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Revision Memo

- 12/2002** This is not considered a full DP Update. Change only addresses an EMESOC (00-02) follow-up item. Added a requirement to designate exchangers in cyclic service. This specification will assist in the selection of a suitable gasket type for the girth flange joints.
- 12/99** Highlights of this revision are:
1. Replaced Pinch Technology Section with Heat Integration Using Pinch Technology.
 2. Renumbered the tables and figures.



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SCOPE

This section presents basic design considerations for all types of heat exchange equipment.

REFERENCES

DESIGN PRACTICES

Other sections of Section IX

Section II	Design Temperature, Design Pressure and Flange Rating
Section XII	Instrumentation
Section XIV	Fluid Flow
Section XV	Safety in Plant Design
Section XXII	Storage Facilities

GLOBAL PRACTICES

GP 3-2-4,	Pressure Relieving Systems
GP 3-6-4,	Vents and Drains, Flushing and Cleaning Connections
GP 6-1-1,	TEMA Type Shell and Tube Heat Exchangers
GP 6-2-1,	Double Pipe Heat Exchangers
GP 6-3-1,	Brazed Aluminum Heat Exchangers
GP 6-4-1,	Air-Cooled Heat Exchangers
GP 18-4-1,	Postweld Heat Treatment of Equipment Handling Alkaline Solutions

OTHER COMPANY LITERATURE

Refinery Construction Materials Manual, EETD028

Chemical Cleaning Manual, ER&E Report No. EE.40ER.68, November 1968.

Chemical and Mechanical Cleaning Manual, EETD 052.

Fouling in Heat Exchangers: New Developments, ER&E Report No. EE.92E.89, December 1989.

Update to Fouling in Heat Exchangers: Incentives for Applying Heat Exchanger On-line Mechanical Cleaning Devices, ER&E Report No. EE.117E.90, December 1990.

OTHER REFERENCES

API Bulletin 25 - 16 (low-pressure tanks)

API Bulletin 25 - 17 (floating roof tanks)

API Bulletin 25 - 18 (cone-roof tanks)

API Bulletin 11k (Data Sheet for the Design of Air Exchange Coolers)

API Standard 660, Shell-and-Tube Heat Exchangers for General Refinery Services

API Standard 661, Air-Cooled Heat Exchangers for Refinery Services

API Standard 662, Plate Heat Exchangers for General Refinery Services

Standards for the Tubular Exchanger Manufacturer's Association (TEMA), 7th Ed. 1988



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GENERAL

Figure 1 outlines the logic for all types of heat exchange equipment design. It is an iterative procedure involving thermodynamic, thermal, mechanical and economic considerations. The purpose of the following section is to introduce the designer to the analytic methods used in the thermal design as well as some mechanical aspects of heat exchange equipment design.

In addition, the Energy Management System (EMS) has Design Leading Practices on instrumentation requirements and cleaning considerations for the energy critical exchangers. Refer to the EMS leading practices books for these practices and for a procedure to identify energy critical exchangers.

The rate of heat transfer from one fluid to another across a heat transfer surface is the product of the overall heat transfer coefficient, the area of the surface, and the effective temperature difference between the hot and cold fluids:

$$Q = U_o A MTD_e \quad \text{Eq. (1)}$$

where: Q = Rate of heat transfer, Btu/h (W)
 U_o = Overall heat transfer coefficient based on surface area A , Btu/h • ft² • °F (W/m² • °C)
 A = Heat transfer surface area across which heat is exchanged, ft² (m²)
 MTD_e = Effective mean temperature difference between hot and cold fluids, °F (°C)

When specifying a heat exchanger, the designer nearly always knows or can readily calculate the Q and MTD_e terms from the process conditions. It is necessary only to evaluate the actual overall heat transfer coefficient U_o in order to find the required heat transfer area. Unfortunately, U_o is a function of the actual exchanger design as well as of the fluid properties and fouling rates. For this reason, the design of a heat exchanger is an iterative procedure.

