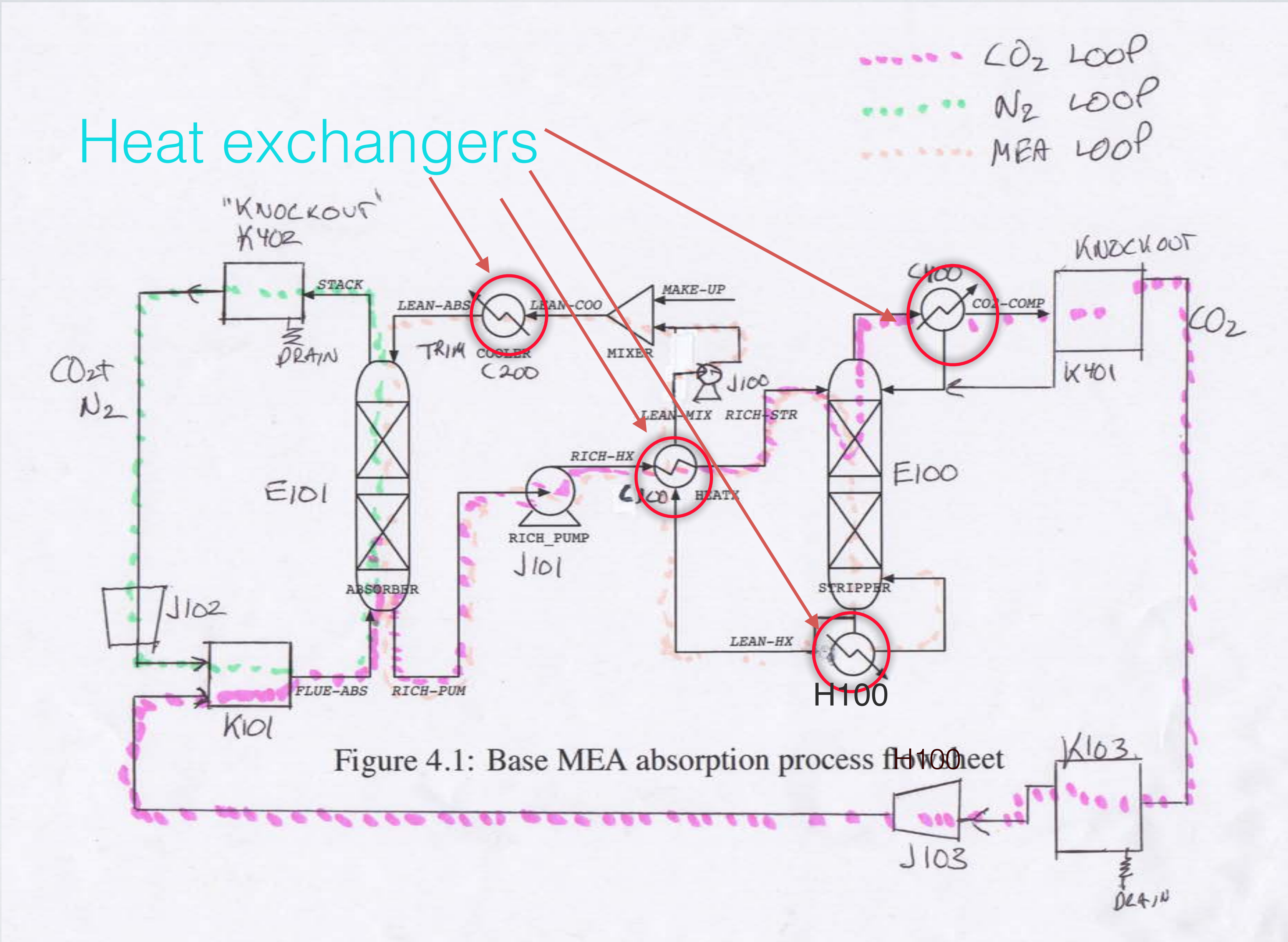

Figure 4.1: Base MEA absorption process flowsheet

IMPERIAL HEAT EXCHANGERS

C100

H100

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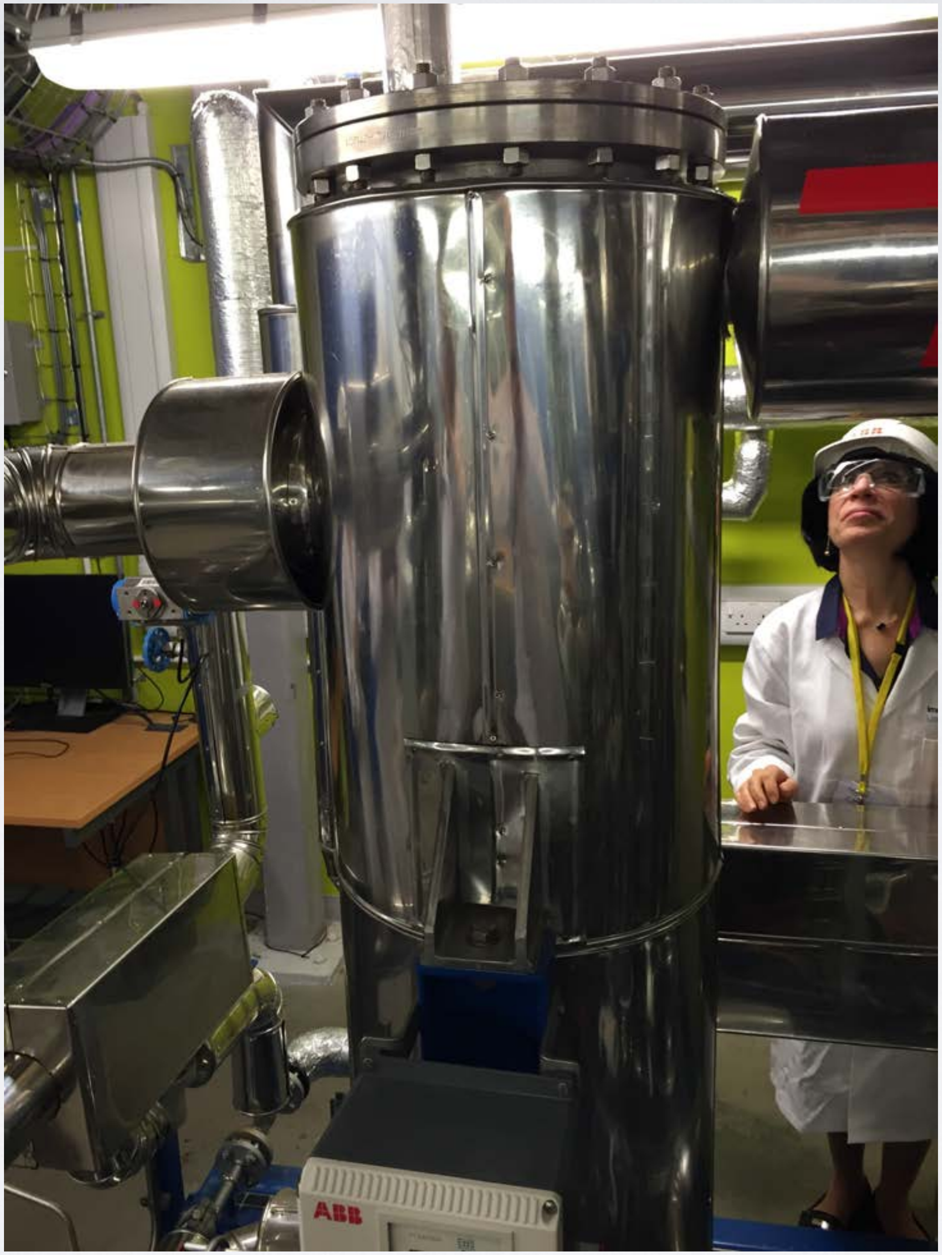
IMPERIAL HEAT EXCHANGERS

C200

H200

C106

PROFESSOR SADDAWI INSPECTS THE REBOILER!



H100

SPIRAL HEAT EXCHANGER

C400

FLOW SHEET HEAT EXCHANGERS

- Regeneration heat (added to bottom of stripping column)
  - "reboiler"
  - boils the MEA-water mixture and the "steam" strips the CO_2
    - Steam provides the heat
- Chiller before MEA is fed to absorber
    - "Trim cooler"
      - Chilled water
- "Intercooler"("clean" and "dirty" streams exchange heat)
    - counter current plate/frame heat exchanger
- "Condenser" on top of stripping column
  - condense water and MEA, let CO_2 pass through to recycle
    - Chilled water: Spiral geometry

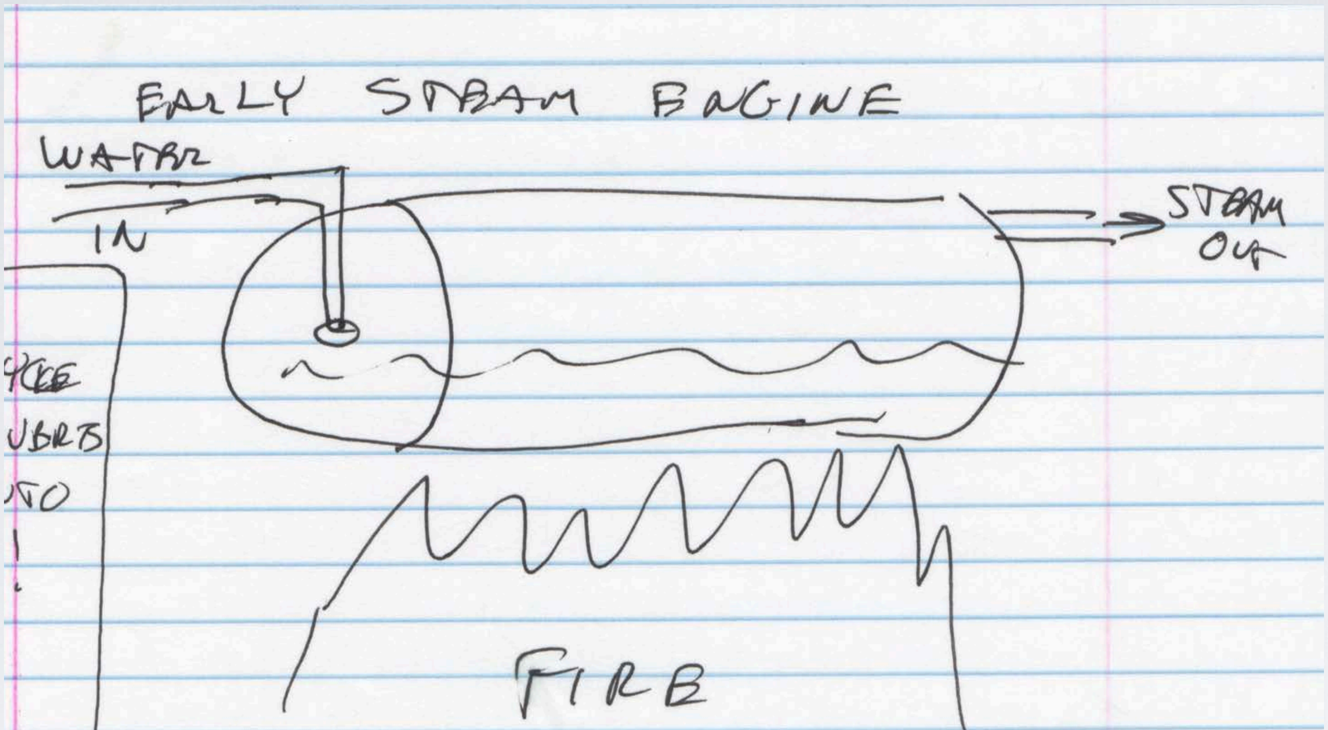
HEAT EXCHANGERS

- Two basic "ideas"… a.k.a. equations:
  - Energy is conserved
    - **First law of thermodynamics**
    - You have already done such calculations!
  - *Rate* of heat transfer will determine the total "transfer area" needed for the heat exchanger
    - Newton's law of cooling
    - This may be new to you

