Understand Heat Flux Limitations on Reboiler Design

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Avoid the transition to film boiling to obtain better control and lower capital and operating costs.

Most chemical engineers involved with heat exchanger design are familiar with the concepts of nucleate versus film boiling and critical heat flux. Although a sensible heating medium, such as hot oil, may be used for some applications, most facilities employ steam at different pressure levels. This article explains how to account for the critical heat flux to avoid film boiling, and discusses the impacts of various heating media on the reboiler design.

Boiling regimes

Heat flux, the heat transferred per unit surface area (Q/A), is equal to the product of the heat-transfer coefficient and the temperature difference across the exchanger:

$$\text{Heat Flux} = Q/A = U \times MTD \qquad (1a)$$

The heat exchanger acts as a series of resistances represented by the inside heat-transfer coefficient h_i, the wall resistance r_w, and the outside coefficient h_o. The heat transferred must pass through every resistance and the flux is constant across each of them. Therefore, for a vertical thermosiphon reboiler with boiling in the tubes, the heat flux expression becomes:

$$Q/A = h_i(T_w - T_b) \qquad (1b)$$

Figure 1 is a typical heat flux curve for a reboiler. As the temperature differential between the heating medium and boiling fluid is increased (up to a maximum value ΔT_{crit}), the heat flux increases to a value of Q_{max}.

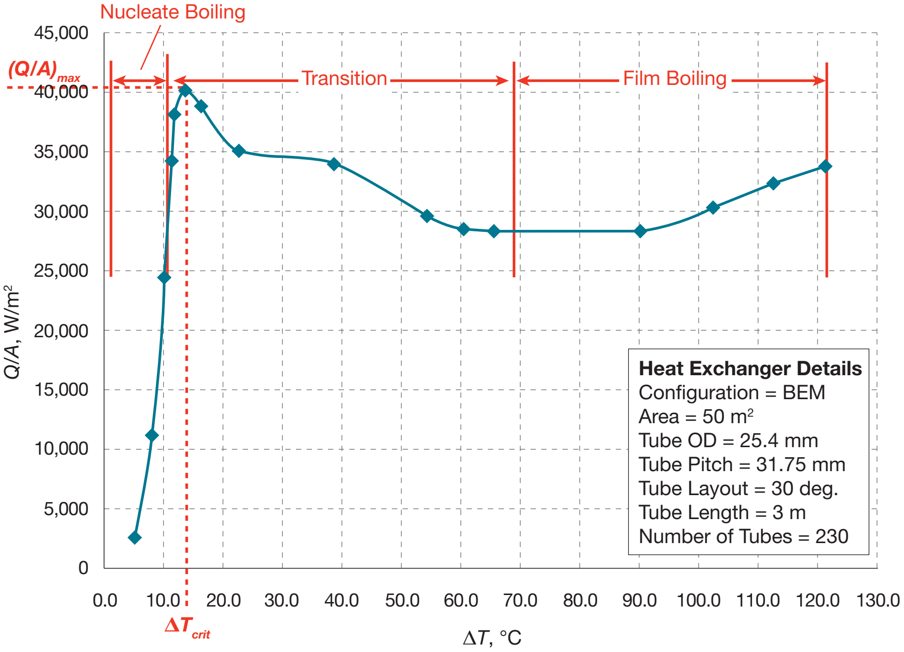
The region to the left of Q_{max} corresponds to nucleate boiling, in which vapor bubbles forming on the heat-transfer surface promote high velocities in the liquid film, thereby increasing the local heat-transfer coefficient. As the temperature difference is increased further, the bubbles begin to cause vapor blanketing, which prevents the liquid from reaching the surface. In this transition region, the heat flux decreases with increasing temperature until stable film boiling begins. If the temperature difference is increased further, stable film boiling continues, and the heat flux again begins to increase with increasing ΔT as convection to the liquid through the vapor barrier increases.

The exchanger depicted in Figure 1 is a vertical thermosiphon reboiler that uses steam to boil trichlorosilane (TCS) at 95°C. Here, Q_{max} = 40,000 W/m^2 at ΔT_{crit} = 14°C. There is a significant dead zone between the transition and film boiling regions, from 60°C to 90°C, where an increase in ΔT has no effect on the heat flux — this results in a difficult system to control.

