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The P-NTU$_I$ Method

Classical Heat Exchanger Performance Analysis Methods

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PRODUCT DESCRIPTION

The heat transfer calculation method that is currently used in the nuclear industry is the corrected log mean temperature difference F-LMTD method. This method has limitations in dealing with the range of heat exchanger analysis requirements encountered in the industry. A primary objective of this report is to provide a concise, service water-focused introduction to the broader family of practical analytical methods. The intent is to inform and educate, while laying the groundwork for additional efforts that will provide improved and optimized options for heat exchanger performance analysis.

Challenges and Objectives

The Nuclear Regulatory Commission (NRC), in Generic Letter 89-13 (GL 89-13), "Service Water System Problems Affecting Safety-Related Equipment," required utilities to demonstrate that safety-related service water heat exchangers perform their design functions. In response, the industry developed a number of standards and guidelines that outline a common approach to performance-testing heat exchangers, evaluating test results, projecting the results to design conditions, and determining the overall uncertainty of the test and the projected results. These industry guidelines relied exclusively on the F-LMTD methodology for heat exchanger analysis.

After the NRC issued GL 89-13, utilities responded to the demand for heat exchanger testing and test evaluations with increasing reliance on heat exchanger modeling software. The methodologies implemented in commercial software include both traditional classical methods and numerical methods. Although heat exchanger analysis software can be quite useful to utility engineers, relying on such packages without an understanding of the underlying, fundamental heat exchanger analysis principles can result in misguided decisions in validating computer results, understanding test or performance anomalies, and evaluating alternatives. With many engineers who are responsible for the performance of safety-related service water heat exchangers nearing retirement, it is increasingly important to strengthen the analytical understanding of heat exchangers at the utilities.

Application, Value, and Use

This report is designed for service water system engineers, service water program owners, mechanical design engineers who are responsible for modeling and calculating GL 89-13 heat exchanger thermal performance, GL 89-13 program owners, and heat exchanger program owners.

EPRI Perspective

In response to GL 89-13 and, more directly, the NRC's recommendation for all licensees to "conduct a test program to verify the heat transfer capability of all safety-related heat exchangers cooled by service water," nuclear utilities, under the auspices of the Electric Power Research Institute (EPRI) and in communication with the NRC, developed technical guidelines for the conduct of performance testing to meet the intent of GL 89-13. EPRI published these guidelines in 1998 in the EPRI report *Service Water Heat Exchanger Testing Guidelines* (TR-107397). This report has been consistently accepted as the industry standard for GL 89-13 heat exchanger thermal performance testing.

The team of industry engineers, consultants, and manufacturers who collaborated to develop EPRI report TR-107397 agreed to base the analytical approach recommended in the performance testing guidelines on the classical F-LMTD method. This methodology and the historical, empirical correlations have been used to support actual in-service testing and test projection of heat exchangers in the commercial nuclear industry. Since 1998, the analytical approaches recommended in EPRI report TR-107397 have been largely incorporated into the performance test procedures, calculations, and industry software that provide the backbone of the nuclear industry's GL 89-13 heat exchanger thermal performance testing programs.

This report reviews classical analytical methods, including their basic conceptual relationships, their overlapping interrelationships, and their differences. This report provides several examples to demonstrate the practicality of the methods to address heat exchanger rating questions (such as how a heat exchanger will perform with a specific number of tubes plugged or how a change in cooling water mass-flow rates and inlet temperatures will affect heat exchanger performance).

In addition, and more importantly to the GL 89-13 program, the unified classical methodology provides a practical basis for analysis of heat exchanger testing results and projection to required operating conditions – the ultimate answer required for performance testing. Therefore, this report is also intended to collectively prepare the EPRI membership for the development and implementation of more straightforward and comprehensive guidelines for testing, test-projection, and uncertainty analysis, with the particular objective of obtaining the minimum credible uncertainty at projected conditions.

Keywords

Generic Letter 89-13
Heat exchanger
Heat exchanger rating
Performance testing
Service water
Uncertainty calculations

CONTENTS

LIST OF FIGURES

LIST OF TABLES

1 INTRODUCTION

Classical heat exchanger analysis methods are commonly used by practicing engineers for performance analysis tasks involving rating, testing, and preliminary design. These classical methods have evolved from basic analytical solution relations developed in the 1930s and 1940s, involving three primary performance parameters—temperature effectiveness, *P,* capacitance ratio *R,* and number of transfer units NTU_I. The classical methods that are widely used in practice are the corrected log mean temperature difference F-LMTD method and closely related P-NTU_I and ε-NTU effectiveness methods. The ε-NTU method has been used in industries that depend on compact heat exchangers since the 1950s; the F-LMTD method has been used almost exclusively in power, process, and petrochemical industries.

In addition to providing the basis for the classical methods that feature the primary performance parameters *P*, *R*, and NTU_I, the underlying solution relations have led to the identification of a fourth key performance parameter known as the effective mean temperature difference *EMTD* which is also commonly designated *MTD*. The recognition of a natural link between *EMTD* and the primary performance parameters has resulted in the formulation of a new analytical approach, referred to as the EMTD method [1, 2], which eliminates the need for calculating the correction factor *F* and the log mean temperature difference *LMTD*. This new method is complementary to the effectiveness methods and a compelling alternative to the F-LMTD method. The *EMTD* linking relations provide the basis for establishing a unified classical heat exchanger analysis methodology that can deal with the range of issues that arise in performance rating, testing, test projection, and uncertainty analysis in a more straightforward and comprehensive way than either the F-LMTD or the effectiveness methods individually [2].

The underlying modeling assumptions upon which the classical methods are based can be used for applications involving nonuniform geometric characteristics, moderate to large variations in fluid properties and/or convection coefficients, and other variations. For applications in which factors such as these might be significant, more general, numerical solution approaches should be considered to confirm, supplement, or even replace the classical methodology.

This report provides an assessment of the current approach to heat exchanger performance analysis in the nuclear industry, an overview of unified classical methodology, guidelines for classical performance rating featuring the P-NTU_I method, and practical examples of performance rating applications associated with nuclear power plant service water systems. This report is intended to provide a foundation for adaptation of the classical methodology to testing, test projection, and uncertainty analysis.