The general procedure used in heat exchanger design is as follows:

1. Establish Q from process considerations.
2. Establish MTD_e from process considerations. Exchanger type and tube arrangement will have some effect on MTD_e as explained in the section titled **EFFECTIVE TEMPERATURE DIFFERENCE**.
3. Assume U_o the actual overall heat transfer coefficient (see Table 1 for typical values).
4. Calculate an assumed A from the assumed U_o using Eq. (1).
5. Determine the physical dimensions of the applicable type of heat exchanger from the calculated A .
6. Calculate the fluid pressure drops (described later in this section) and velocities through the exchanger and modify the internals, if required, to obtain a reasonable balance between pressure drop, velocities, and exchanger size.
7. Calculate U_o (see below) from the physical properties of the fluids, fouling factors and the exchanger geometry.
8. Calculate A based on Q and the calculated values of U_o and MTD_e using Eq. (1).
9. Compare A calculated with A assumed and repeat the calculations until they are approximately equal. For almost any value of U_o there is an exchanger design that satisfies the criterion that A calculated equals A assumed. However, only a few of these designs are reasonable.

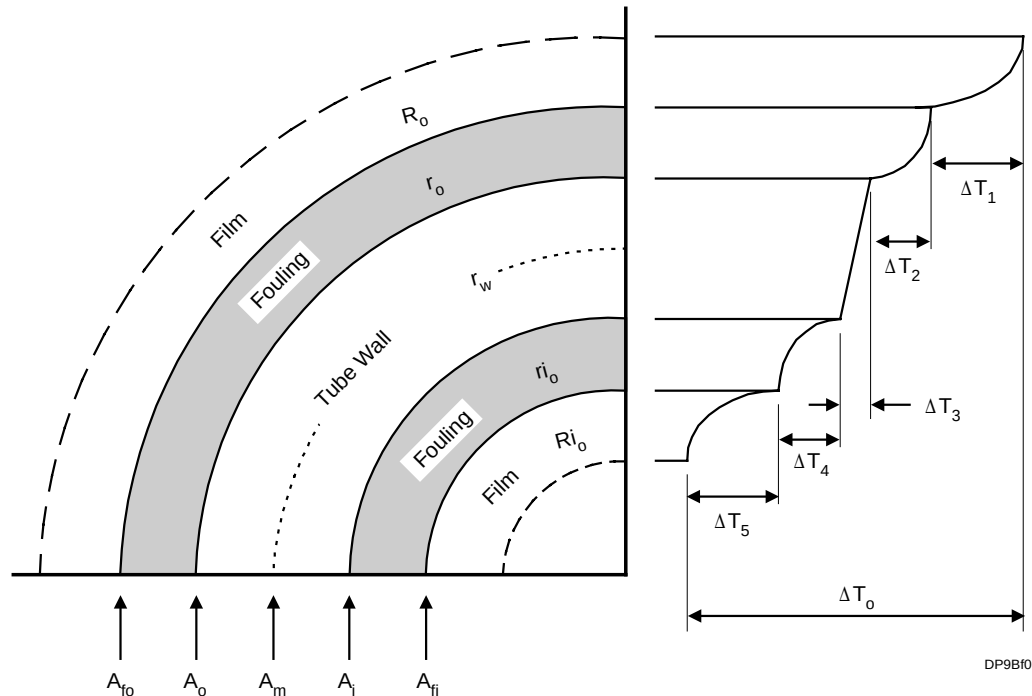
ACTUAL OVERALL HEAT TRANSFER COEFFICIENT - U_o (FOR ALL BUT DIRECT-CONTACT HEAT EXCHANGERS)

When heat flows from a fluid on one side of a surface to a fluid on the other side of the surface, it must overcome the following thermal resistances:

- R_{i0} is the resistance to the convection of heat on the inside of the surface, referenced to the outside surface area. For a shell and tube heat exchanger, R_{i0} is called the tube side film resistance referenced to the outside surface area of the tube.
- r_{i0} is the resistance (fouling factor) of foreign material deposited on the inside of the surface, referenced to the outside surface area.
- r_w is the resistance due to the conduction of heat through the barrier separating the two fluids.
- r_o is the resistance (fouling factor) of foreign material deposited on the outside of the surface.
- R_o is the resistance to the convection of heat on the outside of the surface. For a shell and tube heat exchanger, R_o is called the shell side film resistance.

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ACTUAL OVERALL HEAT TRANSFER COEFFICIENT - U_o (Cont)
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The sum of these five resistances is R_t , the total resistance; and:

$$U_o = \frac{1}{R_t} \quad \text{Eq. (2)}$$

The fouling factors r_{io} and r_o are estimated based on experience or taken from typical values listed in [Tables 3-5](#). The term r_w is calculated from the thickness and thermal conductivity of the wall. R_{io} and R_o are functions of mass velocity and physical properties of the fluid; they are evaluated from the applicable correlations given in the following sections. These correlations are in terms of h_{io} and h_o , where $1/R_o = h_o$, and $1/R_{io} = h_{io}$. The "h" terms are the film coefficients. (The symbol ∞ has been used recently to designate "film coefficients" in S.I. - based publications.)