It is good engineering practice to design a reboiler to operate in the nucleate boiling region *(1–3)*. To prevent control problems, avoid the transition region because an increase in the heating medium flowrate and/or temperature can result in a lower overall duty.

It is sometimes necessary to design a reboiler or vaporizer to operate in the stable film-boiling region. One such application is the boiling of cryogenic fluids, for which there is no suitable low-temperature heating medium. Even in these cases, however, the transition region must be avoided. Since the temperature of the heating medium typically correlates with its cost, designing for nucleate boiling is also the most economic option with respect to operating costs. In the TCS example, the reboiler is best operated with low-pressure (LP) steam at about 2–4 barg. The alternative is to operate in the stable film-boiling regime, which requires a steam pressure of about 35–40 barg.

One way to ensure that a reboiler operates in the nucleate boiling region is to limit the heat flux, by designing the exchanger for either a maximum *U* or a maximum overall *MTD* across the exchanger. Alternatively, some design criteria require only that a certain maximum heat-flux value $(Q/A)_{crit}$ not be exceeded — in these cases, the exchanger area is arbitrarily increased to reduce the flux below the specified maximum value. Design methods that arbitrarily oversurface a heat exchanger by specifying a maximum flux or heat transfer coefficient to address the film boiling issue are not always successful. It is more effective to limit the overall *MTD* such that ΔT_{crit} is not exceeded. Therefore, the selection of the heating medium and design of the process control scheme are critical to the success of the design.



▲ **Figure 1.** A heat flux curve for a vertical thermosiphon reboiler shows the nucleate boiling, transition, and film boiling regions. Heat flux *Q/A* is a function of $\Delta T\,(T_w - T_b)$.

Critical parameters for the transition to film boiling

Various guidelines have been proposed to limit *Q/A* and *U* to avoid film boiling, such as those given by Kern *(2)*: U_{max} = 300 Btu/h-ft²-°F (1,703 W/m²-K) for organics; U_{max} = 1,000 Btu/h-ft²-°F (5,768 W/m²-K) for water; and $(Q/A)_{crit}$ = 12,000 Btu/h-ft² (37,855 W/m²) for organics. The latter does not permit the use of large temperature differences for natural-circulation vaporizers and reboilers; for forced-circulation, the flux limit is relaxed to 20,000 Btu/h-ft² (63,092 W/m²), allowing for a somewhat higher ΔT driving force.

These guidelines for limiting the heat flux are considered conservative. The maximum heat flux for nucleate boiling depends on the physical properties of the boiling fluid, and on the geometry of the heat exchanger — *e.g.,* forced vs. natural circulation, boiling inside vs. outside the tubes, smooth vs. enhanced tube surfaces, etc. *(1–3)*. The maximum heat flux is generally expressed as a function of the reduced pressure, $P_r = P/P_c$, and Q_{max} decreases as the pressure approaches the critical pressure. Typically, the critical heat flux for a single tube is derived from the critical-pressure correlation to which appropriate geometry factors — *e.g.,* a bundle correction factor for boiling outside the tubes, factors for enhanced surfaces, etc. — are applied.

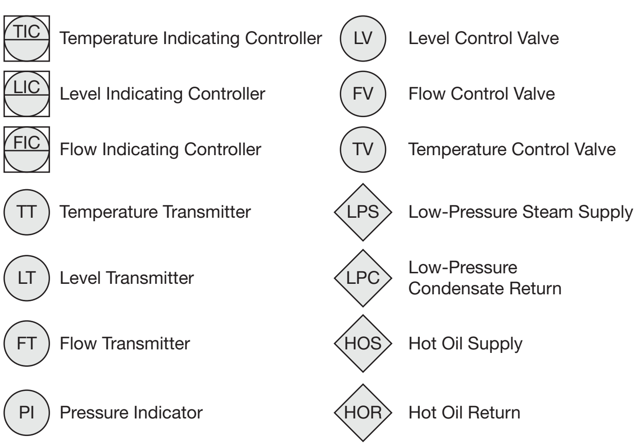
Calculation of Q_{max}, MTD_{crit}, and ΔT_{crit} is outside of the scope of this article; many sources give appropriate equations to use. Modern design programs, such as that offered by Heat Transfer Research, Inc. (HTRI; www.htri.net), in conjunction with physical property databases, such as AIChE's Design Institute for Physical Properties Research (DIPPR) database, predict the critical boiling heat flux for many fluids, and alert the designer to potential film-boiling problems. The designer need not perform the calculations, but does need to be aware of the issue and know what design variables can be manipulated to rectify the problem.