1.1 Background

1.1.1 Industry Experience and Standards: Reliance on F-LMTD Method

Historically, the electric power industry has relied almost exclusively on the classical F-LMTD method for the analysis of thermal performance of heat exchangers. Nuclear Regulatory Commission (NRC) Generic Letter 89-13 (GL 89-13) required utilities to demonstrate that service water cooled, safety-related heat exchangers at all commercial nuclear power plants could perform their design basis functions [3,4]. In the nuclear industry, safety-related heat exchangers often perform dual functions, one during normal operation and a second during design basis accidents. The two operating conditions generally vary significantly in required flow rates and heat removal capability. Although it is relatively easy to test the normal operating performance, the design accident conditions cannot be economically achieved and therefore preclude testing in the worst case scenario. The industry responded with the development of a series of standards and guidelines in an effort to achieve economic, feasible, and reasonable testing procedures with realistic uncertainties. Those standards and guidelines include the following:

- The Electric Power Research Institute (EPRI) report *Heat Exchanger Performance Monitoring Guidelines* (NP-7552), presents guidelines for performance monitoring of heat exchangers that are subject to fouling [5]. Five monitoring methods are discussed: heat transfer, temperature monitoring, temperature effectiveness, change in pressure (ΔP), and periodic maintenance. The heat transfer method, which is the only one that involves direct measurement of heat transfer capability, relies on the F-LMTD method for evaluation of test results and projection to the design basis conditions.
- *ASME OM-S/G—1994 Standards and Guides for Operation and Maintenance of Nuclear Power Plants*, Part 21, "Inservice Performance Testing of Heat Exchangers in Light-Water Reactor Power Plants," establishes preservice and inservice testing requirements to assess the operational readiness of certain heat exchangers used in nuclear power plants [6]. It establishes test intervals, parameters to be measured and evaluated, acceptance criteria, corrective actions, and historical information requirements. The standard also introduces requirements to consider the analytical uncertainties associated with the evaluation and projection analyses, in addition to considering the measured uncertainties. Although this standard does not specify specific analytical methods, all the examples provided in nonmandatory Appendix C use the F-LMTD method.

- *ASME PTC 12.5—2000 Single-Phase Heat Exchanger*s, provides a standard for performance testing of industrial heat exchangers to compare installed capacity with design specifications, assess degradation, and evaluate the effectiveness of performance improvements [7]. It outlines detailed and consistent test practices that are intended to achieve reliable and accurate heat exchanger thermal performance test results. The procedure outlined in the standard is derived exclusively from the F-LMTD analytical methodology. The test evaluation and performance projections are combined into a single analytical step to simplify the post-test analysis. An uncertainty analysis, including the effects of analytical uncertainty and correlations, is included.
- The EPRI report *Service Water Heat Exchanger Testing Guidelines* (TR-107397), reviews industry experience for testing service water heat exchangers and provides guidelines to assist in designing specific test methods and measurement strategies for each heat exchanger application [8]. The test methods presented are intended to provide assurance that results are sufficiently representative of the true capability of the heat exchanger. The report focuses on the thermal performance test and uses the F-LMTD methodology for test evaluation and projection of results to accident conditions.

1.1.2 Characteristics of Standard Industry Approach to Heat Exchanger Rating

The following three basic heat transfer problems are involved in the operation of nuclear power plant service water heat exchangers:

- Heat exchanger rating
- Evaluation of thermal performance tests
- Projection to design conditions based on test fouling resistance

In response to NRC GL 89-13, the nuclear industry developed heat exchanger test programs and standardized test evaluation and projection methodologies [3, 4]. The emphasis on a single analytical method has failed to account for significant differences in the computational characteristics of these problems. This has resulted in unnecessary complications and limitations in evaluating the thermal performance and uncertainty. This report focuses primarily on the rating-problem to familiarize the power plant engineer with the P-NTU_I method and to establish a foundation for use of the unified classical method for performance testing, test projection, and uncertainty analysis.

Heat exchanger rating problems, in general, use and expand on the original design calculations. The utility design engineer specifies flow rates, heat load, inlet temperatures, pressure drops, and, in some cases, an outlet temperature. To predict the heat exchanger performance, engineers perform heat exchanger rating calculations to evaluate off-design conditions. In heat exchanger rating calculations for alternative design conditions, the surface area is generally known, and engineers evaluate changes in component performance due to changes in the mass-flow rates, inlet temperatures, number of available tubes, fouling, or required heat duty.

The utility engineer should perform rating calculations for new or replacement heat exchangers to validate the heat exchanger rating sheet and design provided by the manufacturer. These rating calculations will validate the rated performance, establish available margin, and provide a reproducible basis for comparison in thermal performance tests and evaluations. In many cases, engineers will not be able to reproduce the rating sheet values exactly due to the refined analytical methods used by heat exchanger manufacturers. The engineer must be satisfied that the rating sheet accurately represents the heat exchanger's design performance and that conservative analysis can support future evaluations and performance testing. The engineer's calculations should review the following factors that might impact later evaluations:

- **Shell-side heat transfer coefficient.** The specification of the shell-side heat transfer coefficient is one of the most difficult aspects of heat exchanger performance analysis because of the complex nature of the shell-side flow. Many manufacturers specify the coefficient using proprietary computer codes that feature the stream analysis method. If the owner cannot reproduce the coefficient in evaluations, the results of future performance testing can be significantly affected. This report provides some guidance on calculating the shell-side coefficient, but it does not provide an extensive study of calculation methods for this task. The manufacturer should be required to provide the actual design parameters for several critical operating conditions, including the shell- and tube-side heat transfer coefficients, to support validation of the utility engineer's analytical methods for future testing.

  When practical, the engineer should also evaluate the thermal performance characteristics (including the shell-side coefficient) for critical, safety-related applications by testing during the initial, preinstallation operation, while the heat exchanger is still in a clean condition.

- **Hidden margin (overdesign).** In many cases, manufacturers have provided heat exchangers with heat removal capability beyond that required by the user and have not reflected this margin on the rating sheets. For example, an off-the-shelf heat exchanger design can meet the design specification required by the owner, but in fact, have a heat duty that is higher than required. As a result, the manufacturer provides the required heat duty on the rating sheet and the excess capability is "hidden margin." The owner can identify this margin by carefully reconstructing the design calculations and comparing them to the original rating sheet. Failure to identify this margin introduces an error in the specified design overall coefficient of heat transfer and leads to unnecessarily conservative test and test projection results. Performance testing immediately after installation can identify this hidden margin, which is often revealed in test results as a negative calculated fouling resistance.

- **Fouling resistance**. Electric utilities generally specify the fouling factor (or cleanliness factor) during the design and procurement of a new heat exchanger. Because of the significance of fouling resistance on effective operation, it is common practice for owners to perform rating calculations for the specified, design fouling level to verify that new heat exchangers can satisfy performance requirements. Furthermore, the determination of the effect of a specified level of fouling on the operability of a heat exchanger is an important function of performance rating in electric utilities.

Owners should keep in mind that most manufacturers use computer-based analysis for their heat exchanger design. The owner generally cannot access the same software for later design and rating issues or evaluations. Therefore, it is imperative for the owner to reconcile the rating sheet to the analysis methods available for in-house evaluations. For situations in which the design reconciliation is unsuccessful, the differences should be identified and resolved with the manufacturer. Generally, if the rating can be reconciled using classical heat exchanger design methodology, access to a specific heat exchanger computer code is not required to achieve consistent evaluation results in the future. This report is intended to provide the utility engineer with a good understanding of classical methodologies so that exclusive reliance on commercial computer codes is not required.