The resistance terms contain an area dimension (i.e., sq ft) which usually refers to the surface area at which the resistance occurs. Since the resistance terms must be added to obtain the overall resistance, the area dimensions of each term must refer to the same surface area rather than to its own area. This rationalizes the terms and makes them additive. For shell and tube heat exchangers, standard practice is to use the outside tube area (A_o) as the basis for calculation and specification of exchangers. As shown above, the usual nomenclature indicates this by the subscript "io." For example, " h_{io} " is the "inside coefficient" based on the outside tube area. For a tube or pipe, $h_{io} = h_i (d_i / d_o)$, where h_i is the "inside coefficient" based on the inside tube area. This factor is already included in the correlations present in this manual.

Two overall coefficients should be calculated: a commercially clean coefficient and an actual overall heat transfer coefficient (described above). The commercially clean coefficient is the overall coefficient that can be expected when a new exchanger is first placed into service, before process fouling of the tube occurs. It is calculated according to the following:

$$U_c = \frac{1}{R_c} = \frac{1}{R_{io} + R_o + r_w} \quad \text{Eq. (3a)}$$

Alternatively, the clean coefficient may be calculated by subtracting the fouling resistances from the total resistance:

$$\frac{1}{U_c} = \frac{1}{U_o} - r_{io} - r_o \quad \text{Eq. (3b)}$$

Heat transfer correlations used in ER&E computer programs are based on data taken from commercially clean equipment.

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OPERATING TEMPERATURES

The operating temperatures of the exchanger are usually set by process conditions. However, in certain cases, the exchanger designer will establish the operating temperatures. In a typical refinery or petrochemical plant, exchangers may be operating at temperatures as high as 1000°F (540°C) or as low as -200°F (-130°C). These limits are dictated by material considerations, safety, economics and ASME Code requirements.

TEMPERATURE OF STREAMS TO STORAGE

The maximum (or sometimes minimum) temperature of a stream going to atmospheric storage generally is set by safety, economics or special process conditions. See [Section XV, Safety in Plant Design](#). See also [Offsite DP Section XXII, Storage Facilities](#).

Safety Considerations - The temperature of a stream going to an atmospheric tank at sea level should not exceed that temperature at which its true vapor pressure is equal to the values listed below.

TANK TYPE	TRUE VAPOR PRESSURE	
	(psia)	(kPa abs)
Cone Roof (without vapor recovery)	1.5	10.5
Floating Roof	13.0*	90.0*
Hemispheric	20.0	138.0

Note:

* For U.S. application, use 11.0 psia (76.0 kPa) to meet EPA regulation.

This value is reduced 0.5 psia for each 1000 ft elevation (11.3 kPa abs for each 1000 m elevation).

For heavy streams whose true vapor pressure is difficult to determine, the maximum temperature to tankage should be the lower value of either 50°F (28°C) below the ASTM initial boiling point or 15°F (8°C) below the minimum flash point.

Streams should not be sent to tankage at temperatures above 200°F (90°C) for API 650 tanks, and below 250°F (120°C) for hot tanks. Operation in the 200 to 250°F (90 to 120°C) temperature range could result in condensation when the tank cools forming a water heel which could subsequently flash to steam, resulting in a boilover.

Economics - The selection of the optimum temperature of a stream going to cone-roof tankage is generally based on an economic balance between the cost of incremental cooler surface and cooling water and the savings due to reduced vapor losses. The method of determining vapor losses is given in the following API Bulletins:

- API Bulletin 25 - 16 (low pressure tanks)
- API Bulletin 25 - 17 (floating roof tanks)
- API Bulletin 25 - 18 (cone roof tanks)
- API Bulletin 25 - 19 (internal floating roof tanks)

Since the losses from a floating roof tank can be negligible, the optimum temperature is sometimes the maximum allowed by safety considerations (consult the **EETD Safety, Risk and Site Inspection Section**).

Special Considerations - Opportunity for optimizing the temperature of a stream going to storage is greatest for intermediate products. However, special considerations are required for the following cases.

1. Streams that are stored prior to a process requiring refrigeration of the feed.
2. Streams whose properties are permanently degraded by high storage temperatures.
3. Streams that are stored before blending. The storage temperatures for such streams should be chosen after considering the properties and temperature of the blend, assuming no heat loss in intermediate tankage.