In some cases, constraints may be imposed on the maximum allowable heat flux based on operating experience. For example, heat flux in crude oil heaters is typically limited to 8,000 Btu/h-ft² (25,237 W/m²) based on the tendency of crude oil to foul at higher heat fluxes. While such experience-based values may appear to be very conservative compared with those derived from empirical formulas, they may be justified. Obtain as much information as possible about the system, including historical data, before deciding which methods to use to define limits on heat flux and temperature difference.

Article continues on next page

Because the critical heat flux is a function of *MTD* and ΔT_{crit}, it is perhaps more meaningful to know the critical temperature difference. Adding excess surface area is a somewhat artificial means of limiting the heat flux, since it does not change *U* or *MTD*, which are functions of the physical properties of the system. Ludwig *(3)* provides a table of Q_{max} and MTD_{crit} for various fluids and metals, as well as a graph showing the relationship between MTD_{crit} and the reduced pressure of the boiling fluid. Lacking data, a conservative value for MTD_{crit} may be selected — typically about 25–30°C (50–60°F). In the TCS example of Figure 1, MTD_{crit} is about 35°C.

Legend for Figures



Nomenclature

A	= area, m^2
c_p	= heat capacity, J/kg-K
h_i	= inside wall coefficient, W/m^2-°C
h_o	= outside wall coefficient, W/m^2-°C
H_{vap}	= latent heat, J/kg
MTD	= mean temperature difference, °C
MTD_{crit}	= mean temperature difference at Q_{max}, °C
P	= pressure, barg
P_c	= critical pressure, barg
P_r	= reduced pressure, dimensionless
Q	= duty, W
Q_{max}	= maximum nucleate boiling duty, W
Q/A	= heat flux, W/m^2
$(Q/A)_{crit}$	= maximum heat flux limit, W/m^2
r_w	= wall resistance, m^2-°C/W
T_c	= thermal fluid return temperature, °C
T_h	= thermal fluid supply temperature, °C
T_w	= process-side wall temperature, °C
T_b	= process-side bulk temperature (boiling fluid saturation temperature), °C
U	= overall heat transfer coefficient, W/m^2-°C
U_{max}	= maximum nucleate boiling overall heat transfer coefficient, W/m^2-°C
W	= mass flow, kg/h
ΔP	= pressure drop, bar
ΔT	= temperature difference, °C
ΔT_{crit}	= critical temperature difference at $(Q/A)_{crit}$

Controlling the reboiler heat balance

Equations 2 and 3 describe the heat balance for a reboiler relative to the utility-side enthalpy change for thermal fluid heating (sensible heat transfer):

$$Q = U \times A \times MTD = Wc_p(T_h - T_c) \tag{2}$$

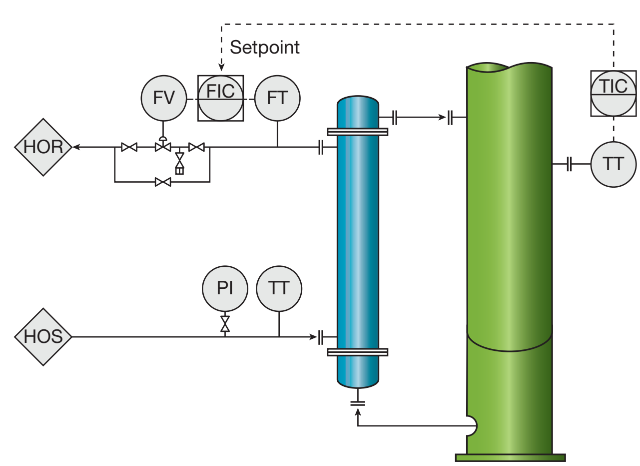
and steam heating (condensing heat transfer):