1.1.3 Characteristics of Standard Industry Approach to Performance Test Analysis

The industry approach to heat exchanger performance testing analysis is well established and documented in the existing guidelines. Performance test evaluation involves measuring five or six of the process parameters (such as mass-flow rates and terminal temperatures). These measurements are used to determine the overall coefficient of heat transfer for test conditions. The difference between the inverse calculations for the test overall coefficient of heat transfer and the predicted clean overall coefficient of heat transfer represents an apparent fouling resistance that is due to tube- and shell-side fouling and the net effect of any other factors that have not been adequately accounted for in the analysis. Examples of non-ideal factors that are difficult to model and that might affect the thermal performance of a heat exchanger include the existence of tube- or shell-side bypass flows, tube leakage, and tubes obstructed by debris or macroscopic fouling.

To minimize the potential impact of failing to identify non-ideal factors that result in an imbalance in tube- and shell-side heat transfer rates, the industry guidelines recommend measuring all six performance parameters (that is, two flows, two inlet temperatures, and two outlet temperatures). This six-point method allows a determination of the heat duty based on both tube- and shell-side performance. Test validation requires that the heat balance error be consistent with measurement uncertainties. Although the six-point method provides a basis for test validation, the use of six process parameters introduces analytical inconsistencies because the calculations for tube- and shell-side heat duty normally differ as a result of measurement uncertainties.

The five-point method provides an alternative for applications involving a process parameter that cannot be measured accurately or at all. However, industry guidelines have not yet been established for five-point testing analysis. Although the use of only five parameters limits the ability to assess the validity of the measured test parameters through use of the conservation of energy, the practical value of the five-point method in a number of nuclear power plant applications provides justification for taking steps to develop test and analysis guidelines with a clear indication of the capabilities and limitations of this approach.

Ultimately, test evaluation provides the basis for calculating the heat duty, overall coefficient of heat transfer, and apparent fouling resistance that exist during the test. Whereas the calculation of heat duty indicates whether or not heat exchangers tested at design conditions are operable, the calculations for overall coefficient of heat transfer or apparent fouling resistance provide the basis for establishing operability for other applications by projecting results to the required operating conditions. However, following standard practice, this step involves the assumption that the apparent fouling resistance is the same for both the test and the projected operating conditions. The implications of this assumption will be addressed in later phases of this research.

1.1.4 Characteristics of Standard Industry Approach to Test Projection

To demonstrate the operability of a heat exchanger that is tested at off-design conditions, the test fouling resistance is used in a rating-type calculation to project to the required operating conditions. This calculation differs from a standard rating calculation in that the evaluation must account for the uncertainty of the fouling resistance determined in the test. Current industry practices combine the test evaluation and test projection into a single step. However, this approach introduces errors that commonly produce an overly conservative calculation for uncertainty due to the failure to account for differences in the computational characteristics of the individual steps.

1.1.5 Role of Uncertainty Analysis for Performance Testing and Test Projection

Projection of test results for safety-related heat exchangers to design conditions must be supported by an uncertainty analysis that accounts for the uncertainty of the test measurements, fluid properties, convection coefficients, and other relevant inputs, with the objective of providing credible calculations for the uncertainty in the projected performance parameters.

The uncertainty analysis can be performed using a classical mathematical approach or iterative numerical calculations. Either approach must identify and consider each independent variable in the evaluation. However, failure to distinguish between independent and dependent variables introduces errors in the calculated uncertainty. Furthermore, the introduction of unnecessary, artificial parameters, such as the correction factor F associated with the F-LMTD method for arrangements other than ideal counterflow or parallel flow, will increase the uncertainty in calculating the overall coefficient of heat transfer and fouling resistance for performance testing and the heat duty and related performance parameters for test projection. This problem, which is caused by the F-LMTD method, has been partially addressed in the ASME and EPRI test guidelines by using the effective mean temperature difference, *EMTD*, and related classical parameters [5–7]. However, preliminary studies indicate that further improvements in the uncertainty analysis can be achieved by using the more comprehensive, unified, classical methodology.

1.2 Summary

This report provides an overview of the unified, classical heat exchanger analysis methodology and describes the use of the P-NTU_I method for performance rating. The descriptions and examples will foster an understanding to support a review of industry heat exchanger performance testing standards, which have historically been based on the F-LMTD method; eliminate undue reliance on vendor computer codes; and provide a valuable reference for utility heat exchanger engineers. The P-NTU_I and EMTD methods offer the potential to streamline the evaluation of thermal performance tests and refine the overall uncertainty associated with the projection of test results to design conditions.

1.3 Conversion Factors

Table 1-1 lists the measurements used in this report along with conversion factors to convert them to International System of Units (SI) measurements.

Table 1-1
Conversion Factors

Parameter	Measurement	SI Unit Conversion
Coefficient of heat transfer	Btu/(h ft^2 °F)	1 Btu/(h ft^2 °F) = 5.6779 W/(m^2 °C)
Density	lb_m/ft^3	1 lb_m/ft^3 = 16.018 kg/m^3
Mass flow rate	lb_m/h	1 lb_m/h = .4536 kg/h
Heat transfer rate	Btu/h	1 Btu/h = 0.293 W
Length	in. ft	1 in. = 2.54 cm 1 ft = 0.3048 m
Mass flux	lb_m/(ft^2 h)	1 lb_m/(ft^2 h) = 4.8824 kg/(m^2 h)
Specific heat	Btu/(lb_m °F)	1 Btu/(lb °F) = 4.1867 kJ/(kg °C)
Surface area	ft^2	1 ft^2 = 0.0929 m^2
Temperature	°F	°F = (9/5) °C +32
Thermal conductivity	Btu/(h ft °F)	1 Btu/(h ft °F) = 1.7306 W/(m °C)
Thermal resistance (total)	h °F/Btu	1 h °F/Btu = 1.896 °C/W
Thermal Resistance (flux)	h ft^2 °F/Btu	1 h ft^2 °F/Btu = 0.1761 m^2 °C/W
Velocity	ft/s	1 ft/s = 30.48 cm/s

1.4 Nomenclature, Symbols, and Subscripts

1.4.1 Nomenclature

A_s	Reference surface area, ft^2
A_s'	Surface area per unit length, ft
B_c	Baffle cut, %
C	Capacity rate, Btu/(h °F)
C^*	Capacitance ratio associated with $C_I = C_{\min}$
CF	Cleanliness factor defined by U/U_C
c_P	Specific heat [Btu/(lb_m °F)]
$D_{i,s}$	Shell inside diameter, in.
d_i	Tube inside diameter, in.
d_o	Tube outside diameter, in.
$EMTD$	Effective mean temperature difference, °F; equivalent equation, $q/(UA_s)$
f	Fanning friction factor
F	*LMTD* correction factor
F_f	Fouling factor
fn	Functional solution relation
fn^{-1}	Inverse functional solution relation
G_s	Shell-side mass flux defined by $\dot{m}_s/A_c$, $lb_m/(ft^2\ h)$
G_t	Tube-side mass flux defined by $\dot{m}_t/A_t$, $lb_m/(ft^2\ h)$
h	Coefficient of heat transfer, Btu/(h ft^2 °F)
k	Thermal conductivity, Btu/(h ft °F)
L, L_{eff}	Length, ft; effective length, ft
L_b	Baffle spacing (central), ft
$LMTD$	Log mean temperature difference, °F
$\dot{m}$	Mass-flow rate, lb_m/h