HEAT TRANSFER FUNDAMENTALS

- We see that understanding heat transfer is essential to knowing exactly how the process operates.
- Let's see if we can efficiently learn some fundamentals.

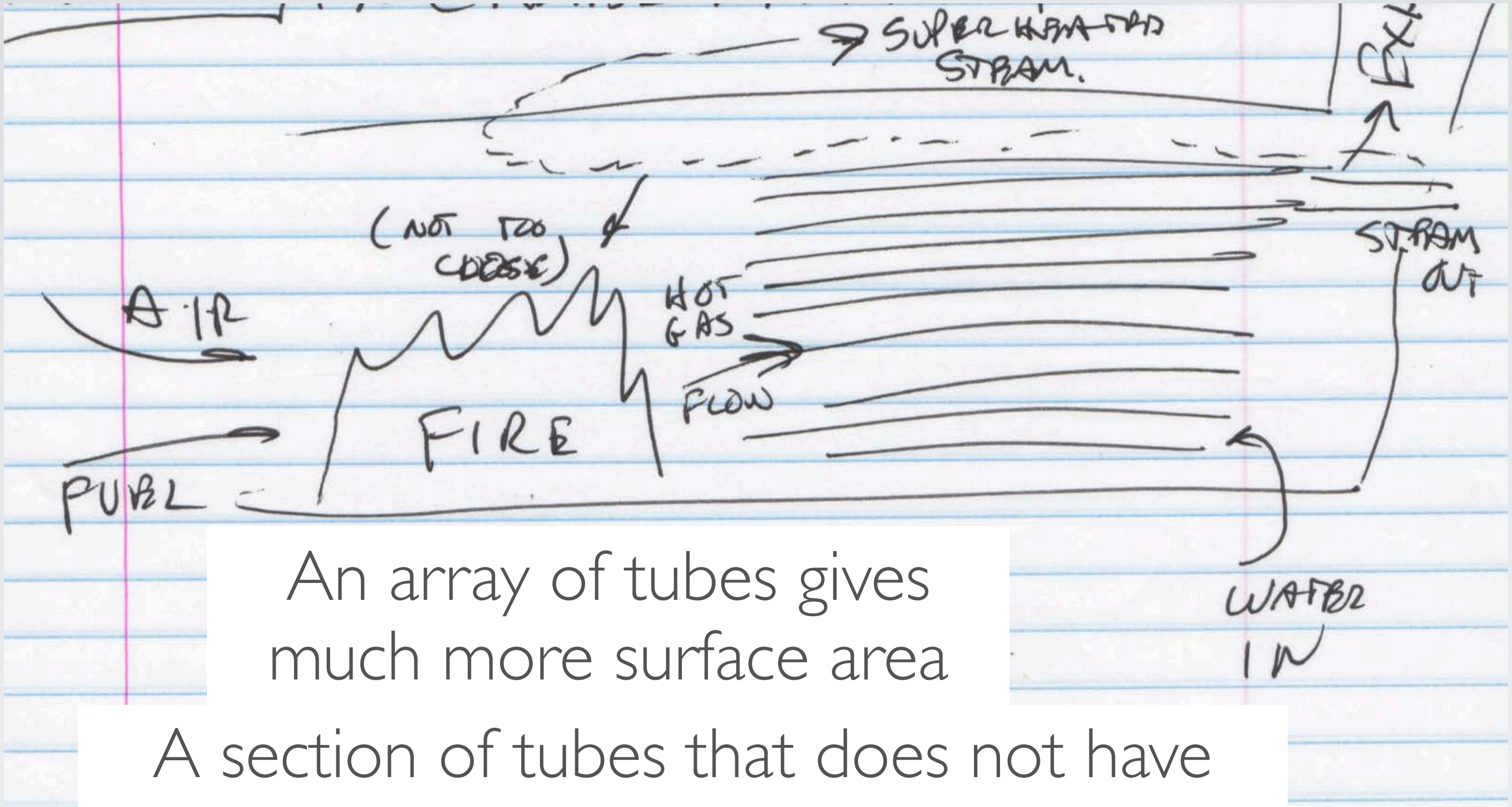
A PRIMITIVE BUT TECHNOLOGICALLY IMPORTANT HEAT EXCHANGER



This device could produce saturated steam at a pressure somewhat above 1 atmosphere

We can see that a lot of the heat from the fire is lost and does not heat the water.
We need more contact area between the hot combustion gases and the liquid water

A BETTER HEAT EXCHANGER



An array of tubes gives much more surface area

A section of tubes that does not have liquid water in it allows the steam to become superheated and hence increases its ability to do work.

"BIG BOY" STEAM LOCOMOTIVE

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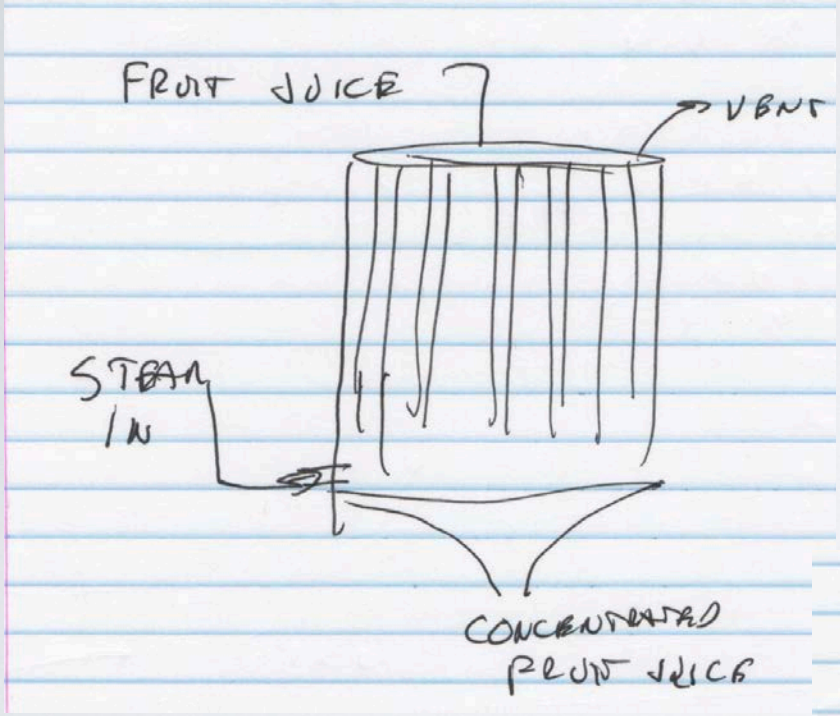
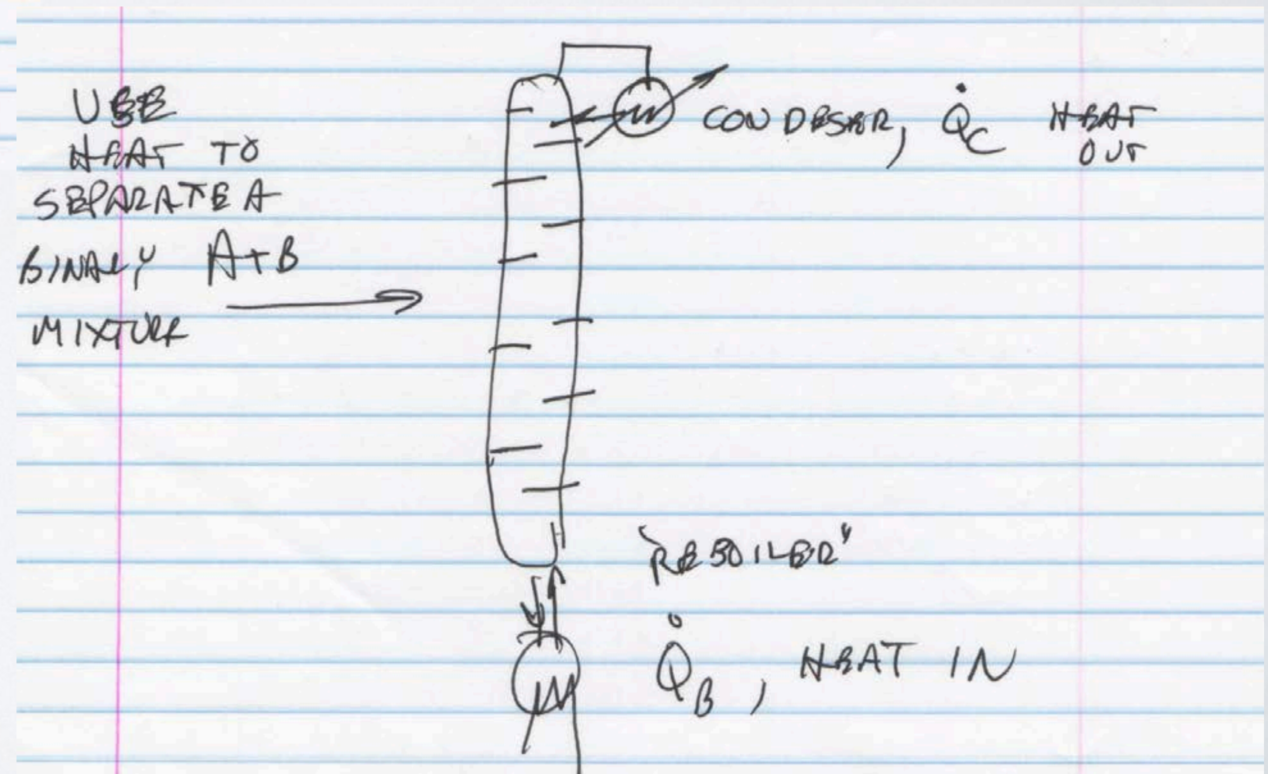
OTHER TECHNOLOGY EXAMPLES