$$Q = U \times A \times MTD = WH_{vap} \tag{3}$$

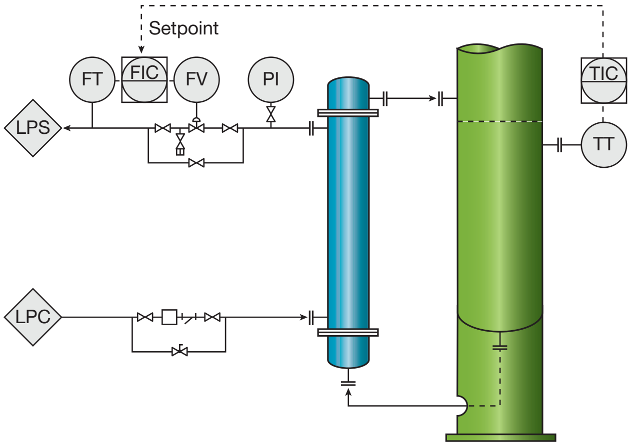
Process-side heat transfer is omitted from these equations. It is assumed that in the nucleate boiling region, the process-side heat transfer is not limiting, and that the exchanger geometry is sufficient to transfer the heat delivered by the heating medium. However, as Q_{max} is exceeded and the exchanger begins to operate in the transition region, the process-side coefficient will decrease, and the process heat transfer will become limiting.

These equations show that the duty of a reboiler is controlled by varying the flowrate, *W*, of the heating medium (steam or thermal fluid). Alternatively, for a condensing medium, the duty may be controlled by back-flooding the heat exchanger, thereby varying the area, *A*. The latter is most effective in vertical units with a linear response, such as a vertical thermosiphon reboiler.

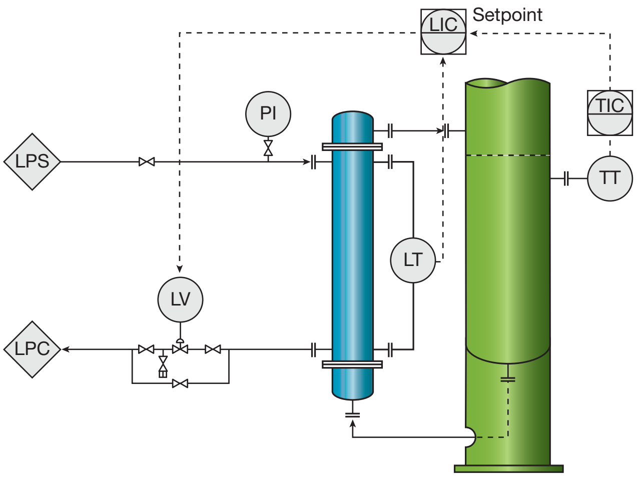
For a reboiler using a sensible heating medium (Figure 2) operating in the nucleate boiling region, the heat duty



▲ **Figure 2.** The column bottoms temperature of a thermosiphon reboiler heated by a sensible heating medium (hot oil) is controlled by varying the hot oil flowrate.



▲ **Figure 3.** The column bottoms temperature of a thermosiphon reboiler heated by steam is controlled by varying the steam flowrate.



▲ **Figure 4.** The column bottoms temperature of a thermosiphon reboiler heated by steam is controlled by varying the condensate level in the exchanger, and thus, the available area for heat transfer.

will increase (approximately) linearly as the flowrate of the heating medium is increased. This is due to the strong correlation between the velocity of the liquid heating medium and the heat-transfer coefficient — in Equation 2, as W increases, U increases at a similar rate. Note that as film boiling starts to develop in the transition region, the overall heat transfer decreases — U decreases on the left side of Eq. 2, while the temperature change of the thermal fluid decreases on the right side, making the heat-transfer fluid less effective as it transfers less of its heat to the system. In this way, both sides of the heat transfer equation are balanced.

However, condensing media such as steam behave differently. Figure 3 shows a thermosiphon reboiler with flow control on the steam side to regulate the condensing steam. In the nucleate boiling region, the exchanger duty will increase as the steam flow is increased. On the right side of Eq. 3, H_{vap} is relatively constant over the range of operation, such that condensation of the heating medium at a higher rate must result in an increase in the heat transferred. On the left side of Eq. 3, the condensing coefficient (U) is relatively constant (for steam condensation). Since the heat transfer area is fixed, MTD must change. Because the process-side temperature is essentially constant, the condensing temperature must change to satisfy the heat balance. Since there is a direct relationship between the temperature and the pressure of the heating medium, the chest pressure on the shellside of the exchanger must also change.