N	Number of tubes
N_{at}	Number of active tubes
N_b	Number of baffles
N_{tp}	Number of tube passes
NTU	Number of transfer units defined by $UA_s/C_{\min}$
NTU_I	Number of transfer units
Nus, Nut	Nusselt number defined by $h_s d_o/k_s$, $h_t d_i/k_t$
P	Effectiveness
$PLUG$	Fraction of tubes plugged
Pr	Prandtl number defined by $\mu c_P/k$
q	Heat transfer rate, Btu/h
r_f	Fouling resistance, h ft^2 °F/Btu (flux form)
r_w	Wall resistance, h ft^2 °F/Btu (flux form)
R	Capacitance ratio; Total thermal resistance, h °F/Btu
Re_s, Re_t	Reynolds number defined by $G_s d_o/\mu_s$, $G_t d_i/\mu_t$
s	Surface area, factor defined in Table 4-1
S_t	Tube pitch, in.
T_b	Bulk-stream temperature representing T_h, T_c, T_s, T_t, T_I, or $T_{II,}$ °F
U	Overall coefficient of heat transfer, Btu/(h ft^2 °F)
U_b	Bulk-stream velocity, ft/s
U_C	Clean overall coefficient of heat transfer, Btu/(h ft^2 °F)

1.4.2 Greek Symbols

δT	Absolute temperature difference, °F
δT_c	$(T_{c,o} - T_{c,i})$, °F
δT_i	$(T_{h,i} - T_{c,i})$, °F
δT_I	$\lvert T_{I,i} - T_{I,o} \rvert$, °F
δT_{II}	$\lvert T_{II,i} - T_{II,o} \rvert$, °F
δT_h	$(T_{h,i} - T_{h,o})$, °F
δT_s	$\lvert T_{s,i} - T_{s,o} \rvert$, °F
δT_t	$\lvert T_{t,i} - T_{t,o} \rvert$, °F
δ_w	Wall thickness, in.
ΔT, ΔT_1, ΔT_2	Temperature difference, °F
ε	Effectiveness associated with $C_I = C_{\min}$
ψ_{prop}	Property factor defined by Equation 4-12
$\phi_{h,t}$	Property correction factor for tube-side fluid
η_o	Net fin surface efficiency
ρ	Density, lb_m/ft^3
μ	Viscosity, $lb_m/(ft\ h)$
ν	Kinematic viscosity defined by μ/ρ, ft^2/h

1.4.3 Subscripts

ave	Average
b	Baffle spacing
c	Cold stream
cf	Counterflow
DES	Actual thermal design specifications
des	Thermal design specifications reported by manufacturer

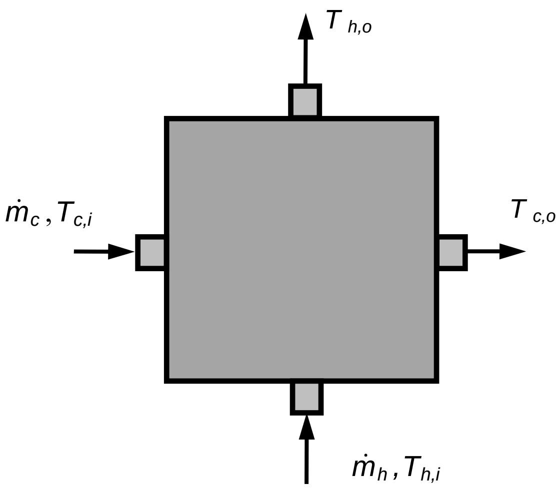
eff	Effective
h	Hot stream
i	Inlet, inside surface
I	Reference stream
II	Secondary stream
Ideal	Counterflow or parallel flow arrangements
min	Stream with smaller capacity rate
max	Stream with larger capacity rate
o	Outlet, outside surface
P	Pressure
pf	Parallel flow
ref	Reference conditions such as test, des or DES
s	Surface, shell-side
t	Tube-side
test	Test conditions
w	Wall
1, 2	Hot stream inlet and outlet temperatures

2
OVERVIEW OF UNIFIED CLASSICAL METHODOLOGY

The key relations for the unified classical approaches to heat exchanger analysis are summarized in this section [1, 2].

2.1 Frame of Reference for Classical Methods

The general perspective and notation applicable to standard shell-and-tube, double-pipe, cross-flow, and plate-and-frame heat exchangers is shown in Figure 2-1.



Key Design and Process Parameters

Arrangement and geometric characteristics

Surface areas $A_{s,o}$ and $A_{s,i}$

Reference surface area A_s

Mass-flow rates and terminal temperatures:

Hot Stream $\dot{m}_h, T_{h,i}, T_{h,o}$

Cold Stream $\dot{m}_c, T_{c,i}, T_{c,o}$

Basic Thermal Performance Parameters

Heat-transfer rate q

Effective overall coefficient of heat transfer U

Absolute temperature differences:

Inlet $\delta T_i = T_{h,i} - T_{c,i}$

Hot stream $\delta T_h = T_{h,i} - T_{h,o}$

Cold stream $\delta T_c = T_{c,o} - T_{c,i}$

Capacity rates:

$C_h = \dot{m}_h c_{P,h}$ $C_c = \dot{m}_c c_{P,c}$

Figure 2-1
General Perspective and Notation for Standard Heat Exchangers

The hot and cold stream heat-rate relations are given in Equations 2-1(a) and 2-1(b):

$$q_h = \dot{m}_h c_{P,h} (T_{h,i} - T_{h,o}) = C_h \ \delta T_h \qquad \textbf{Eq. 2-1(a)}$$

$$q_c = \dot{m}_c c_{P,c} (T_{c,o} - T_{c,i}) = C_c \ \delta T_c \qquad \textbf{Eq. 2-1(b)}$$

2.1.1 Reference Stream

Designation of *reference stream I* (hot or cold, shell- or tube-side) and *secondary stream II* (cold or hot, tube- or shell-side). Either stream can normally be designated for reference.

2.1.2 Absolute Temperature Differences, Capacity Rates, and Heat Rates

The absolute temperature differences, capacity rates, and heat rates are given by the following equations:

$$\delta T_I = |T_{I,i} - T_{I,o}| \qquad \textbf{Eq. 2-2(a)}$$

$$\delta T_{II} = |T_{II,i} - T_{II,o}| \qquad \textbf{Eq. 2-2(b)}$$

$$\delta T_i = |T_{I,i} - T_{II,i}| \qquad \textbf{Eq. 2-2(c)}$$

$$C_I = (\dot{m}c_P)_I \qquad \textbf{Eq. 2-3(a)}$$

$$C_{II} = (\dot{m}c_P)_{II} \qquad \textbf{Eq. 2-3(b)}$$

$$q = C_I\, \delta T_I = q_I \qquad \textbf{Eq. 2-4(a)}$$

$$q = C_{II}\, \delta T_{II} = q_{II} \qquad \textbf{Eq. 2-4(b)}$$

2.2 Heat Transfer Solution Relations

2.2.1 Standard Energy Balance Assumption

The standard energy balance assumption is

$$q = q_I = q_{II}$$

2.2.2 Primary Performance Parameters

The classical method solution relations involve three primary, dimensionless performance parameters: effectiveness P (sometimes designated P_I), capacitance ratio R (sometimes designated R_I), and number of transfer units NTU_I.