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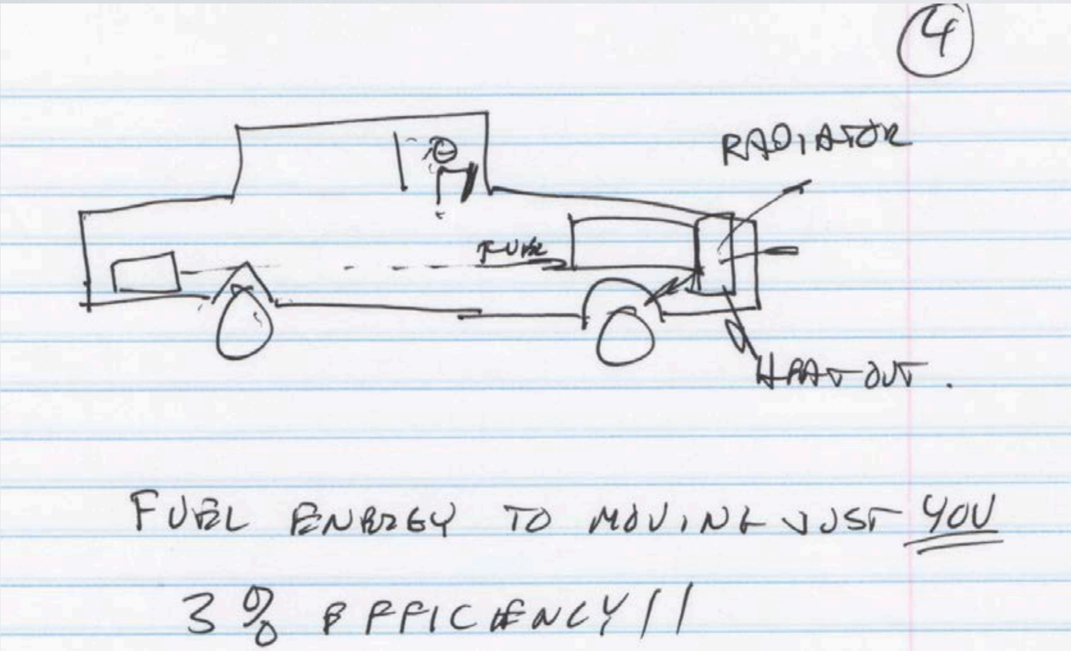
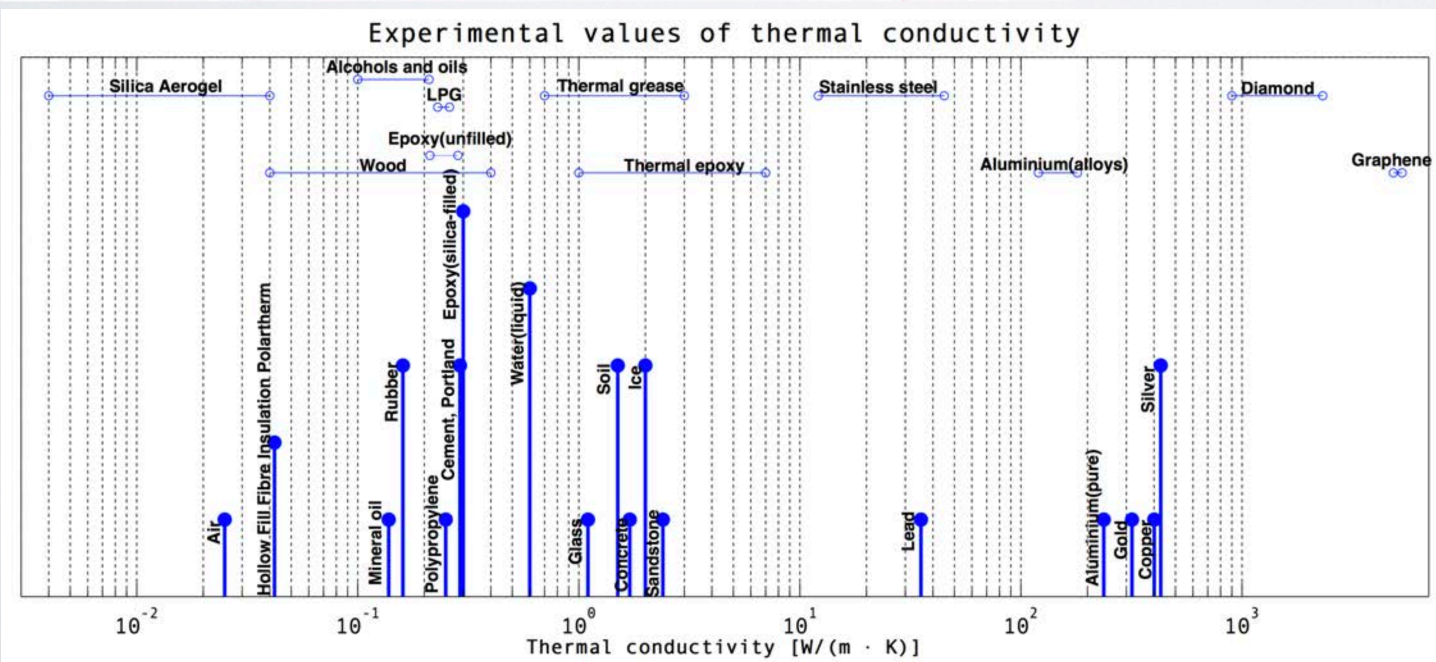
A CAR:



- For air exchange note that heat exchangers typically have "fins". This is because a low density gas is a good insulator, not a good heat transfer fluid!



https://upload.wikimedia.org/wikipedia/commons/1/1e/Thermal_conductivity.svg

MODES OF HEAT TRANSFER

- Radiation
  - Heat transport by electromagnetic waves
- Conduction
  - Transport by molecular/atoms vibrating (for solids). Free electrons for metals. Random molecular motion for gases and liquids.
- Convection
  - Transport by net motion of fluid. (molecular motion that is correlated, not random)

RADIATIVE HEAT TRANSFER

- The third mode of heat transfer is "radiation". This is transfer of energy through a "transparent" medium by electromagnetic waves.

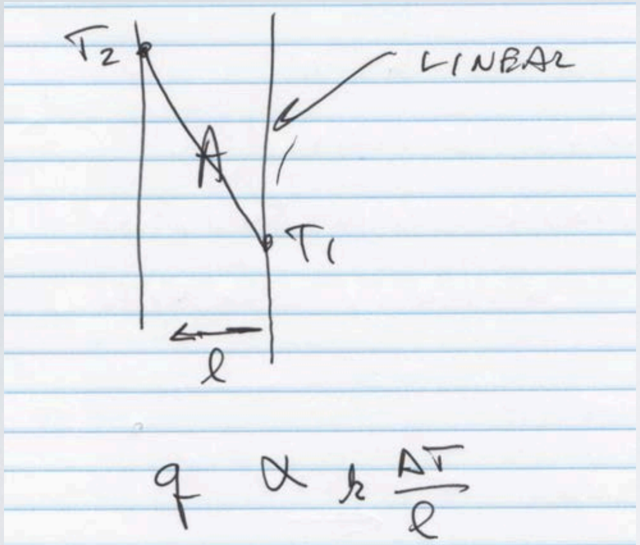
- The power of temperature in the driving force is "4", that is

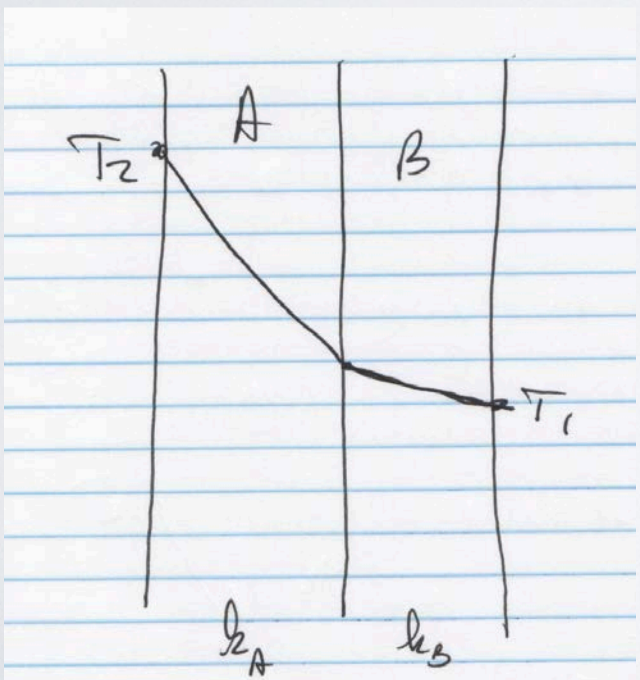
  - $q \sim \boldsymbol{\varepsilon}\, \boldsymbol{\sigma}\, (T^4 - T_0^4)$, **ε**, is the "emissivity"and **σ** is the Stefan-Boltzmann constant .

  - We don't neglect radiation entirely as you can see that the outside of the absorber and stripper are "shiny metal", for which **ε**=~.06. The emissivity is close to 1 for dark colored, slightly roughened surfaces and exactly 1 for a "black body".

STEADY CONDUCTION IN A SOLID

- Fourier's "Law" of heat conduction.