In the case of variable area control using steam, as shown in Figure 4, the left side of Eq. 3 is manipulated by changing the area of the exchanger by back-flooding with condensate. The chest pressure is constant and equal to the supply pressure of the steam. The steam flowrate will then vary such that the equality of Eq. 3 is maintained — *i.e.,* the reboiler will condense an amount of steam based on the geometry (taking into account the exposed condensing surface) and the process-side heat transfer parameters. As the steam condenses, a slight vacuum relative to the supply pressure will develop and steam will flow into the reboiler at a rate equal to the condensation rate, thereby satisfying the heat balance.

Heat flux considerations for reboiler design

There is a misconception among many engineers that the only requirement to ensure that a reboiler will operate safely within the nucleate boiling region is to keep the heat flux below a specified safe value by providing adequate excess surface area, or by specifying an arbitrarily low overall heat-transfer coefficient (which has the same effect). This is not always the case, and it can depend on the type of heating medium and the control scheme that are selected. In fact, design methods that specify oversurface to limit U or the overall heat flux do not directly address the nucleate boiling issue — the additional surface provided compensates for the lower film boiling coefficient, rather than preventing film boiling.

Sensible heating medium. For a sensible heating medium such as a thermal fluid or hot oil, it is important to consider the supply temperature. Select the thermal fluid inlet temperature so that the temperature difference between the heating fluid and the process side is less than MTD_{crit}, where the transition to film boiling would occur.

In Eq. 2, the process-side temperatures can be considered constant, as is the hot oil supply temperature T_h. The system may be controlled by throttling the control valve to vary W, which will affect U, T_c, and MTD. Regardless of the geometry of the exchanger, MTD at the first increment near the shellside hot oil depends on the inlet temperature of the hot oil. If this temperature is high enough to cause ΔT_{crit} to be exceeded, film boiling will occur in the first increment of the exchanger. Since the heat-transfer effectiveness is diminished due to film boiling, the temperature change of the hot oil in the first increment will be smaller, and, therefore, the inlet temperature to the next increment will be higher. In this way, transition or film boiling propagates from one increment to the next. The result is an exchanger that operates in the transition or film-boiling regime, and in the worst case, an undersized exchanger.

A similar result is obtained if an arbitrary limit is chosen for U in order to provide over-surface to avoid film boiling. In operation, there is no limit on U, which is based on the physical properties of the heating and process fluids and exchanger geometry. Therefore, a high actual heat-transfer coefficient may result in film boiling despite selecting a lower U for design purposes.

Thus, be aware that although some design programs provide a limit to the maximum allowable heat flux, this is not always sufficient to prevent film boiling and degradation of reboiler performance. The HTRI design program is robust in this respect, since it performs zone-by-zone analysis to determine whether a transition to film boiling would occur, and if so, it accounts for this in the calculation of the expected heat-transfer coefficient.

A good strategy is to create a first-pass design of the reboiler, including parametric studies to optimize the design, followed by a rating or simulation of the exchanger to fine-tune design parameters such as baffle spacing and cut and to check for problems such as vibration and a transition to film boiling.

In a recent in-house engineering project, heat exchanger design software was used to design a vertical thermosiphon reboiler with adequate over-surface to theoretically limit the heat flux to below 30,000 Btu/h-ft^2 (94,638 W/m^2). However, when it was rated using more detailed methods, it was found to be 40% undersized due to operation in the film-boiling regime in all zones of the reboiler. This illustrates the importance of carefully examining film boiling. Had this exchanger been designed with only the heat flux limitation, it would have underperformed significantly when installed.

Condensing heating medium. A similar problem occurs if variable area control (see Figure 4) is used with a condensing heat-transfer medium, such as steam. Because the heat balance is manipulated by changing the exchanger area, the shellside chest pressure is constant and equal to the steam supply pressure. If the steam pressure is too high, its saturation temperature may cause ΔT_{crit} to be exceeded, causing film boiling to occur and the process-side heat-transfer coefficient to decrease. This will result in reduced heat transfer and lower steam consumption than the design values. If the exchanger is operating in the transition region, it will be difficult to control, since reducing the area could result in locally higher film-boiling rates and an indeterminate overall heat-transfer coefficient.

It might not be possible to fine-tune the steam levels available in the plant to eliminate the possibility of film boiling. A preferred means of control, then, is to install a control valve to throttle the steam flow to the reboiler (Figure 3). This has the advantage of reducing the pressure, and thereby temperature of the steam side, to eliminate any possibility of film boiling. If the equipment and controls are designed properly, this will, in fact, preclude any possibility of film boiling occurring in the reboiler.