The standard defining relations are shown in Equations 2-5, 2-6, and 2-7:

$$P = \frac{q}{C_I \, \delta T_i} \quad \textbf{Eq. 2-5}$$

$$R = \frac{C_I}{C_{II}} \quad \textbf{Eq. 2-6}$$

$$NTU_I = \frac{UA_s}{C_I} \quad \textbf{Eq. 2-7}$$

Alternative defining relations take Equations 2-5, 2-6, and 2-7 combined with Equations 2-4(a) and 2-4(b) to obtain Equations 2-8, 2-9, and 2-10 [1, 2]:

$$P = \frac{\delta T_I}{\delta T_i} \quad \textbf{Eq. 2-8}$$

$$R = \frac{\delta T_{II}}{\delta T_I} \quad \textbf{Eq. 2-9}$$

$$NTU_I = \frac{UA_s \, \delta T_I}{q} \quad \textbf{Eq. 2-10}$$

Equations 2-5 and 2-7 provide the basis for the following practical definitions for P and NTU_I:

- The effectiveness P represents the efficiency of a heat exchanger in transferring heat relative to the reference stream thermodynamic limit. $P \leq 1$.[1]
- The number of transfer units NTU_I represents the thermal size of a heat exchanger relative to the reference capacity rate. NTU_I can be greater than or less than 1.

[1] According to Equation 2-8, the effectiveness can also represent the efficiency of a heat exchanger in increasing or decreasing the temperature of the reference fluid relative to the inlet temperature difference.

The values of effectiveness P_2, capacitance ratio R_2, and number of transfer units NTU_2 for fluid 2 specified as reference are expressed in terms of P_I, R_I, and NTU_I by Equations 2-11, 2-12, and 2-13:

$$P_2 = R_1 P_1 \quad \text{Eq. 2-11}$$

$$R_2 = 1/R_1 \quad \text{Eq. 2-12}$$

$$NTU_2 = R_1 NTU_1 \quad \text{Eq. 2-13}$$

2.2.3 Functional Solution Relations for P and NTU_I

The solution relations available in the literature that express the heat transfer rate q in terms of mass-flow rates, terminal temperature differences, reference surface area, A_s, and effective overall coefficient of heat transfer U are specified in terms of P, R, and NTU_I by functional relations of the forms shown in Equations 2-14 and 2-15:

$P = fn\ (NTU_I, R, I, \text{arrangement})$ *all standard exchangers* **Eq. 2-14**

$NTU_I = fn^{-1}\ (P, R, I, \text{arrangement})$ *explicit for classic exchangers* **Eq. 2-15**

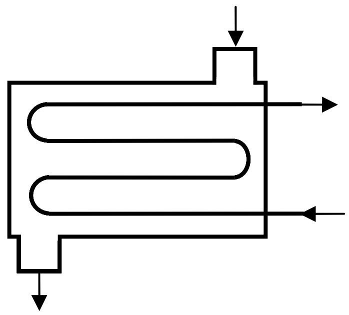
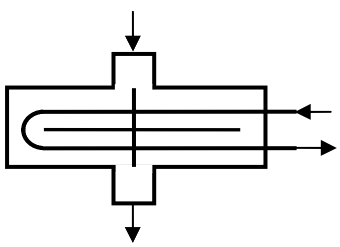
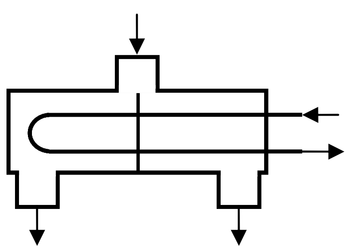
Tables 2-1 and 2-2 show the solution relations associated with classic and other representative arrangements.

Table 2-1
Functional Solution Relations: Classic Arrangements [2]

Arrangement	fn	fn^{-1}
Limiting solution for $R = 0$	$P = 1 - \exp(-NTU)$	$NTU_I = NTU = \ln[1/(1-P)]$
Counterflow	$P = \dfrac{1-\exp[-NTU_I(1-R)]}{1-R\exp[-NTU_I(1-R)]}$	$NTU_I = \dfrac{1}{1-R}\ln\dfrac{1-PR}{1-P}$
Parallel flow	$P = \dfrac{1-\exp[-NTU_I(1+R)]}{1+R}$	$NTU_I = -\dfrac{\ln[1-P(1+R)]}{1+R}$
1-2 TEMA E shell-and-tube: mixed	$P = \dfrac{2}{1+R+\sqrt{1+R^2}(1+e^{-\Gamma_1})/(1-e^{-\Gamma_1})}$ $\Gamma_1 = NTU_I\sqrt{1+R^2}$	$NTU_I = \dfrac{1}{\sqrt{1+R^2}}\ln\dfrac{2-P(1+R-\sqrt{1+R^2})}{2-P(1+R+\sqrt{1+R^2})}$
Crossflow mixed-unmixed: $C_I = C_t$ $C_I = C_s$	$P = [1-\exp(-\Gamma_2 R)]/R$ $\Gamma_2 = 1-\exp(-NTU_t)$ $P = 1-\exp(-\Gamma_2/R)$	$NTU_t = \ln\dfrac{1}{1+(1/R)\ln(1-PR)}$ $NTU_s = \dfrac{1}{R}\ln\dfrac{1}{1+R\ln(1-P)}$

TEMA is the Tubular Exchanger Manufacturers Association, Inc.

Table 2-2
Functional Solution Relations: Other Standard Arrangements [2]

Arrangement	*fn*
1-4 TEMA E Shell-and-Tube Heat Exchanger 	$C_I = C_t$ $P = \dfrac{4}{2(1+R)+\sqrt{1+4R^2}\coth(\Gamma_3/4)+\tanh(NTU_t/4)}$ $\Gamma_3 = NTU_t\sqrt{1+4R^2}$ $C_I = C_s$ $P = \dfrac{4}{2(1+R)+\sqrt{R^2+4}\coth(\Gamma_3/4)+R\tanh(R\,NTU_s/4)}$ $\Gamma_3 = NTU_s\sqrt{R^2+4}$
1-2 TEMA G Shell-and-Tube Heat Exchanger 	$C_I = C_t$ $P = \dfrac{1}{R}\left(\dfrac{B-\alpha^2}{A+B/R+2}\right)$ $A = -2(1-\alpha)^2/(2R+1)$ $B = [4-\beta(2+1/R)]/(2-1/R)$ $\alpha = \exp[-NTU_t(2R+1)/4]$ $\beta = \exp[-NTU_t(2R-1)/2]$ $C_I = C_s$ $P = \dfrac{B-\alpha^2}{A+BR+2}$ $A = -2R(1-\alpha)^2/(2+R)$ $B = [4-\beta(2+R)]/(2-R)$ $\alpha = \exp[-NTU_s(2+R)/4]$ $\beta = \exp[-NTU_s(2-R)/2]$
1-2 TEMA J Shell-and-Tube Heat Exchanger 	$B = (A^\lambda + 1)/(A^\lambda - 1)$ $C = A^{(1+\lambda)/2}/[\lambda - 1 + (1+\lambda)A^\lambda]$ $D = 1 + \lambda A^{(\lambda-1)/2}/(A^\lambda - 1)$ $C_I = C_t$ $P = \dfrac{1}{R}\left[\dfrac{1}{1+\lambda B-2\lambda CD+1/(2R)}\right]$ $A = \exp(R\,NTU_t)$ $\lambda = \sqrt{1+1/(4R^2)}$ $C_I = C_s$ $P = \dfrac{1}{1+\lambda B-2\lambda CD+R/2}$ $A = \exp(NTU_s)$ $\lambda = \sqrt{1+R^2/4}$