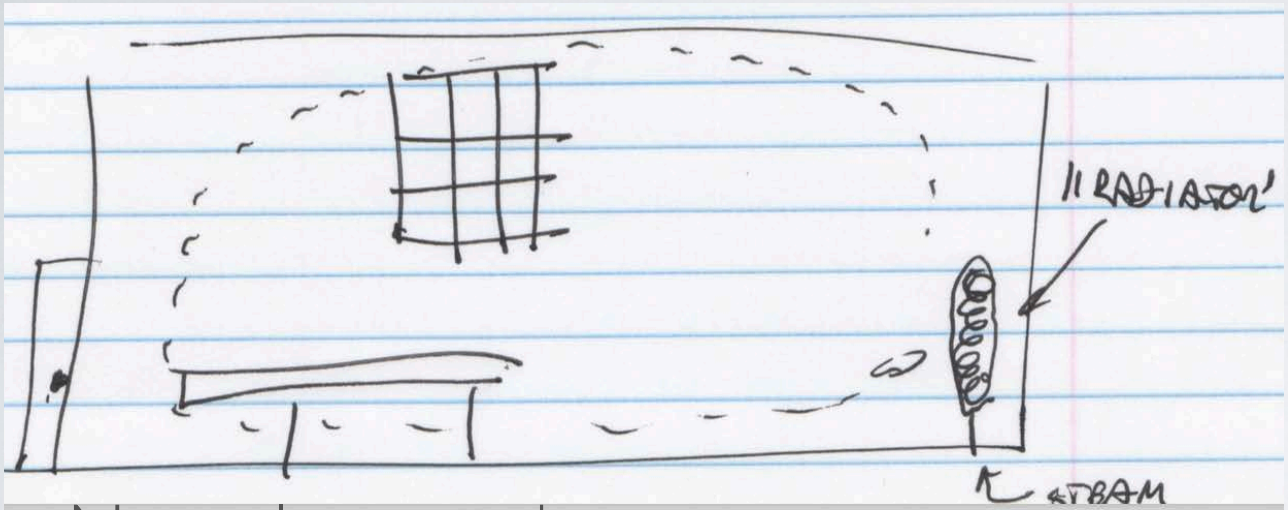




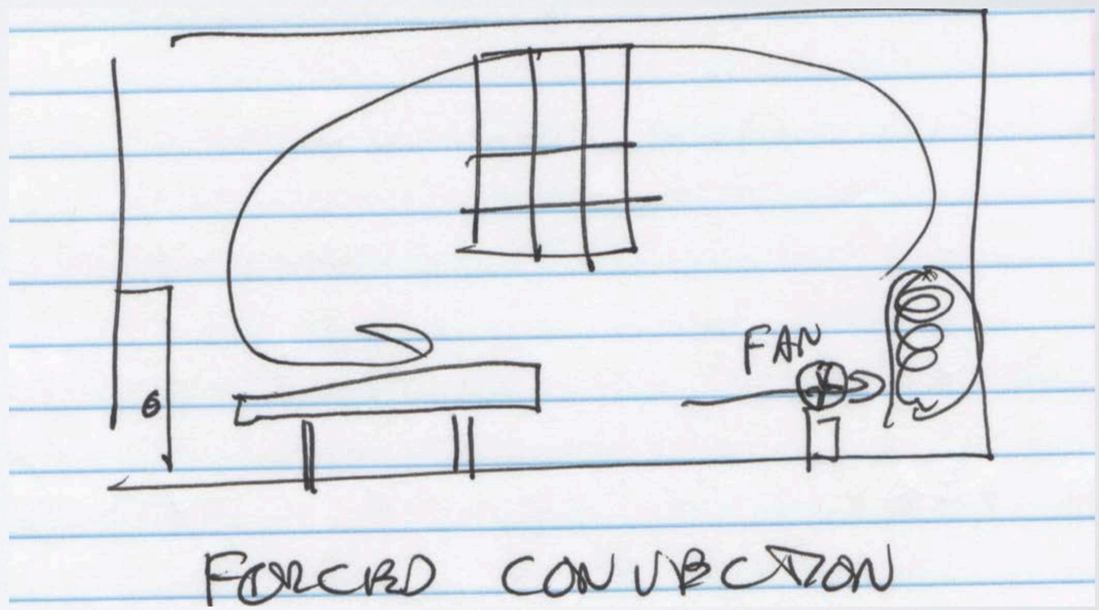
$$q = \text{CONST}$$

$$k_A \frac{\Delta T_A}{\ell} = k_B \frac{\Delta T_B}{\ell}.$$

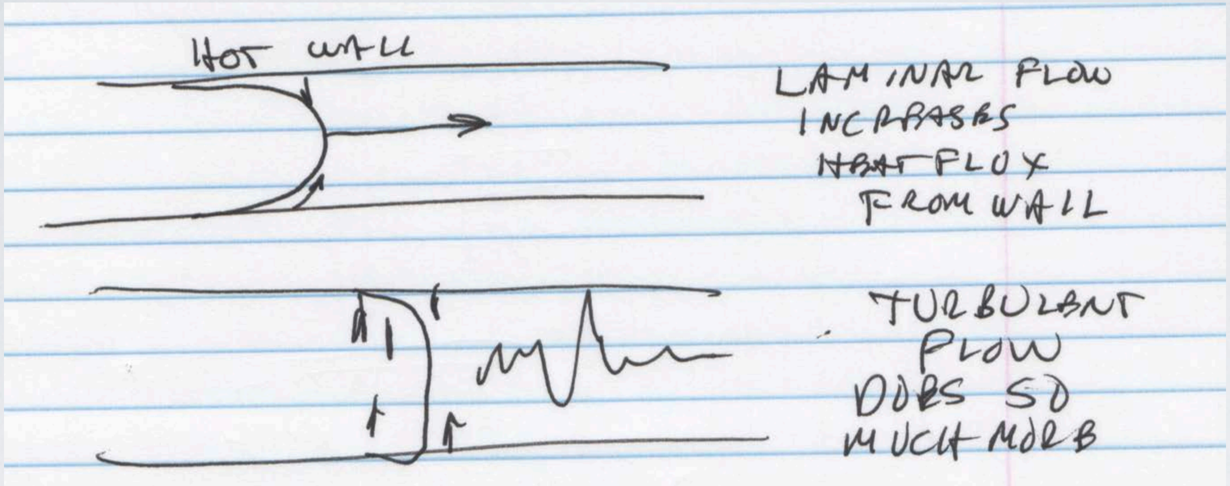
CONVECTION



Natural convection — flow is from buoyancy of gas



TURBULENCE GREATLY INCREASES HEAT TRANSFER



- Note that direction of heat transfer for this problem is the radial coordinate of the pipe.
- We hypothesize a constitutive equation for these convective flow situations (since it does not seem possible to solve the differential equations for turbulent flow…

$$q' = hA(T_w - T_\infty)$$

q' – heat flux

h – heat transfer coefficient

A – area of contact

T_w – wall temperature

T_∞ – temperature away from wall in the stream

NEWTON'S LAW OF COOLING

- $Q = hA(T - T_0)$

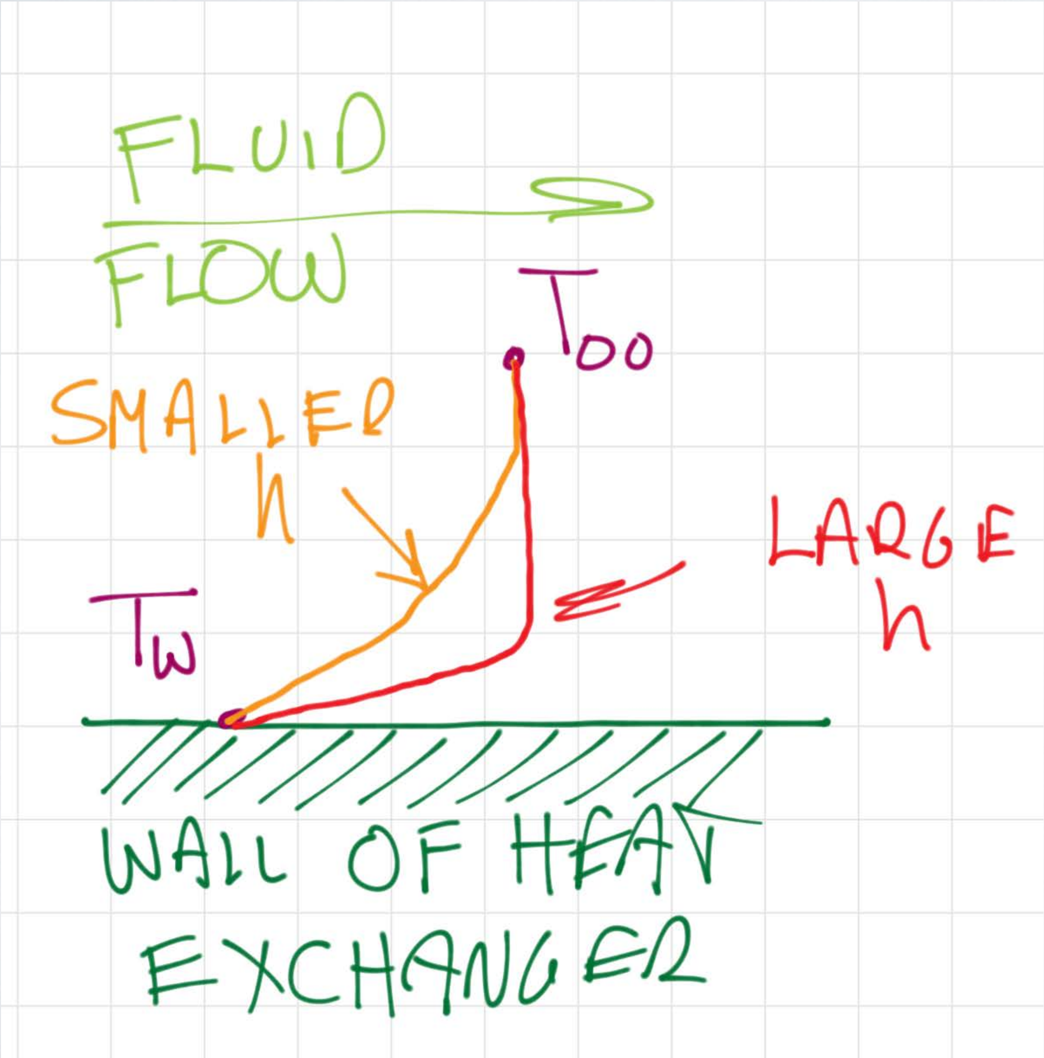
- This *empirical* equation says that the heat flow from a boundary (say a pipe wall), for a flowing fluid is the product of the temperature difference: *Thermodynamic* driving force and

- A variable, *h*, the "heat transfer coefficient" that is a function of the intensity of the fluid mixing and the physical properties (e.g., thermal conductivity) of the fluid

  - The underlying physical processes that determine "*h*" are the subject of the courses of *Transport Phenomena*

  - We see that *Thermodynamics* tells us what can occur and *Transport Phenomena* tells us how fast it will occur.