When the steam-throttling valve begins to operate (opening from the fully closed position), a relatively small amount of steam is admitted to the reboiler, where it condenses at a temperature (and corresponding pressure) that satisfies the heat balance. Since the reboiler is capable of transferring more heat — *i.e.,* condensing more steam — than is being added to the system via the control valve, a vacuum develops on the shellside of the reboiler. Because the control valve position is held constant, the vacuum increases the available ΔP across the control valve, thereby increasing the steam flowrate. The steam flow increases until the flow through the control valve is equal to the rate of steam condensation in the reboiler, based on the chest pressure and *MTD* that satisfy the heat balance. Therefore, the steam throttling valve not only controls the flow of steam to the reboiler, it also controls the reboiler steam-side pressure and temperature — thereby controlling the reboiler *MTD* and the overall reboiler duty.

As the control valve is opened further to admit more steam, the steam pressure on the reboiler shellside increases as the reboiler's ability to condense steam comes into balance with the rate at which the control valve is supplying steam. As the steam chest pressure increases, the steam saturation temperature, the *MTD* across the reboiler, and the heat transfer rate also increase. If the control valve continues to open to the full-open position, there is no longer control over the system and the reboiler will condense steam to the maximum extent possible, based on its total area and the process-side heat transfer requirements. If the steam header pressure (and therefore temperature) is relatively high, and if the control valve is oversized (design pressure drop is low), film boiling in the reboiler could result.

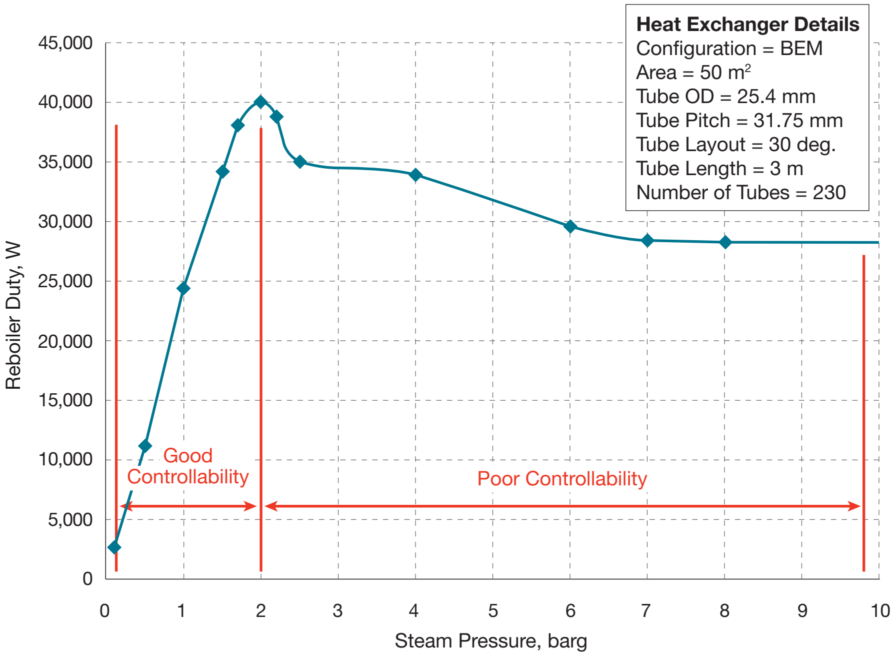
Although the steam superheat is significant for some applications, for most steam-heated reboiler designs it may be neglected. For example, for a superheated steam supply at 8 barg and 205°C and a steam chest pressure of 1.5 barg at a saturation temperature of 127.5°C, the steam superheat (the difference between 205°C supply and 127.5°C) is only about 5% of the total enthalpy change.

Steps for reboiler design

First, determine either the maximum heat flux limit, $(Q/A)_{crit}$, or ΔT_{crit} (based on the heating medium temperature and overall MTD_{crit}), for the process fluid. This may be a conservative estimate based on operating experience, or a published rule-of-thumb criterion, calculated based on the critical pressure, or obtained by trial and error using a design program such as HTRI's. When using such a program, vary the heating fluid conditions (temperature and/or pressure) until a transition to film boiling is no longer detected and stable nucleate boiling is present in all increments. If $(Q/A)_{crit}$ is used, calculate a corresponding ΔT_{crit} or MTD_{crit}, based on the actual overall heat-transfer and film coefficients achieved. Note that the temperature difference needs to be referenced to the resistance — via the overall *MTD* if an overall *U* is used, but via $(T_w - T_b)$ if the film resistance (h_i) is used. In general, use of $(T_w - T_b)$ is more instructive, since it relates to the physical limit of the system (*i.e.*, inside wall temperature) rather than vs. incipient film boiling.