In addition to the limitations given above, the temperature of a stream going to final storage is normally limited to a maximum of 135°F (57°C) if the product is to be shipped by barge or tanker. However, temperatures up to 150°F (66°C) are sometimes used, provided that the high-temperature operating procedure is approved by shipping personnel.

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COOLING WATER TEMPERATURES

The maximum allowable cooling water outlet bulk (or "mixing-cup") temperatures, set by fouling considerations, for coolers other than box coolers (or worm coolers) are:

- Salt water: 120°F (48°C)
- Brackish water: 125°F (51°C)
- Fresh water: 130°F (54°C)

The maximum temperature used for a project should be checked with the affiliate, since this has an important bearing on materials selection.

Another important criterion is the maximum allowable cooling water outlet film temperature. The maximum film temperature is the water temperature at the liquid-metal interface occurring at the hot end of the exchanger. A conservative estimate of this film temperature is the tube metal temperature at the hot end of the exchanger. These limits are:

- Salt water: 140°F (60°C)
- Fresh water: 150°F (65°C)

Note: For box coolers, the maximum cooling water outlet temperature is 150°F (65°C) for both salt water and fresh water.

If water film temperatures are allowed to exceed these values, severe fouling and corrosion could result.

In those cases where the hot stream outlet temperature is equal to or lower than the maximum allowable cooling water outlet temperature, a brief economic study may be required to set the optimum cooling water outlet temperature. The study would compare material versus cooling water requirements for varying water outlet temperatures. However, the cooling water outlet temperature is normally determined by designing the exchanger so that the log mean temperature difference correction factor (F_n , discussed in the section titled **EFFECTIVE TEMPERATURE DIFFERENCE**) is equal to the minimum allowable value (0.8). Keep in mind the possibility of using a two-shell pass unit or two shells in a series.

At times, a condenser or cooler might be designed that uses a large amount of cooling water with a relatively low outlet temperature. Consideration should be given to reuse this water in other coolers where the water outlet temperature may be the maximum permissible. Box coolers always operate with reused water.

A list of recommended cooling water inlet temperatures at various refinery and chemical plant locations is given in [Table 2](#).

EFFECTIVE TEMPERATURE DIFFERENCE

The driving force for heat transfer is the "effective temperature difference," MTD_e , between the hot and cold fluids. This temperature difference is calculated from the counter-current log mean temperature difference with a correction factor applied to account for the actual flow arrangement.

For exchangers where the flow of hot and cold fluids is truly counter-current or co-current, MTD_e is equal to the log mean temperature difference, LMTD. A chart for calculating LMTD appears in [Section IX-D](#). Almost all exchanger designs attempt to achieve counter-current flow since this flow arrangement has the highest obtainable value of MTD_e for any given set of fluid temperatures.

In most commercial exchangers, however, using multiple tube passes or shell baffles causes the fluid to flow partially counter-current and partially co-current. Since co-current flow results in lower LMTDs, a correction factor must be applied to the counter-current LMTD. This correction factor, called F_n , is a function of the particular exchanger design, the number of units in series and stream temperatures.

$$MTD_e = F_n (LMTD) \quad \text{Eq. (4)}$$

NO CHANGE OF PHASE

Values for F_n for shell and tube exchangers can be obtained from [Figures 2A through 2F of Section IX-D](#). On these charts, the number of shell passes refers to the number of single-pass shells or the total number of shell passes connected in series. These charts are valid only when Q is a linear (or approximately linear) function of temperature.

The correction factor F_n for other types of exchangers may differ from that for shell and tube units. Methods for obtaining F_n are given in the appropriate sections.

CHANGE OF PHASE

In the case of condensation or vaporization, the relationship between Q and the temperature of the fluid usually deviates appreciably from a straight line. It is necessary to divide the exchanger into zones so that Q is approximately linear within the

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temperature for each zone. Based on the logarithmic mean temperature differences and heat duties for the individual zones, a weighted MTD_e is obtained for the entire exchanger. Methods for obtaining a weighted MTD_e are discussed in the sections on change of phase, [Sections IX-E and F](#).

TEMPERATURE APPROACH

Temperature approach is the difference of the hotside and coldside fluid temperatures at any point within a given exchanger.