WALL REGION OF HEAT EXCHANGER: ''FORCED CONVECTION''



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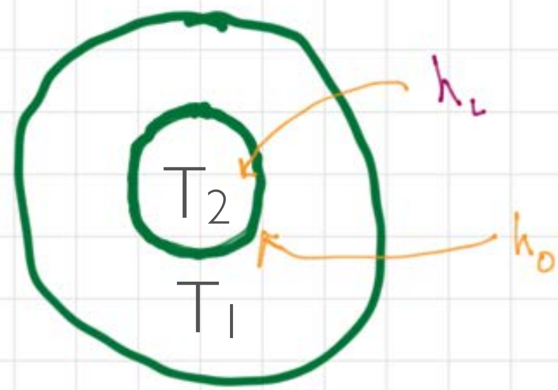
Steeper gradient is associated with faster heat transfer

ANOTHER BIT OF LINGO: *BOUNDARY-LAYER*

- For process flows, the Reynolds number is usually very much larger than unity and in most cases the flow is turbulent.
- Thus, convection is the dominant mode of heat transfer across most of the pipe.
  - In this region, the temperature changes very little.
- However, because the fluid velocity is "0" at the wall: (no slip), the *convection* near the wall is greatly <u>decreased</u> and hence *conduction* becomes relatively more important.
- This region, near the wall (or potentially at the boundary between two fluid phases) is termed a "boundary-layer".
  - In this region conductive and convective transport effects are of the <u>same magnitude.</u>
  - Also the <u>temperature gradient</u> is very much larger within the boundary-layer than far away from the boundary where convection is dominating.

HEAT EXCHANGER

Cross-section



Heat transfer resistance occurs at both inside and outside of inner pipe, hence there is a heat transfer coefficient for each side.

Sum of resistances used to get an overall heat transfer coefficient, U

$$U_i \equiv \frac{q}{A_i \Delta T}$$

$$\Delta T = T_2 - T_1$$

$$\frac{1}{A_i U_i} = \frac{1}{h_i A_i} + \frac{1}{k\Delta r} + \frac{1}{h_o A_o}$$

inside resistance

outside resistance

resistance associated with pipe wall

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{A_i}{k\Delta r} + \frac{A_i}{h_o A_i}$$

The resistance of the pipe wall will usually be much smaller than the contributions from the heat transfer coefficients if the pipe is made of a metal, but it we want to be precise we should write it as

$$\frac{A_i \ln \frac{d_{outer}}{d_{inner}}}{2 \pi L}$$

Note that all of the areas are $\pi d L$ for a specified L.

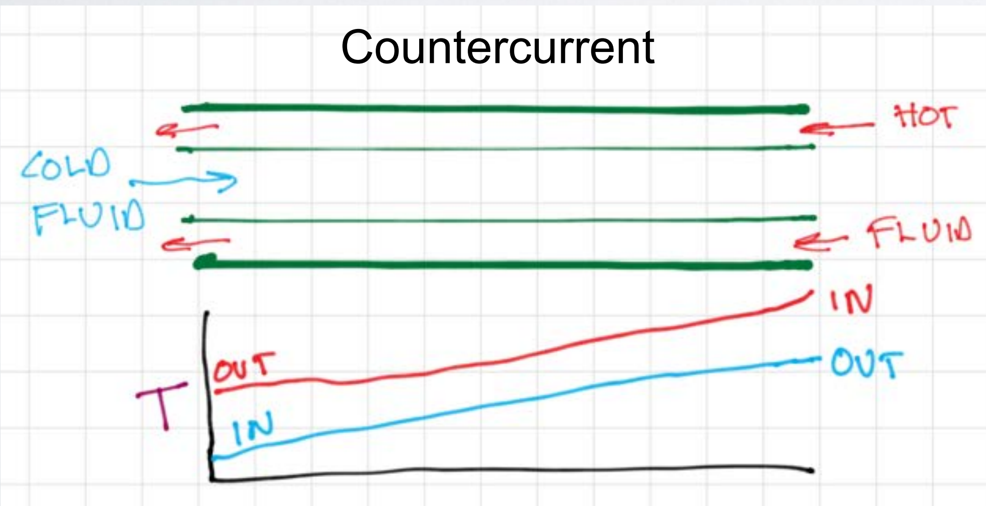
A U BASED ON THE OUTSIDE OF THE (INNER) PIPE IS:

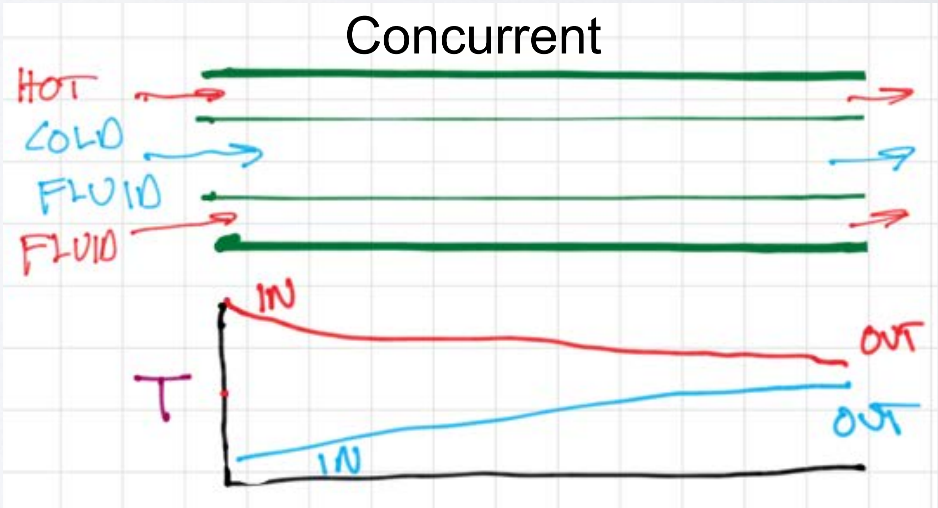
$$U_i \equiv \frac{q}{A_i \Delta T}$$

$$\frac{1}{U_o} = \frac{A_o}{A_i h_i} + \frac{A_o \ln d_o/d_i}{2 \pi L k} + \frac{1}{h_o}$$

HEAT TRANSFER CALCULATIONS

- With this short introduction to some fundamentals of heat transfer we now turn to example situations of interest.
- The device we will consider is a "double-pipe" heat exchanger in which one liquid is being heated by either a second liquid or condensing steam.
- The two fluids are not mixed.
- Countercurrent usually will provide a larger overall ΔT for a given length





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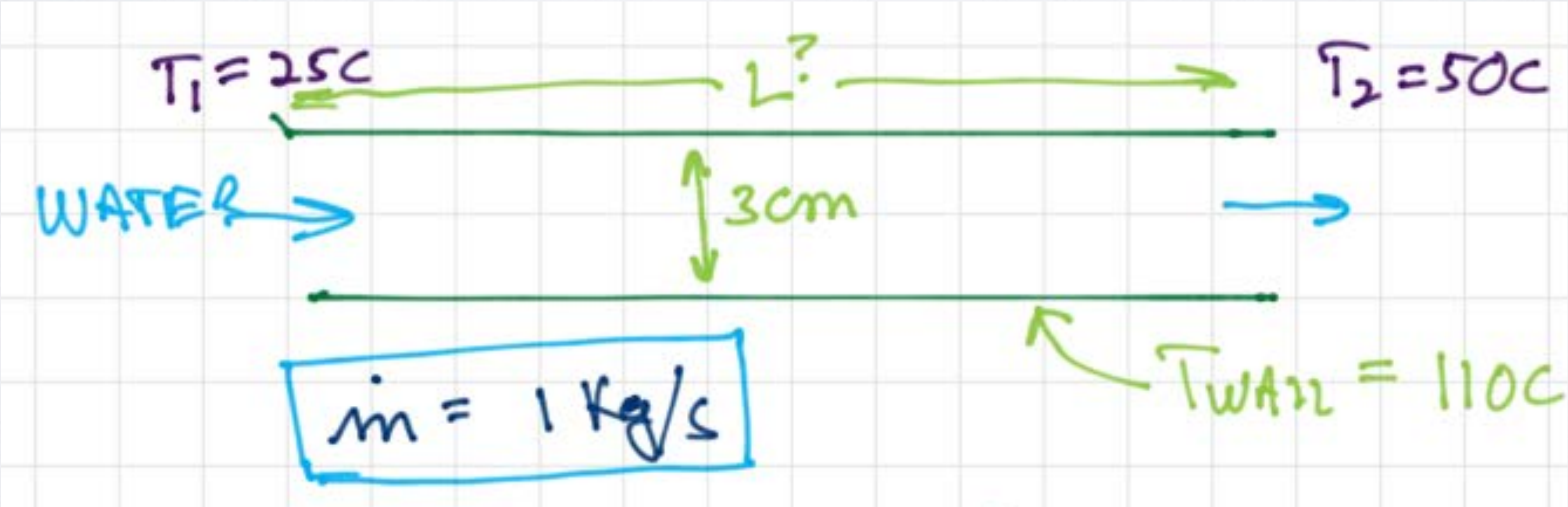
HEAT TRANSFER PROBLEM

- The 1 kg/s water stream is flowing in a 3 cm pipe in a countercurrent, double pipe heat exchanger. The water temperature must be raised from 25 C to 50C. Condensing steam, saturated at 110C will provide the heat.
    - What flow rate of steam is needed?
    - How long should the pipe be?
- The heat transfer coefficient for condensing steam will be much larger than for the water flow so we can assume that the wall temperature will be constant at 110C.