2.3 Classical Heat Exchanger Performance Analysis Methods

Equations 2-2 through 2-15 provide the basis for the classical P-NTU$_I$, ε-NTU, F-LMTD, and EMTD methods for heat exchanger performance analysis.

2.3.1 P-NTU$_I$ Method

The heat transfer rate is specified by Equation 2-16:

$$q = PC_I\,\delta T_i \qquad \textbf{Eq. 2-16}$$

2.3.2 ε-NTU Method

The stream with lower capacity rate is specified as the reference stream. The heat transfer rate is specified by Equation 2-17:

$$q = \varepsilon C_{\min}\,\delta T_i \qquad \textbf{Eq. 2-17}$$

with $C_I = C_{\min}$ and P, R, and NTU_I designated by ε, C*, and NTU, respectively.

2.3.3 F-LMTD Method

The heat transfer rate is specified by Equation 2-18:

$$q = UA_s\,F\,LMTD_{cf} \qquad \textbf{Eq. 2-18}$$

where the correction factor F is expressed in terms of the primary performance parameters by

$$F = \frac{NTU_{I,cf}}{NTU_I} = \frac{1}{NTU_I(1-R)}\ln\frac{1-PR}{1-P} \qquad \textbf{Eq. 2-19}$$

and the log mean temperature difference, $LMTD$, is defined in terms of terminal temperature differences by Equation 2-20:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \qquad \textbf{Eq. 2-20}$$

where

$$LMTD = LMTD_{cf} \qquad \textbf{Eq. 2-21(a)}$$

$$\Delta T_1 = T_{h,i} - T_{c,o} \qquad \textbf{Eq. 2-21(b)}$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \qquad \textbf{Eq. 2-21(c)}$$

for ideal counterflow.

Equation 2-18 is the more general form to Equation 2-22:

$$q = UA_s\, LMTD \qquad \textbf{Eq. 2-22}$$

where:

$LMTD = LMTD_{cf}$ for ideal counterflow, and

$$LMTD = LMTD_{pf} \qquad \textbf{Eq. 2-23(a)}$$

$$\Delta T_1 = T_{h,i} - T_{c,i} \qquad \textbf{Eq. 2-23(b)}$$

$$\Delta T_2 = T_{h,o} - T_{c,o} \qquad \textbf{Eq. 2-23(c)}$$

for ideal parallel flow. $F\text{-}LMTD_{cf}$ corresponds to *EMTD*.

2.3.4 EMTD Method

The heat transfer rate is specified by Equation 2-24:

$$q = UA_s\, EMTD \qquad \textbf{Eq. 2-24}$$

where the effective mean temperature difference *EMTD* is expressed in terms of the primary performance parameters and terminal temperature differences, δT_I or δT_i, by Equations 2-25 and 2-26, which are based on Equation 2-10 [1, 2].

$$EMTD = \frac{\delta T_I}{NTU_I} \qquad \textbf{Eq. 2-25}$$

$$EMTD = \frac{P\, \delta T_i}{NTU_I} \qquad \textbf{Eq. 2-26}$$

2.4 Characteristics of Classical Rating, Testing, and Design

2.4.1 Standard Rating

Independent performance parameters are calculated by using Equations 2-6 and 2-7.[2]

$$R = C_I / C_{II} \qquad \textbf{Eq. 2-6}$$

$$NTU_I = UA_s / C_I \qquad \textbf{Eq. 2-7}$$

P is calculated by use of functional solution relations of the form used in Equation 2-14:

$$P = fn\ (NTU_I,\ R)\ \textit{explicit for all standard arrangements} \qquad \textbf{Eq. 2-14}$$

The heat transfer rate q is calculated by using Equation 2-16,

$$q = PC_I\,\delta T_i \qquad \textbf{Eq. 2-16}$$

or, equivalently, by using Equations 2-18 and 2-24:

$$q = UA_s\ F\ LMTD_{cf} \qquad \textbf{Eq. 2-18}$$

$$q = UA_s\ EMTD \qquad \textbf{Eq. 2-24}$$

Outlet temperatures $T_{h,o}$ and $T_{c,o}$ are calculated by Equations 2-27(a) and 2-27(b), (based on Equations 2-1(a) and 2-1(b) for q):

$$T_{h,o} = T_{h,i} - q/C_h \qquad \textbf{Eq. 2-27(a)}$$

$$T_{c,o} = T_{c,i} + q/C_c \qquad \textbf{Eq. 2-27(b)}$$

[2] The F-LMTD method traditionally features the consideration of P and R as the independent performance parameters.

2.4.2 Standard Testing and Design

Independent performance parameters are calculated by using Equations 2-8 and 2-9:

$$P = \frac{\delta T_I}{\delta T_i} \qquad \textbf{Eq. 2-8}$$

$$R = \frac{\delta T_{II}}{\delta T_I} \qquad \textbf{Eq. 2-9}$$

NTU_I is calculated by use of inverse functional relations in the form of Equation 2-15:

$$NTU_I = fn^{-1}(P, R) \textit{ explicit for classic arrangement; iterative for others} \qquad \textbf{Eq. 2-15}$$

UA_s is calculated by use of Equation 2-28, (based on Equation 2-7 for NTU_I),

$$UA_s = NTU_I \, C_I \qquad \textbf{Eq. 2-28}$$

or equivalently by use of Equation 2-29 or 2-30, (based on Equation 2-18 or 2-24 for q),

$$UA_s = \frac{q}{F \; LMTD} \qquad \textbf{Eq. 2-29}$$

$$UA_s = \frac{q}{EMTD} \qquad \textbf{Eq. 2-30}$$

Heat rates q_h and q_c are calculated by using Equations 2-1(a) and 2-1(b), which provide the basis for establishing the heat transfer rate, q:

$$q_h = C_h \, \delta T_h \qquad \textbf{Eq. 2-1(a)}$$

$$q_c = C_c \, \delta T_c \qquad \textbf{Eq. 2-1(b)}$$

2.5 Advantages and Limitations of Classical Methods

2.5.1 P-NTU$_I$ Method

The P-NTU$_I$ method provides the most straightforward and flexible approach to rating standard heat exchangers and complex arrangements involving multiple heat exchangers. It also provides a practical approach for performance testing and design. However, the standard P-NTU$_I$ method introduces an analytical error as a result of a formulation-solution inconsistency for six-point testing.