In a counter-current flow heat exchanger, it is possible for the exit temperature of the cold fluid to be higher than the exit temperature of the hot fluid. This condition is known as a temperature cross and is illustrated in [Figures 2A through 2C](#). Three cases are depicted:

- [Figure 2A](#): Temperature distribution in a purely co-current flow heat exchanger (i.e., double pipe exchanger) with no temperature cross.
- [Figure 2B](#): Temperature distribution in a double pipe exchanger with a temperature cross.
- [Figure 2C](#): Temperature distribution in a shell and tube exchanger with a single shell-pass and two tube-passes with a temperature cross.

A temperature cross indicates a negative driving force for heat transfer between the fluids. It requires either a large area for heat transfer or high fluid velocities to increase the overall heat transfer coefficient.

If outlet temperatures form a cross in a multi-tubepass heat exchanger, a lower than desirable LMTD correction factor (F_n -factor) will occur. A simple way to avoid this is to use more exchanger shells in series. To determine how many shells to use in series, refer to [Figure 2A through 2F in Section IX-D](#), with $F_{min} = 0.8$ criterion, or use the graphical method illustrated in [Figure 2D](#) of this section. An exchanger design with an $F_n < 0.8$ will have outlet temperatures that cross. These exchangers require a large increase in surface area for a small increase in heat duty.

Although there is no definite value of minimum F_n -factor for design, an F_n -factor less than 0.75 should not be used. An engineering judgement must be made based on economic and technical factors. Adding more shells in series will increase cost significantly, and the designer may choose to use a larger-size single unit with a slight temperature cross. A general guideline for minimum F_n -factor, especially when designing new exchangers, is to use 0.8 which is a widely accepted engineering practice.

For an economical design of exchangers, several factors should also be considered such as incremental area and shell costs, overall heat transfer coefficient, fluid heat capacities, inlet temperature difference and the savings/investment ratio, etc. An analysis based on these factors to determine temperature approach and number of shells is presented in Exxon Technical Report EE.6E.83, *Incremental Heat Flux Method for Heat Exchanger Optimization* dated January 1983 by Ghamarian, Thomas and Sideropoulos.

In some cases (existing exchangers), a small temperature cross may be tolerated [up to 5°F (-15°C)], as shown in [Figure 2C](#). However, for all new designs a temperature cross should be avoided if possible. A temperature cross might occur for an oversurfaced or cleaned exchanger service since the outlet temperature will probably exceed the design operating temperatures where there is no temperature control. The sharp drop of the F_n -factor (and of course the effective MTD), due to closer temperature approach, will offset the extra surface of the exchanger.

Although the co-current flow arrangement ([Figure 2A](#)) does not allow any temperature cross and results in a lower MTD_e , it has some unique characteristics:

- Co-current flow produces a more uniform tube wall temperature.
- The highest tube wall temperature is comparatively lower than that for counter-current flow.
- Since the cold end of the hot stream meets the warmer end of the coolant medium, co-current flow is often used to avoid a freezing problem of the warmer fluid (e.g., airfin coolers in pitch service).
- Since the MTD is higher at the entrance region, it will initiate an earlier nucleate boiling in vaporizing service. The possibility of excessive heat flux must be explored in this case (vapor blanketing).

The minimum temperature difference between the fluids is the minimum temperature approach or temperature pinch. Some general rules apply:

- Typical values of temperature approach are shown in the table below.
- Excessive MTD [$> 110^\circ\text{F}$ (60°C)] should be avoided in heating and boiling systems.
- The temperatures listed for recuperative exchangers are minimum sustained temperatures.

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The following table lists the minimum recommended temperature approach for heat exchanger equipment:

HEAT EXCHANGER TYPE	MINIMUM TEMPERATURE APPROACH	
	°F	°C
Shell and Tube	15	9
Double Pipe	10	6
Air-cooled	25	14
Plate-fin, Plate Frame, Spiral	5	3
Recuperative Exchangers	36	20

Note: Examples of recuperative exchangers are exhaust-gas/air preheaters.

A more rigorous method exists to determine the minimum approach temperature. This technology is described in more detail in the following section titled Pinch Technology.

HEAT INTEGRATION USING PINCH TECHNOLOGY

This section provides the procedures to evaluate heat integration and guide development of more efficient systems, both within a given unit and across an entire site using pinch technology. Pinch technology is based on fundamental thermodynamic principles. This section also provides a procedure to apply pinch technology to optimize heat integration and process-to-utility heat exchange on capital projects.