STEADY STATE HEATING OF A LIQUID FLOWING IN A PIPE: HEAT LOAD

$$0 = \dot{m}(\hat{H}_{IN} - \hat{H}_{OUT}) + q$$

$$q = \dot{m} c_p (T_{OUT} - T_{IN})$$

$$= (1\ \mathrm{Kg/s})\left(4180\ \frac{\mathrm{J}}{\mathrm{Kg\,K}}\right)(50-25)\ \mathrm{K}$$

$$= 104{,}500\ \mathrm{J/s}$$

$$q = 100\ \mathrm{KW}$$

STEAM FLOWRATE

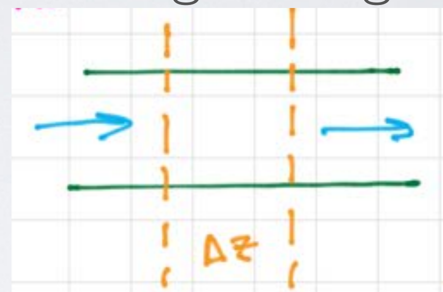
$$\hat{H}_{VAP} = 2691.3\ \frac{\mathrm{KJ}}{\mathrm{Kg}},\quad \hat{H}_{LIQ} = 461.3\ \frac{\mathrm{KJ}}{\mathrm{Kg}}$$

$$\dot{m}_{STEAM} = \frac{q}{(\hat{H}_{VAP} - \hat{H}_{LIQ})}$$

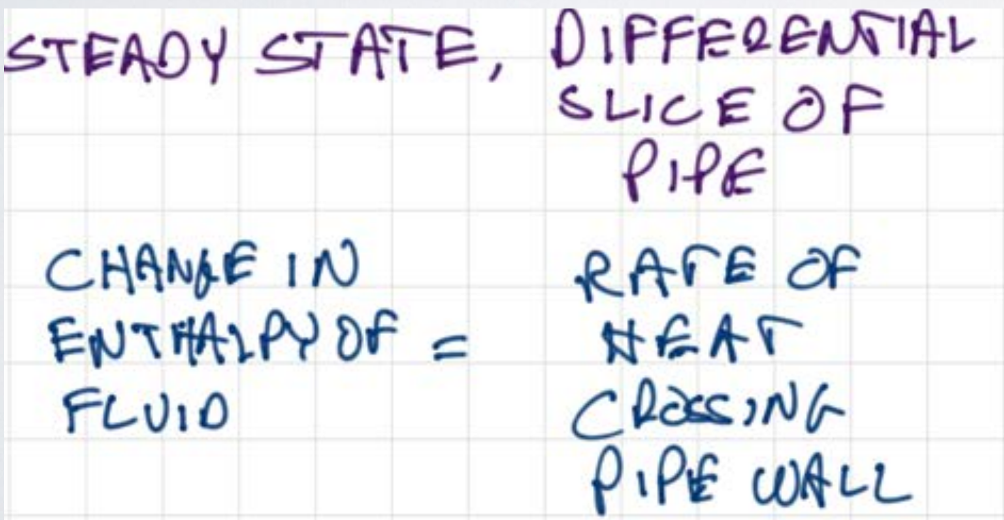
$$\dot{m}_{STEAM} = .05\ \mathrm{Kg/s}$$

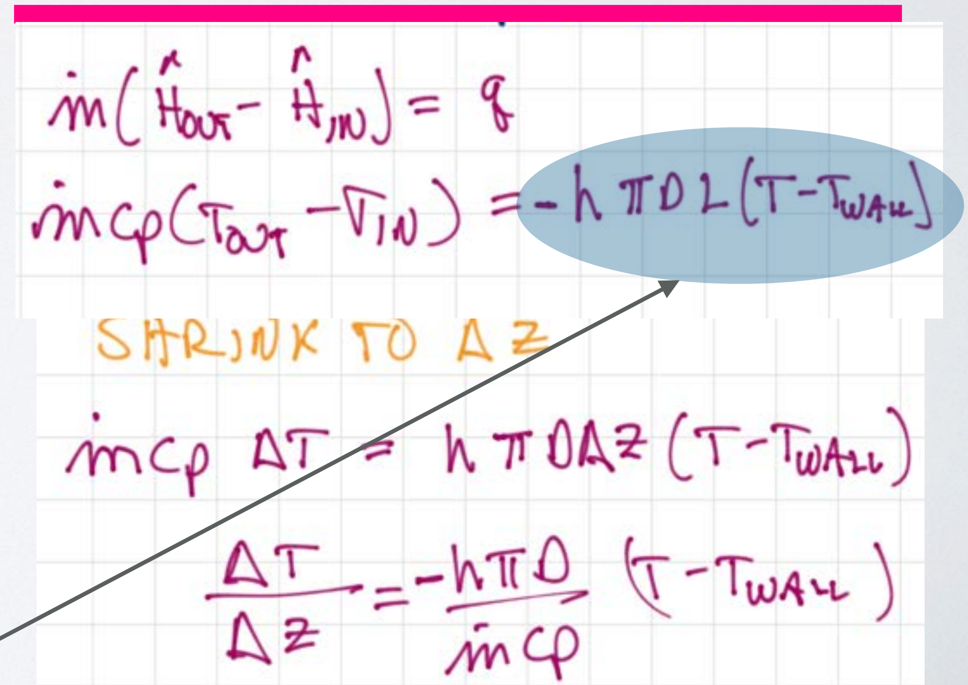
DIFFERENTIAL ANALYSIS TO GET LENGTH OF HEAT EXCHANGER

- The steam keeps the pipe wall at a constant temperature, but the temperature of the water in the pipe is changing. We will need a differential formulation of the temperature change along the pipe and then need to integrate to get the answer.



- Consider a differential slice of pipe.





Newton's Law of cooling

$\Delta Z \to 0$

$$\frac{dT}{dZ} = -\frac{h \pi D}{\dot{m} C_p}\left(T - T_{WALL}\right)$$

NOW INTEGRATE FROM T = 25C TO T = 50C

$$\int_{T=25}^{T=50} \frac{dT}{T - T_{WALL}} = -\frac{h \pi D}{\dot{m} C_p}\int_0^L dZ$$

$$\ln\left(\frac{T_{OUT} - T_{WALL}}{T_{IN} - T_{WALL}}\right) = -\frac{h \pi D}{\dot{m} C_p} L$$

Logarithmic mean temperature difference

- Log mean temperature difference arises from this differential analysis when both streams are changing temperature

$$LMTD = \frac{\Delta T_A - \Delta T_B}{\ln\left(\frac{\Delta T_A}{\Delta T_B}\right)} = \frac{\Delta T_A - \Delta T_B}{\ln \Delta T_A - \ln \Delta T_B}$$

$$Q = U \times Ar \times LMTD$$

CORRELATIONS FOR HEAT TRANSFER COEFFICIENT

- As with the "friction factor", we look for the appropriate correlation that uses the correct dimensionless groups.

- For heat transfer we need to find a value for the Nusselt number, *Nu*, in terms of the Reynolds number, *Re*, and the Prandtl number, *Pr*.

$$Nu \equiv \frac{hD}{k}$$

$$Pr \equiv \frac{\nu}{\alpha} = \frac{\mu/\rho}{k/\rho C_p} = \frac{\mu C_p}{k}$$

THE APPROPRIATE CORRELATION IS:

$$Nu = 0.023\, Re^{.8}\, Pr^{.4}$$

SOME CORRELATIONS FROM BRODKEY & HERSHEY

wall roughness conditions. The modern form [M3, S6] of the Dittus–Bo
correlation [D3], which is based on Eq. (11.65), is

$$N_{\mathrm{Nu,mb}} = \bar{h} d_{\mathrm{i}}/k_{\mathrm{mb}} = 0.023(N_{\mathrm{Re,mb}})^{0.8}(N_{\mathrm{Pr,mb}})^{n} \quad (11$$

$$0.7 \le N_{\mathrm{Pr,mb}} \le 100$$

$$10\,000 \le N_{\mathrm{Re,mb}} \le 120\,000$$

$$L/d_{\mathrm{i}} \ge 60 \quad \text{(smooth tubes)}$$

where n is 0.4 for heating ($T_{\mathrm{w}} > T_{\mathrm{b}}$) and 0.3 for cooling. Note that
conditions listed below Eq. (11.66) are the range of data used in

For large ΔT, another equation by Sieder and Tate [S4] is recommended:[1]

$$N_{\mathrm{Nu,mb}} = 0.027(N_{\mathrm{Re,mb}})^{0.8}(N_{\mathrm{Pr,mb}})^{1/3}(\mu_{\mathrm{mb}}/\mu_{\mathrm{w}})^{0.14} \quad (11.67)$$

$$0.7 < N_{\mathrm{Pr,mb}} \le 160$$

$$N_{\mathrm{Re,mb}} \ge 10\,000$$

$$L/d_{\mathrm{i}} \ge 60 \quad \text{(smooth tubes)}$$

Friend–Metzner analogy. The Friend–Metzner analogy uses an equation of substantially different form in order to correlate data over wide ranges of N_{Pr} and N_{Sc} [F3]. Their correlation for heat transfer is

$$N_{\mathrm{Nu,mb}} = \frac{N_{\mathrm{Re,mb}} N_{\mathrm{Pr,mb}} (f/2)(\mu_{\mathrm{mb}}/\mu_{\mathrm{w}})^{0.14}}{1.20 + (11.8)(f/2)^{1/2}(N_{\mathrm{Pr,mb}} - 1)(N_{\mathrm{Pr,mb}})^{-1/3}} \quad (11.83)$$

$$0.5 \le N_{\mathrm{Pr,mb}} \le 600 \qquad N_{\mathrm{Re,mb}} \ge 10\,000$$

USEFUL INFORMATION

TABLE 11.4
Approximate magnitudes of heat transfer coefficients*

	Range of values	
Application	***h*, W m^{-2} K^{-1}**	***h*, Btu ft^{-2} h^{-1} °F^{-1}**
Steam (dropwise condensation)	3×10^4–1×10^5	5×10^3–2×10^4
Steam (film-type condensation)	5×10^3–2×10^4	1×10^3–3×10^3
Boiling water	2×10^3–5×10^4	300–9×10^4
Condensing organic vapors	1×10^3–2×10^3	200–400
Water (heating)	300–2×10^4	50–3×10^3
Oils (heating or cooling)	60–2×10^3	10–300
Steam (superheating)	30–100	5–20
Air (heating or cooling)	1–60	0.2–10

* From McAdams, *Heat Transmission*, 3d ed., p. 5, McGraw-Hill, New York, 1954. By permission.