2.5.2 ε-NTU Method

The ε-NTU method, which represents a special form of the P-NTU_I method, has the same characteristics as the P-NTU_I method. However, this method restricts analysis flexibility and masks the relation between the effectiveness, ε, and the primary classical performance parameters, P, R, and NTU_I, as well as F and *EMTD*.

2.5.3 F-LMTD Method

The F-LMTD method has been successfully used for performance analysis for the past 80 years. However, this popular method unnecessarily complicates performance rating by introducing supplementary parameters F and $LMTD_{cf}$ into the solution relations. Use of the traditional form of the F-LMTD method for basic rating requires iteration on the unspecified terminal temperatures; the use of other classical methods is considerably less involved. The F-LMTD method also complicates performance analysis of component cooling systems and other systems involving multiple heat exchangers, and it has undesirable effects on uncertainty analysis.

2.5.4 EMTD Method

The EMTD method involves the specification of *EMTD* in terms of the primary performance parameters, P, R, and NTU_I, thereby eliminating the need for introducing the correction factor F and $LMTD_{cf}$. The EMTD method represents a simpler and more straightforward mean temperature difference approach than does the F-LMTD method. The EMTD method also eliminates the analytical error associated with use of the standard effectiveness methods for six-point performance testing. As in the case of the P-NTU_I method, the EMTD method can be readily used for rating, testing, and design.

2.6 Summary

The characteristics of the classical methods are such that the most straightforward and flexible, unified classical approach to heat exchanger analysis involves the use of the P-NTU_I method for performance rating and the EMTD method for performance testing and design. However, aside from the introduction of analytical errors in the use of the effectiveness methods for six-point testing and computational limitations associated with the F-LMTD method for some applications involving rating, testing, and design, the effectiveness, F-LMTD, and EMTD methods give rise to the same solution results.

3

CLASSICAL PERFORMANCE RATING: P-NTU$_I$ METHOD, PROCEDURAL STEPS

The procedural steps for standard performance rating by use of the P-NTU_I method are the following:

1. (a) Specify geometric characteristics and reference surface area, A_s.

   (b) Specify mass-flow rates, inlet temperatures, fouling factors, or cleanliness factor.

   (c) Estimate the fluid properties (ρ_h, $c_{P,h}$, k_h, μ_h, ρ_c, $c_{P,c}$, k_c, μ_c) and specify k_w.

   (d) Calculate δT_i, C_h, and C_c.

2. Calculate h_h, f_h, h_c, f_c, wall/fouling resistance, U_C, and U.

3. Specify reference stream I, set C_I and C_{II}, and calculate basic performance parameters.

   (a) $NTU_I = UA_s/C_I$ and $R = C_I/C_{II}$

   (b) $P = fn\ (NTU_I, R)$[3]

4. Calculate heat transfer rate q using[4]

   $q = PC_I\ \delta T_i$

5. Calculate unspecified terminal temperatures by using Equations 2-27(a) and 2-27(b).

6. Iterate to refine properties if necessary.

---

[3] Optional calculations:
$EMTD = P\ \delta T_i/NTU_I$.
F correction factor: $\Delta T_1 = T_{h,i} - T_{c,o}$, $\Delta T_2 = T_{h,o} - T_{c,i}$.
$NTU_{I,cf} = (1-R)^{-1} \ln [(1-PR)/(1-P)]$ $\qquad F = NTU_{I,cf}/NTU_I$
$\Delta T_1 = T_{h,i} - T_{c,o}$, $\Delta T_2 = T_{h,o} - T_{c,i}$. $\qquad LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln (\Delta T_1/\Delta T_2)$

[4] Equivalent $EMTD$ method: $q = UA_s\ EMTD$. Equivalent F-$LMTD$ method: $q = UA_s\ F\ LMTD_{cf}$.

4

EFFECTIVE OVERALL COEFFICIENT OF HEAT TRANSFER: PRACTICAL RELATIONS

4.1 General Relations for *U*, U_C, and Wall/Fouling Resistances

Table 4–1 summarizes key relations for U, U_C, and wall/fouling resistances.

Table 4-1
Effective Overall Coefficient of Heat Transfer: Summary of Key Relations [2]

Standard Relations: General Forms		
Effective overall coefficient of heat transfer	$\frac{1}{U} = \frac{1}{U_C} + r_f$	$\frac{1}{U_C} = \frac{s_h}{h_h} + r_w + \frac{s_c}{h_c}$
Plain surface area factors	$s_h = A_s / A_{s,h} \quad s_c = A_s / A_{s,c}$	
Finned surface area factors	$s_h = \frac{A_s / A_{s,h}}{\eta_{o,h}} \quad s_c = \frac{A_s / A_{s,c}}{\eta_{o,c}}$	
Fouling resistance: specified fouling factors or cleanliness factor, $CF = U/U_c$	$r_f = s_h F_{f,h} + s_c F_{f,c}$	$r_f = (1 - CF) / U$
Standard Relations for Clean Overall Coefficient of Heat Transfer Representative Particular Forms (Unfinned Surfaces)		
Plane wall	$\frac{1}{U_C} = \frac{1}{h_h} + r_w + \frac{1}{h_c}$	
Thermal wall resistance		$r_w = A_s R_W$
Tube wall, arbitrary reference surface	$A_s = A_{s,o} \quad \frac{1}{U_C} = \frac{A_s / A_{s,i}}{h_i} + r_w + \frac{A_s / A_{s,o}}{h_o}$	$r_w = \frac{d_s}{2k_w} \ln \frac{d_o}{d_i}$
	$A_s = A_{s,o} \quad \frac{1}{U_C} = \frac{d_o/d_i}{h_i} + r_w + \frac{1}{h_o}$	$r_w = \frac{d_o}{2k_w} \ln \frac{d_o}{d_i}$
	$A_s = A_{s,I} \quad \frac{1}{U_C} = \frac{1}{h_i} + r_w + \frac{d_i/d_o}{h_o}$	$r_w = \frac{d_i}{2k_w} \ln \frac{d_o}{d_i}$

4.2 Calculation of Tube-Side Convection Coefficients f_t and h_t: Procedural Steps—Fully Turbulent Flow in Smooth Tubes

Calculate the following tube-side parameters:

1. Flow area A_t:

$$A_t = (\pi d_i^2/4)(N/N_{tp}) \qquad \textbf{Eq. 4-1}$$

2. Mass flux G_t and bulk-stream velocity $U_{b,t}$:

$$G_t = \dot{m}_t/A_t \qquad \textbf{Eq. 4-2}$$

$$U_{b,t} = G_t/\rho_t \qquad \textbf{Eq. 4-3}$$

3. Reynolds number Re_t and Prandtl number Pr_t:

$$Re_t = G_t d_i/\mu_t = U_{b,t} d_i/\nu_t \qquad \textbf{Eq. 4-4}$$

$$Pr_t = \mu_t c_{P,t}/k_t \qquad \textbf{Eq. 4-5}$$

4. Nusselt number Nu_t and friction factor f_t using credible correlations such as those shown in Equations 4-4 and 4-5 were developed by Petukhov et al.[5] to approximate fully developed, fully turbulent, and uniform property conditions [9, 10]:

$$Nu_t = \frac{(f_t/2)\, Re_t\, Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} \qquad \textbf{Eq. 4-6}$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} \qquad \textbf{Eq. 4-7}$$