APPLICABILITY

While heat recovery analysis using pinch technology can be completed for any process unit, Energy Management System (EMS) rollouts will use one or more of the following criteria to select process units for analysis:

- Total thermal energy consumption is \$3 M/yr or greater
- There are margin incentives due to heat input or recovery constraints
- There is close proximity to a unit with a potential of inter-unit heat integration

The capital element of EMS provides guidance on when it is relevant to apply this technology to capital projects. In this case, a heat integration analysis should be performed as soon as the heat and material balance is available. The capital project could be

- A grassroots unit
- Revamp with significant change in heat and material balance (e.g., when de-bottlenecking or when installing or modifying heat transfer networks)

Site-wide steam supply/demand profiling can be used in any of the following situations:

- When a site-wide energy survey/analysis is performed. For example, during an EMS site integration process (SIP) analysis
- When an existing process unit with significant interaction with utilities undergoes major revamp
- When a new process unit with significant interaction with utilities is being installed at a site
- When the utility system undergoes a major revamp.

The Heat Transfer and Energy Section of ER&E should be contacted for the following advanced concepts not covered here:

- Steps to configure a grassroots heat exchanger network to meet energy targets
- Methods to modify the existing heat exchanger network to improve the energy utilization efficiency

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HEAT RECOVERY ANALYSIS FOR A PROCESS UNIT

The heat recovery analysis on a process unit would calculate the ideal heat recovery target, compare it with the current usage to determine the gap, identify specific inefficiencies, and develop alternatives to close the gap. The identified modifications are checked to ensure the technical and economical feasibility. The procedure given below of applying heat recovery analysis using pinch technology on individual process unit is used in conjunction with [Figure 3](#).

Collect accurate and consistent **data** to perform a successful process heat integration analysis using pinch technology. The data required for process units are

- Process Flow Diagram
- Prices for utilities and exchangers
- Furnaces and Boilers
- Process-process and process-utility exchangers
 - Temperatures in and out of each exchanger
 - Exchanger duty
 - Existing exchanger area

The source of data could be

- Plant Measurement
- Simulation (HEXTRAN, PRO II or equivalent)
- Heat & Material Balance

The templates in [Tables 6 - 9](#) can be used for organizing data collected from the plant.

Perform the following steps to calculate the **heat recovery targets**:

1. Separate all streams requiring heating ("cold" streams) and all streams requiring cooling ("hot" streams).
2. Calculate composite cold and hot curves, representing all the heat required and available at each temperature level using the temperature - enthalpy data collected.
3. Select the appropriate minimum temperature difference, ΔT_{min} . A list of experience based ΔT_{min} for retrofit designs are provided in [Table 10](#). The ΔT_{min} value can be optimized for grassroots designs using energy to capital trade off. For details, refer to the Pinch Technology course or contact the Heat Transfer and Energy Section of ER&E.
4. Move the two composite curves horizontally until ΔT_{min} is reached at one point, known as the "pinch point", as presented in [Figure 4](#).
5. Calculate maximum feasible heat recovery and minimum external heating and cooling requirements from the composite T - H graph at a given ΔT_{min} , shown in [Figure 4](#). This establishes the targets, thermodynamically optimal utility requirement.

HEXTRAN or other pinch technology software can be used to perform these steps. Contact the Heat Transfer and Energy Section of ER&E for current recommendations on tools.

Synthesize the heat exchanger network to meet the targets for first of a kind grassroots project. Details of the synthesis method are outside the scope of this design practice. Contact the Heat Transfer and Energy Section of ER&E for assistance. A grassroots project based on a previous heat exchanger network design can use that design as the starting point and follow the retrofit work process.

Determine the energy utilization gap as the difference between the targets and actual usage. This is the opportunity to improve heat recovery in the process unit. The design is satisfactory if the gap is less than 15 MBtu/hr (4 MW). No further analysis is required. In case of larger gap, proceed to next step below. The one exception is for the analysis of a grassroots unit based on a previous design. In this case the satisfactory gap is 5 MBtu/hr (1.5 MW) or 2% of the total heat recovery, whichever is larger.

Identify the inefficiencies and non-countercurrents in the heat exchanger network using ESCORT or heat-across-the-pinch analysis. Consider an exchanger for improvement if the inefficiency or heat across the pinch in the exchanger is greater than 5 MBtu/hr (1.5 MW) for retrofits and 2 MBtu/hr (0.6 MW) for a grassroots unit with an existing design.

Identify modifications to the heat exchanger network that would eliminate or reduce the inefficiencies, non-countercurrents, and heat across the pinch. If major modifications are needed to a grassroots process unit heat exchanger network then synthesize a new network to meet the target. Contact Heat Transfer and Energy Section of ER&E for assistance with modifying the existing or synthesizing new heat exchanger network.