FOR OUR PROBLEM

- Check these numbers..
  - Re = 66000, Pr = 5 ==> Nu = 290
  - h = 6060 W/(m^2 K)
  - L = 2.6 m

CHECKS

$$Re = \frac{VDs}{\mu}$$

$$V = \frac{(1\,kg/s)}{(1000\,kg/m^3)}\frac{4}{\pi(.03m)^2}$$

$$= 1.4\,m/s$$

$$Re = \frac{(1.4)(.03)(1000)}{.0007} = 60630$$

$$Nu = .023\,(60630)^{.8}(5)^{.4}$$

$$= 293$$

$$h = \frac{293}{.03m}(.62\,W/mK)$$

$$= 6063\,W/(m^2K)$$

$$\therefore\ L = -\frac{(1\,kg/s)\left(\frac{4200\,J}{kg\,K}\right)}{\left(6060\frac{J}{s\,m^2K}\right)\pi(.03m)}\ln\left(\frac{50-110}{25-110}\right)$$

$$L = 2.6\,m$$

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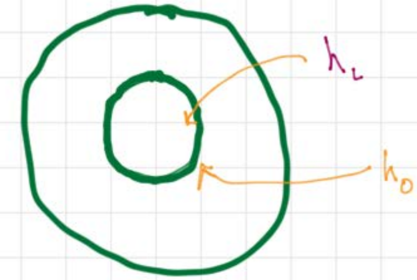
SUPPOSE THAT HOT WATER IS BEING USED TO HEAT THE COLD WATER

- In this case, both streams will be changing temperature

- We now will need to set up energy balances for the two streams and then integrate along the length of the heat exchanger

Cross-section



Heat transfer resistance occurs at both inside and outside of inner pipe, hence there is a heat transfer coefficient for each side.

Sum of resistances used to get an overall heat transfer coefficient, U

$$U_i \equiv \frac{q}{A_i \Delta T}$$

$$\frac{1}{A_i U_i} = \frac{1}{h_i A_i} + \frac{1}{k \Delta r} + \frac{1}{h_o A_o}$$

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{A_i}{k \Delta r} + \frac{A_i}{h_o A_i}$$

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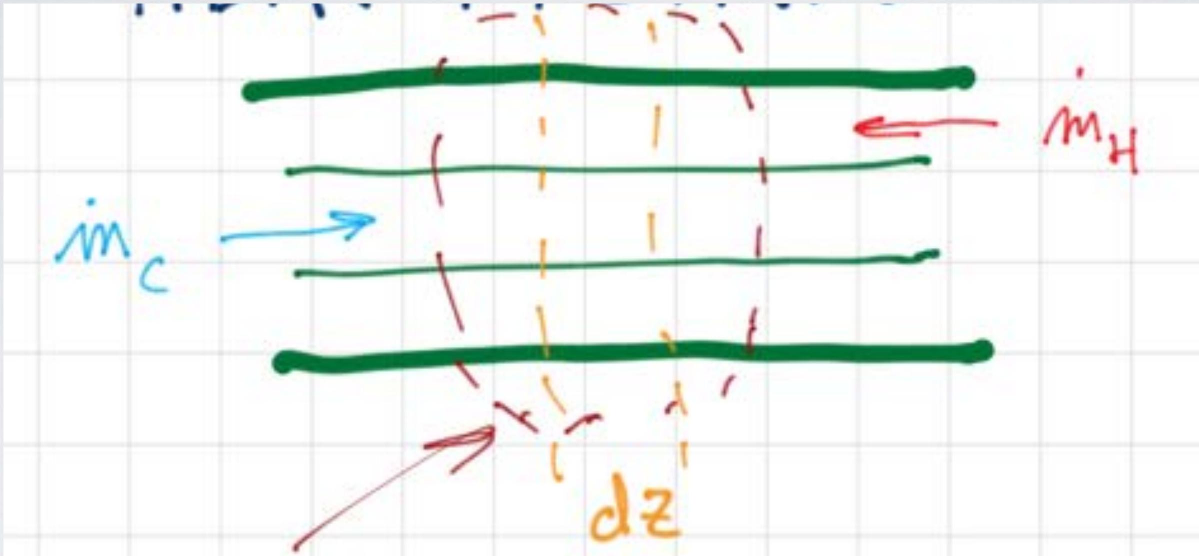
ANALYSIS OF DOUBLE PIPE HEAT EXCHANGER



OVERALL ENERGY BALANCE

$$0 = \dot{m}_C \, d\hat{H}_C + \dot{m}_H \, d\hat{H}_H$$

COLD STREAM

$$\dot{m}_C \, d\hat{H}_C = \dot{m}_C C_p dT_C = dq_C$$

HOT STREAM

$$\dot{m}_H \, d\hat{H}_H = \dot{m}_H C_p dT_H = dq_H$$

Since heat just leaves the hot stream and enters the cold stream, $dq_C = -dq_H$

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- Recall that d*q*, the rate of heat transfer into or out of a stream, is modeled with Newton's law of cooling,

$$dq = U_i (T_H - T_c)\, dA_i$$
$$= U_o (T_H - T_c)\, dA_o$$

USING THE TWO dT'S

$$dT_c = \frac{dq}{\dot{m}_c C_p}, \quad dT_H = \frac{-dq}{\dot{m}_H C_p}$$

$$d(T_H - T_c) = -dq\left(\frac{1}{\dot{m}_H C_p} + \frac{1}{\dot{m}_c C_p}\right)$$

$$\frac{d(T_H - T_c)}{(T_H - T_c)} = -\left(\frac{1}{\dot{m}_H C_p} + \frac{1}{\dot{m}_c C_p}\right) U_i\, dA_i$$

- Let's make a "ΔT"

$$d\Delta T \equiv d(T_H - T_c)$$

$$dA_i = \pi D_i \, dz$$

THUS WE CAN INTEGRATE

$$\int_{\Delta T(z=0)}^{\Delta T(z=L)} \frac{d\Delta T}{\Delta T} = -\frac{U_i \pi D_i}{\dot{m}_H c_p + \dot{m}_c c_p} \int_0^L dz$$

(ASSUMING THAT c_p, U_i = CONST)

$$\ln\left(\frac{\Delta T_L}{\Delta T_0}\right) = -\frac{U_i \pi D_i}{(\dot{m}_H c_p + \dot{m}_c c_p)} L$$

RECALL

$$q = \dot{m}_c c_p (T_{c,out} - T_{c,in})$$

$$q = -\dot{m}_H c_p (T_{H,out} - T_{H,in})$$

REPLACING THE $\dot{m} c_p$'S WE HAVE

$$\frac{q}{U_i A_i} = \frac{(T_{H,out} - T_{c,in}) - (T_{H,in} - T_{c,out})}{\ln\left(\frac{(T_{H,out} - T_{c,in})}{(T_{H,in} - T_{c,out})}\right)}$$

$$= \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$

$$= \Delta T_{LM}$$

LM = "LOG-MEAN"

ANOTHER WATER HEATING PROBLEM

- We wish to heat the 1 kg/s water from 25-50C with the inside pipe diameter the same 3 cm, inside a double pipe heat exchanger where instead of steam, a 75C water stream of 2 kg/s is available.

- For simplicity we will neglect the resistance of the pipe wall, which is 0.2cm thick. Also, we know that the outside heat transfer coefficient is 3000 W/(m^2 K).

- Consider both concurrent and countercurrent configurations.

FIRST WE NEED THE EXIT WATER TEMP

$$\dot{m} c_p \Delta T = q$$

$$\Delta T = \frac{100000 \, W}{(2 \, kg/s)(4180) \frac{J}{K \, kg}}$$

$$\Delta T = 12 \, K$$

WATER EXIT T IS 63C

LETS GET U_i

$$\frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o}$$

$$\frac{1}{U_i} = \frac{1}{6000} + \frac{.03}{3000(.032)}$$

$$U_i = 2100 \frac{W}{m^2 K}$$

$$A_i = \frac{q}{U_i \Delta T_{LM}}$$

COCURRENT:

$$= \frac{100000\ W}{2100\ \frac{W}{m^2 K} \left(\frac{(75-25)-(63-50)}{\ln \frac{(75-25)}{(63-50)}} \right)}$$

$\Delta T_{LM} = 27.5\ K$

$A_i = 1.7\ m^2$ $L = 19\ m$

A larger driving force for countercurrent gives a shorter length

COUNTERCURRENT

$$A_i = \frac{100000\ J/s}{2100\ \frac{W}{m^2 K} \left(\frac{(75-50)-(63-25)}{\ln \left(\frac{75-50}{63-25} \right)} \right)}$$

$A_i = 1.5\ m^2$ $L = 16\ m$

$\Delta T_{LM} = 31\ K$

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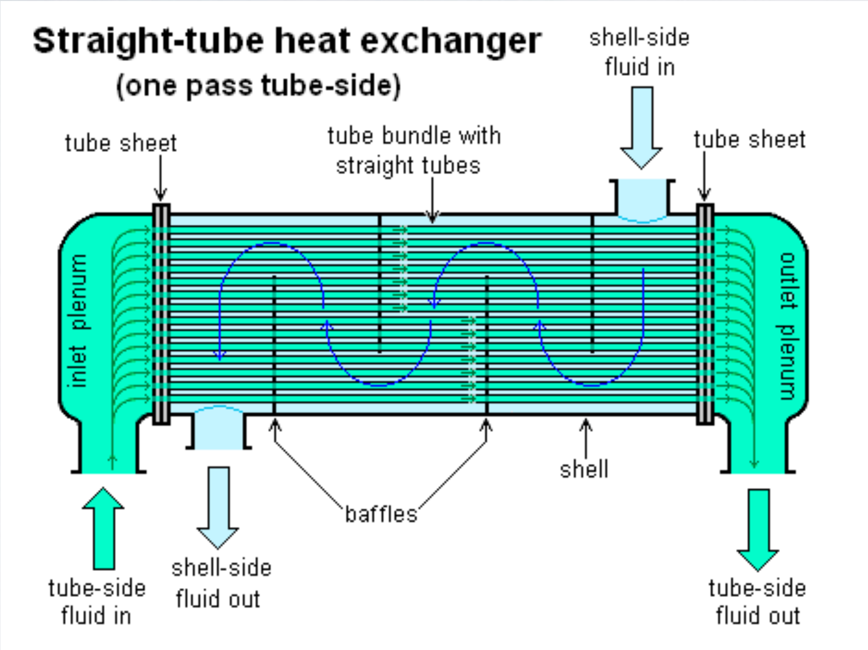
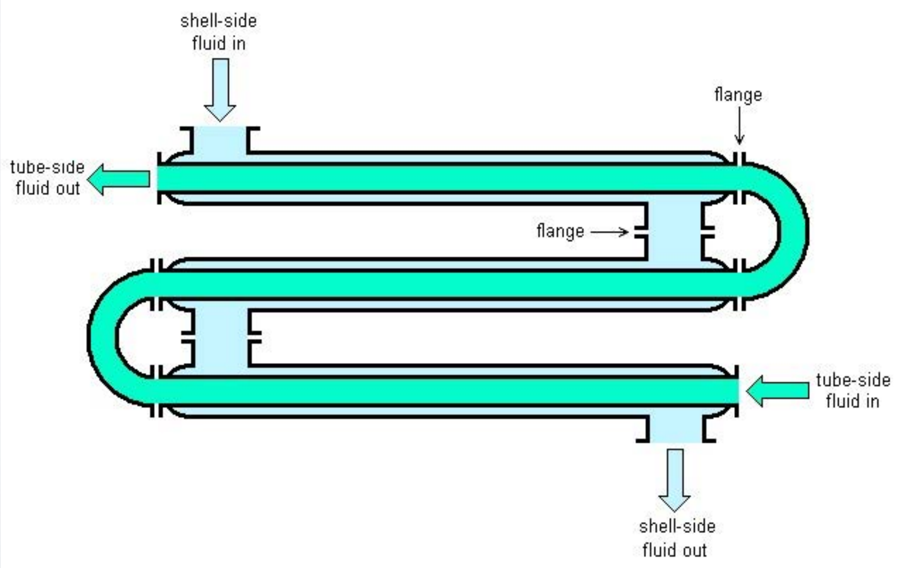
HEAT EXCHANGERS