Table 1. A steam control valve was sized based on an HTRI exchanger simulation.

	Turndown, clean	Turndown	Normal	Design
Flowrate, kg/h	1,450	1,450	2,900	3,480
Heat Duty, *W*	893,000	899,000	1,761,000	2,101,000
Steam Chest Pressure, barg (calculated)	0.7	0.4	1.4	1.7
Saturated Steam Temperature, °C	114.3	108.3	125.5	129.9
ΔT (with process T = 95.2°C)	19.1	13.1	30.3	34.7
MTD (corrected), °C	18.1	11.9	29.2	33.6
Steam Supply Pressure, barg	8	8	8	8
Control Valve Available Pressure Drop, barg	7.3	7.6	6.6	6.3
Valve Size = 2 in.; Rated *Cv* = 75 (sized for design rate based on calculated chest pressure)				
% *Cv*	20.2%	20.2%	42.1%	51.5%
Valve Size = 3 in.; Rated *Cv* = 155 (sized for design rate assuming chest pressure of 7 barg, valve ΔP = 1 bar)				
% *Cv*	9.7%	9.7%	19.4%	23.5%



▲ **Figure 5.** A reboiler duty vs. steam pressure curve for a thermosihpon reboiler shows the best range for controllability.

Once the "safe" *MTD* corresponding to ΔT_{crit} and the heating medium temperature is known, the exchanger may be designed by typical methods to determine the overall heat-transfer coefficient, geometry, and total area. In design programs such as HTRI's, the optimization algorithms may be used for the initial parametric studies; however, the final design should be optimized and evaluated using the program's rating tool.

After the exchanger geometry has been fixed based on the design case, rate the exchanger for alternative operating conditions, such as normal and turndown operation. For a steam-heated reboiler, this requires iteration of the steam conditions to determine the required steam temperature and pressure for the various operating cases. Once these alternative steam-side pressures are determined, the steam control valve can be specified, with its pressure drop equal to the difference between the steam supply pressure and the calculated steam pressure in the exchanger shell.

In some cases where a lower steam pressure is required to prevent film boiling, it may be necessary to specify a control valve pressure drop that is significantly larger than the 0.7–1.0 bar typically specified for controllability. It is important to base the control valve pressure drop on the calculated chest pressure to ensure that the reboiler is controllable over its operating range. This also avoids oversizing the control valve, which eliminates controllability problems at low pressures.

A steam control valve for the reboiler discussed earlier was sized for turndown, clean turndown (nonfouled exchanger), normal, and design steam rates based on an HTRI exchanger simulation. First, the valve was correctly sized based on the pressure drop required to reduce the steam pressure to the calculated operating steam chest pressure, and then compared to a control valve sized based on a pressure drop of 1 bar. As the results in Table 1 show, the correctly sized valve is not only smaller, but will also operate in a more-controllable range, illustrated in the duty vs. steam pressure curve for this exchanger in Figure 5.

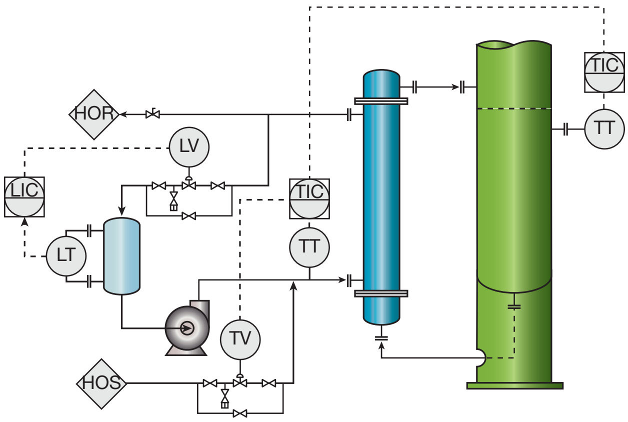
A related issue is the ability to return condensate from the reboiler. The chest pressure is a function of the reboiler area and prevailing heat transfer rate, and it will vary depending on whether the reboiler is operating at design, normal, or turndown rates, as well as on the degree of fouling. From the standpoint of minimum condensate pressure, the worst case is a clean reboiler operating at turndown conditions. To determine the minimum steam pressure, the reboiler should be rated at turndown duty with no fouling factor. In general, if the condensate return pressure is close to the condensate return header design operating pressure, a condensate pump or pumping trap will be required.