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Check each modification for process constraints such as stream pressure drop and pumparound duties, temperatures, and flows and for mechanical constraints such as available pump head, design pressures and temperatures, and flange rating. All the process constraints should be satisfied. Evaluate the economics for technically feasible modifications.

SITE-WIDE STEAM SUPPLY/DEMAND PROFILING

The site-wide steam supply/demand profiling, known as total-site analysis in the literature, identifies opportunities to increase/decrease steam usage at each level, to increase/decrease steam pressures/levels and to implement heat integration between different units. The site-wide steam supply/demand profiling procedure given below is used in conjunction with [Figure 5](#)

Select the units to be included in the site-wide steam supply/demand profiling.

Collect data (duty, process side T_{in} and T_{out} , utility level, and area) for utility exchangers in these units. Utility exchanger data are organized using the template in [Table 8](#).

Generate site source/sink curves using pinch software. These are site-wide composite curves for the process streams currently served by utility heaters or coolers. They are utilized to determine optimum utility generation and consumption profile or targets by utility level.

Determine the **gap** as the difference between the targets and actual usage at each level. This is the opportunity to improve the energy utilization. The gaps are shown in [Figure 6](#). The design is satisfactory if the gap is less than 15 MBtu/hr (4 MW). No further analysis is required. In case of a larger gap, proceed to the next step below.

Identify any possible gap closure alternatives such as steam level switching, changing steam pressure/level, adding new steam level, and inter-unit heat integration.

Check appropriate process units for technical feasibility and applicability of these alternatives.

Evaluate economics for the feasible alternatives using the preliminary utility pricing. **Develop** a short list of economic alternatives.

Simulate the utility system with the economic alternatives included to confirm the technical feasibility. The economics are re-checked with the revised utility pricing.

FOULING FACTORS

The increased resistance to both heat transfer and fluid flow caused by deposits on a heat transfer surface is called fouling. The following categories of fouling have been recognized:

1. **Precipitation Fouling** - Most frequently observed as a crystallization on the heat transfer surface of a relatively insoluble salt dissolved in a stream (e.g., calcium carbonate).
2. **Particulate Fouling** - Often seen as sedimentation where particles suspended in the fluid, such as sand or rust, deposit on the heat transfer surface.
3. **Chemical Reaction Fouling** - Coking, the deposit of hydrocarbons on high temperature heat transfer surfaces, is the most prevalent form observed in the refinery or chemical plant.
4. **Corrosion Fouling** - Deposits, such as iron oxide, formed by a chemical reaction involving the heat transfer surface.
5. **Biological Fouling** - The adherence and growth of organic organisms on heat transfer surfaces where untreated water is present. An example is the formation of algae from sea water.
6. **Freezing Fouling** - The solidification of a liquid onto a heat transfer surface.

Combinations of these categories often exist.

Fouling works as an insulating layer on the heat transfer surface, reducing heat transfer efficiency (reduced duty) or decreasing available flow area (reduced throughput). The increased resistance to heat transfer is represented by a quantity referred to as the fouling thermal resistance, which is added to the total thermal resistance. The values of fouling thermal resistance have generally been observed to increase with time. To account for the effect of fouling on pressure drop requires an estimate of the fouling layer thickness.

For design purposes a maximum permissible or expected value of the time-varying thermal resistance is used. This quantity is the fouling factor (denoted by the symbols r_{f0} and r_0 described in the beginning of the section). The net effect of the fouling factor is to increase the required amount of heat transfer surface area. At startup, the exchanger is oversized while just before scheduled cleaning, it is barely adequate.

The assignment of fouling factors allows the designer to identify where fouling dominates in a heat exchanger (shell side or tube side for a shell and tube exchanger). Fouling resistances can be compared to the other thermal resistances (such as film resistances R_{f0} and R_0). Based on this comparison, one may be able to alter the design to reduce fouling. In addition, one can determine the amount of surface area allocated for fouling.

Fouling factors should **not** be used as safety factors to account for some uncertainties in the design.