- Shell and Tube
  - a combination of counter current and cross flow



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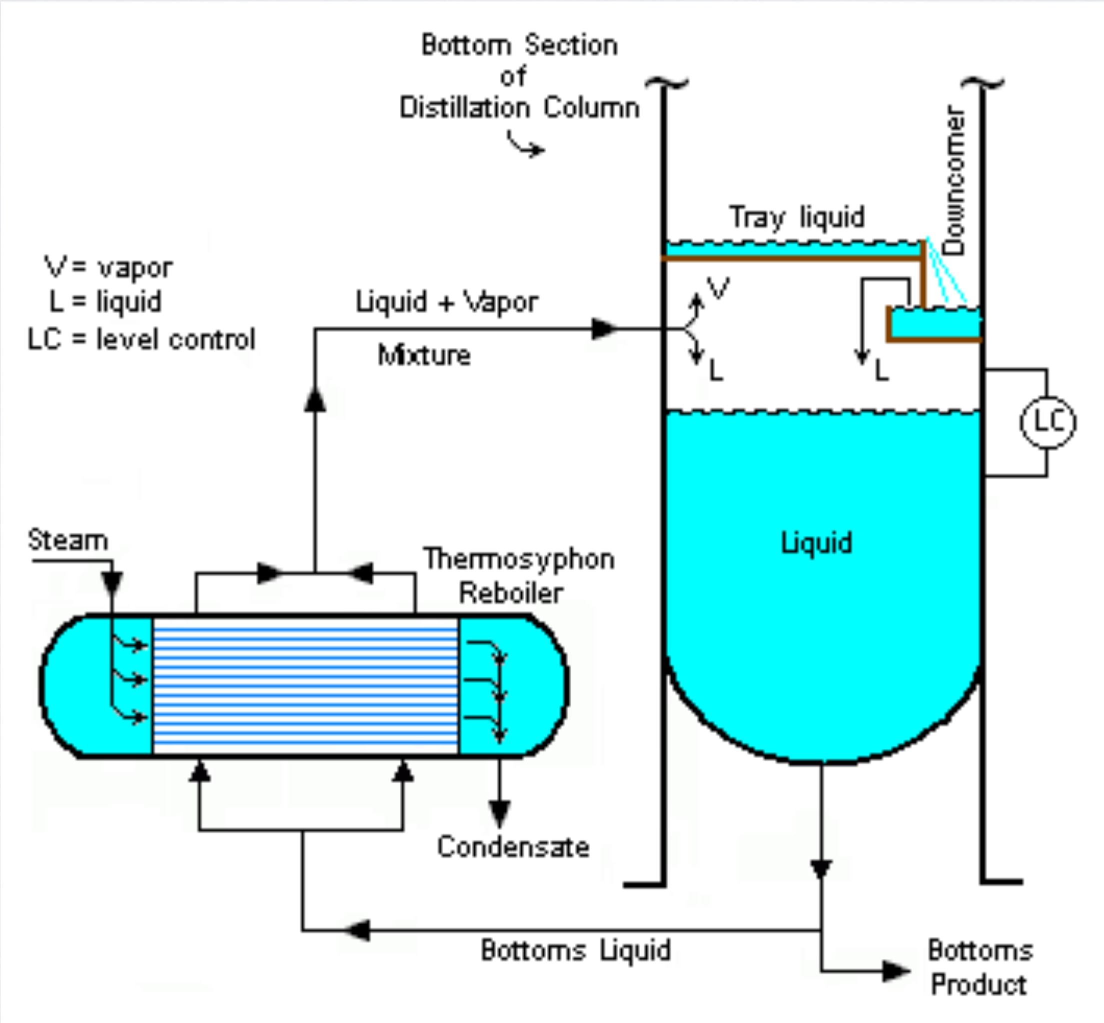
- Double pipe
  - true counter current

THERMOSIPHON



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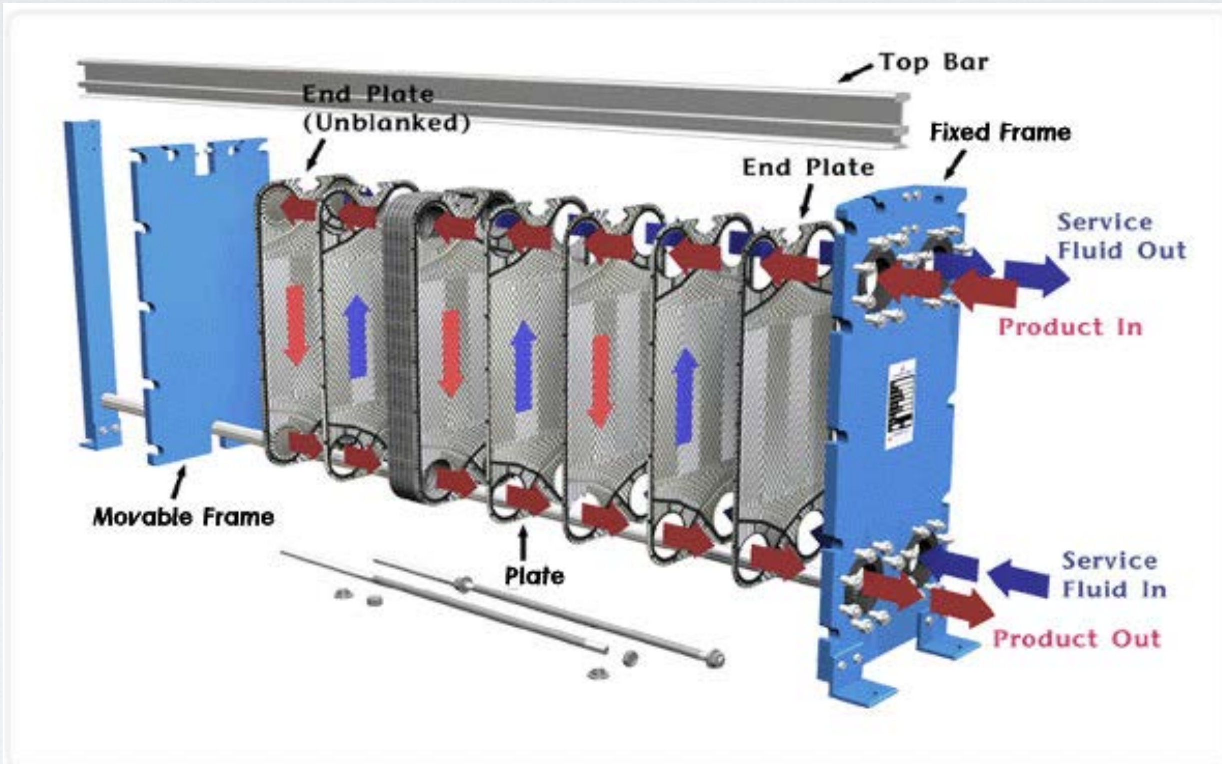
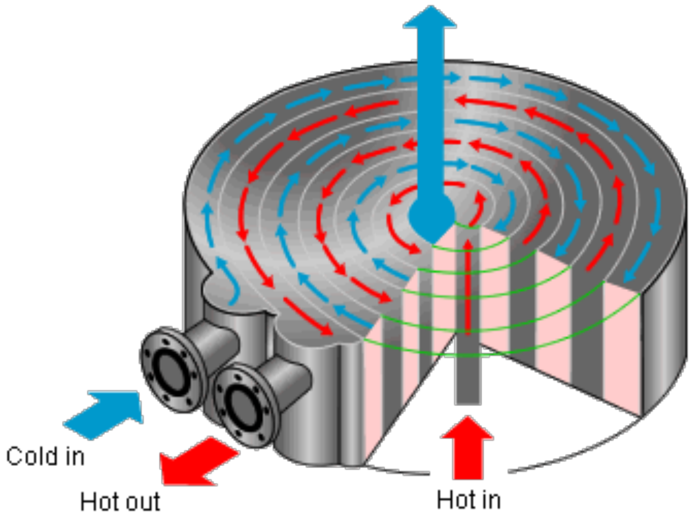
"PLATE" HEAT EXCHANGERS

- Air-Air



- More plates could be added

SPIRAL HEAT EXCHANGER



effectively a high surface area, “double-pipe” heat exchanger in a compact space

SUMMARY OF HEAT TRANSFER FUNDAMENTALS

- Three modes of heat transfer can occur:
  - Radiation (electromagnetic radiation) $q \sim \varepsilon \sigma (T^4 - T_0^4)$
  - Conduction (random motion of molecules, atoms and electrons) $q \sim k_A \frac{\Delta T_A}{\ell}$
  - Convection (heat transfer that is aided by bulk fluid motion)

HEAT EXCHANGER SUMMARY

- Heat exchangers are first analyzed using an energy balance $\dot{m}_c d\hat{H}_c = \dot{m}_c C_p dT_c = dq_c$
- The rate of transfer across the walls is modeled using Newton's Law of cooling $q = hA(T_w - T_\infty)$
- We get individual *h*'s from correlations $Nu = 0.023 Re^{0.8} Pr^{0.4}$
- We get a *U*'s from a sum of resistances $\frac{1}{U_i} = \frac{1}{h_i} + \frac{A_i}{k \Delta r} + \frac{A_i}{h_o A_i}$
- Because the temperature difference between the two sides of the heat exchanger is changing along the pipe, we formulate the problem as a differential slice of pipe and integrate. This gives the temperature driving forces as a "Log-Mean delta T"

$$\frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$