If a sensible heating medium is to be used, select a heating medium inlet temperature T_h such that $T_b - T_w$ remains safely below ΔT_{crit} in all zones of the exchanger. In an existing plant, if the supply temperature of the heating medium is too high, a tempered hot-oil loop should be used. In a tempered loop (Figure 6), the thermal fluid flow through the exchanger is continuous, and the temperature of the loop is controlled by admitting hot thermal fluid to manipulate the hot oil supply temperature and exchanger *MTD*, and control the exchanger duty.

Evaluating an existing steam-heated reboiler

Reboiler evaluation based on the process-side conditions may be difficult if not impossible. The process-side temperature rise may be small, the circulation rate in a thermosiphon reboiler unknown, and the measurement of the column vapor rate impractical. Performance must be inferred from feed, reflux, and bottoms flowrates and product specifications.

Evaluation of a thermosiphon reboiler based on steam-side measurements is relatively straightforward. These measured parameters include the steam flowrate, steam



▲ **Figure 6.** The column bottoms temperature of a thermosiphon reboiler heated by a tempered sensible heating medium (hot oil) is cascaded to the hot oil loop temperature, which controls the flowrate of hot oil into the loop.

chest pressure at the reboiler inlet, condensate return temperature, and process-side temperatures for the column bottoms (reboiler liquid supply) and the vapor to the first tray (which is equal to the reboiler vapor/liquid return temperature).

The reboiler is assumed to operate with the total area available for heat transfer, and the steam-side temperature is taken as the saturation temperature at the steam-side operating pressure. Since process-side temperatures for the inlet liquid and outlet streams (the vapor temperature is measured) are known, the reboiler *MTD* is known. With the installed area of the heat-transfer surface also known, the prevailing overall heat-transfer coefficient (U) can be easily calculated.

A condensate return temperature that is significantly lower than the saturation temperature at the steam chest pressure indicates condensate backup, which causes subcooling of the condensate. This renders a portion of the exchanger area relatively ineffective, since it is sensibly cooling the condensate rather than condensing steam. Subcooled condensate could indicate insufficient chest pressure to return condensate to the header, or problems with the steam trap.

If the unit operates at a relatively constant duty, the steam-side pressure can be monitored to track the fouling of the exchanger over time. As a clean exchanger becomes fouled, the overall heat-transfer coefficient decreases, requiring a higher *MTD*, and thus a higher steam chest pressure, to transfer the same duty. Similarly, the steam control valve will have to gradually open more over time.

Wrapping up

It is good engineering practice to design a reboiler to operate in the nucleate boiling region, and to avoid the transition to film boiling, where control becomes difficult. In addition, operation in the nucleate boiling region will typically result in lower capital and operating costs compared with an exchanger designed to operate with stable film boiling. The latter should be avoided except in cases where it is unavoidable, such as in cryogenic heat exchangers.

Avoiding film boiling is not as simple as limiting the heat flux or overall heat-transfer coefficient by adding excess surface area to the exchanger. Oversurfacing will work for a condensing heating medium such as steam — but only if the steam pressure is throttled using a control valve, and if the control valve is properly sized, (*i.e.*, the design accounts for the operating chest pressure). Duty control via condensate level should be avoided unless the steam supply temperature and pressure are such that ΔT_{crit} will not be exceeded, or another means is provided to regulate the steam supply to a low enough temperature to avoid the transition to film boiling.

In the case of a variable-area steam condensing reboiler, the steam supply pressure selected must limit the exchanger *MTD* so that ΔT_{crit} is not exceeded. Similarly, when a sensible heating medium, such as hot oil, is used, the supply temperature must be selected so that each increment of the exchanger operates safely below ΔT_{crit}. If this is not possible, a tempered loop should be used to limit the hot oil supply temperature. CEP

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