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RECOMMENDED GOOD PRACTICE

Physical factors that affect fouling include:

1. **Fluid Properties and the Tendency for Fouling** - For example, the feed to hydroformers, catalytic reformers and catalytic cracking plants is sometimes severely fouling due to organic reactions with oxygen that take place while the feed is in intermediate tankage. Inert gas blanketing of the tankage is often used to reduce the fouling tendencies of the fluid.
2. **Surface and Bulk Temperatures** - Higher temperatures generally result in faster rates of fouling.
3. **Fluid and Velocities** - Higher velocities reduce fouling rates, while stagnant regions or "dead zones" lead to higher rates of fouling. This is especially true for cooling water that contains salt, crude with sand, slurry solutions and dusty gases. Shell side velocities are limited by flow-induced vibration while tube side velocities are restricted by erosion considerations.
4. **Material Selection and Configuration** - Selecting the proper tube material may strongly affect corrosion. Finned tubing may not be appropriate for heavy fouling services. It has been found that algae growth on titanium surfaces in seawater service is much less than that on copper alloys.
5. **Exchanger Geometry and Orientation** - The geometry of the exchanger can affect the distribution of tube side and shell side flows. The orientation of the exchanger and pitch of the tubes can be altered for easy cleaning.
6. **Mode of Heat Transfer** - Fouling for a particular fluid may be significantly different depending on the mode of heat transfer (single-phase heating or cooling, boiling or condensing). Vaporization, for example, can cause severe precipitation fouling by concentrating the foulants in the remaining liquid to the point of supersaturation.
7. **Fluid Purity and Treatment** - Filtering streams to remove impurities and treating fluids (such as with additives) can drastically alter fouling. Crude oils, which normally contain salt-bearing water and suspended solids, are often placed in a desalter or a flash drum to remove water before the crude reaches the "salting-out" temperature.
8. **Placement of the More Heavily Fouling Fluid** - Cleaning of the tube side is generally easier than the shell side. Also, there is less chance of stagnant or "dead zones" on the tube side. Therefore, the more heavily fouling fluid should be placed on the tube side.
9. **Cleaning Method and Schedules** - The cleaning method (such as online mechanical cleaning, offline cleaning, bundle removal or in-place) should be determined at the design stage. Access to the exchanger should be considered to allow easy cleaning.

The best source of values for the fouling factor is plant data obtained from heat exchangers in similar services under similar operating conditions. Often these data are not available and the designer must rely on the values in [Tables 3-5](#). [Tables 3-5](#) gives normal fouling factors for a variety of process services, as recommended by the Tubular Exchanger Manufacturers Association (TEMA) and modified by experience. Note that the resistances given apply to the surface on which the fouling occurs. Although it appears to be somewhat ridiculous to adjust a value of such crude precision to a reference area, this step is required for computer calculations and is done during manual calculations for consistency. The tabulated fouling factors are intended to protect the exchanger from delivering less than the required process heat load for a period of about a year to a year and a half. However, this table is only a guide, and if site- (and unit-) specific data are available to determine a more accurate fouling factor for a particular service, the data should be used in preference to the values in [Tables 3-5](#).

The actual importance of the fouling factors depends on the value of the clean coefficient U_c ; the larger the coefficient, the larger the effect of the fouling factor. After a preliminary calculation of U_c , it is easy to determine the effect of doubling (or halving) the assumed fouling factors on the size of the exchangers. If this effect is small (5% or less), it is not worthwhile to try to determine a more accurate fouling factor. There are many times, however, when U_c is so large that the size of the exchanger depends almost entirely on the value of the fouling factor. For these cases, plant data, when available, should be closely examined.

The precision of the fouling factors used in the design should be weighed against the importance of the exchanger in its service. For example, one would want very precise fouling factors for an exchanger in a critical service. On the other hand, less accurate fouling factors may suffice where backup exchangers or bypass lines are available.

[Figure 7](#) provides a decision tree for the designer who wishes to evaluate the effects of fouling on the design of a shell and tube heat exchanger. After selecting values of the fouling thermal resistance, the designer is asked to run the HEXTRAN computer program. The output is first examined for the percentage of overall thermal resistance due to fouling. Next, hydraulic effects are considered. A fouling deposit is simulated using either data from operating experience, the values shown on [Figure 8](#), or by increasing the tube OD by an amount which results in a 10% decrease in tube to tube gap and/or decreasing the tube ID by specifying the next lower tube gauge. HEXTRAN is again run and tube side and shell side pressure drops are checked to see if they have exceeded the maximum allowable pressure drop. Also, tube side and shell side velocities are compared with those published in the TEMA manual or Exxon Design Practices. Finally, additional guidance can be provided by either local heat exchanger experts or the Heat Transfer Equipment Section at ER&E.