5. Convection coefficient h_t, with the property correction factor $\phi_{h,t}$ specified by standard correlations [2]:

$$h_t = (Nu_t\, k_t/d_i)\phi_{h,t} \qquad \textbf{Eq. 4-8}$$

[5] The standard deviation error for the Petukhov convection correlation is ±6% for $10^4 < Re_t < 5 \times 10^6$ and $0.5 < Pr_t < 200$, and 10% for $200 < Pr_t < 2000$ over the same Re_t range. The following, more conservative relation recommended by IHS ESDU is among the alternative correlations that are commonly used for the Nusselt number associated with fully turbulent tube flow [11]:

$$Nu_t = 0.0225\, Re_t^{0.795}\, Pr_t^{0.495} \exp\left[0.0225 (\ln Pr_t)^2\right]$$

for $4 \times 10^4 < Re_t < 10^6$ and $0.3 < Pr_t < 300$. This equation lies about 8% below the Petukhov correlation.

4.3 Specification of Shell-Side Convection Coefficients: Practical Considerations

Due to the level of uncertainty normally associated with the standard methods of specifying the shell-side convection coefficient h_s for shell-and-tube heat exchangers, it is recommended that h_s be calculated using empirical convection correlations based on test measurements obtained immediately after installation or cleaning, whenever feasible. As a practical alternative for applications in which test measurements are not available, it is common practice to evaluate h_s on the basis of the manufacturer design specifications.[6] This approach is used in Section 5, Examples: Performance Rating Applications—The P-NTU$_\mathrm{I}$ Method, to illustrate the P-NTU$_\mathrm{I}$ method and the unified classical perspective. Heat exchangers are commonly overdesigned beyond what is required by the specified fouling factors; therefore, back-calculation of h_s based on values of U_{des} or q_{des} reported on the Tubular Exchanger Manufacturers Association (TEMA) specification sheet for specified effective surface area $A_{s,\mathrm{eff}}$ should be expected to be overly conservative. Examples 3 through 6 in Section 5 use an approach that accounts for overdesign in the back-calculation of h_s.

4.4 Back-Calculation of Reference Convection Coefficient $h_{s,ref}$, Based on U_{ref}

Back-calculation of the reference convection coefficient $h_{s,\mathrm{ref}}$ based on reference overall coefficient of heat transfer U_{ref} and fouling resistance $r_{f,\mathrm{ref}}$ involves the use of the following relations:

$$\frac{1}{U_{C,\mathrm{ref}}} = \frac{1}{U_{\mathrm{ref}}} - r_{f,\mathrm{ref}} \qquad \textbf{Eq. 4-9}$$

$$\frac{1}{h_{s,\mathrm{ref}}} = \frac{1}{s_s}\left(\frac{1}{U_{C,\mathrm{ref}}} - \frac{s_t}{h_{t,\mathrm{ref}}} - r_w\right) \qquad \textbf{Eq. 4-10}$$

[6] Design specifications provided by heat exchanger manufacturers normally are based on proprietary computer codes that feature the use of the stream analysis method with specified values of clearances (shell-to-bundle, shell-to-baffle, and tube-to-baffle), as well as other geometric characteristics.

4.5 Back-Calculation of Reference Convection Coefficient $h_{s,\text{ref}}$, Based on q_{ref}

For applications in which $h_{s,\text{ref}}$ is to be calculated on the basis of specified reference heat transfer rate q_{ref}, it is recommended that the following general steps be taken to evaluate U_{ref}, and that Equations 4-9 and 4-10 be used to calculate $h_{s,\text{ref}}$:

1. Calculate temperature differences δT_t and δT_s based on energy balance requirement.
2. Calculate performance parameters P and R.
3. Calculate $NTU_I = fn^{-1}(P, R)$.
4. Calculate $U_{\text{ref}} = NTU_I\, C_I/A_s$ or $U_{\text{ref}} = q_{\text{ref}}/(A_s\, EMTD)$.

4.6 Calculation of Convection Coefficient h_s: Change in Operating Conditions

A practical, approximate approach to characterizing the shell-side thermal characteristics for turbulent flow, moderate variations in mass-flow rate, and properties provides the basis for specifying h_s in terms of a reference value $h_{s,\text{ref}}$ (that is, $h_{s,\text{test}}$, $h_{s,\text{des}}$) by a relation of the form shown in Equation 4-11 [2]:

$$h_s = \psi_{\text{prop}} \left(\dot{m}_s/\dot{m}_{s,\text{ref}}\right)^{0.6} h_{s,\text{ref}} \qquad \textbf{Eq. 4-11}$$

where the property factor ψ_{prop} is defined by Equation 4-12:

$$\psi_{\text{prop}} = \left(\frac{\mu_{s,\text{ref}}}{\mu_s}\right)^{0.6} \left(\frac{Pr_s}{Pr_{s,\text{ref}}}\right)^{1/3} \frac{k_s}{k_{s,\text{ref}}} \qquad \textbf{Eq. 4-12}$$

Calculations based on the comprehensive stream analysis method indicate a variation in h_s with change in operating conditions that tends to be larger than specified by Equations 4-11 and 4-12. With this point in mind, and considering the nature of the sensitivity in the calculation of heat transfer rate q to the error in h_s, the decision whether a higher-order shell-side convection model or more extensive testing is needed to evaluate the effects of change in operating conditions on h_s should involve a comprehensive uncertainty analysis for critical applications.

5
EXAMPLES: PERFORMANCE RATING APPLICATIONS

This section presents the following six examples:

EXAMPLE 1. Diesel generator cooler: TEMA 1-1 E

- Check of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for plugged tubes
- Rating-calculation for decrease in cooling water mass-flow rate
- Rating-calculation for increase in shell-side mass-flow rate

EXAMPLE 2. Diesel generator cooler: TEMA 1-2 E

- Check of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for plugged tubes
- Rating-calculation for increase in cooling water mass-flow rate

EXAMPLE 3. Component cooling water heat exchanger: TEMA 1-2 F

- Confirmation of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for plugged tubes
- Rating-calculation for increase in cooling water temperature

EXAMPLE 4. Turbine building cooling water heat exchanger: TEMA 1-4 G

- Confirmation of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for plugged tubes

EXAMPLE 5. Decay heat removal heat exchanger: TEMA 1-2 J

- Confirmation of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for off-design conditions

EXAMPLE 6. Lube oil cooler: TEMA 1-4 E, plain tube

- Confirmation of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for plugged tubes

5.1 Example 1. Diesel Generator Cooler: TEMA 1-1 E

A TEMA 1-1 E shell-and-tube heat exchanger is used to cool a diesel generator. Design and thermal performance specifications for the heat exchanger are shown in Figure 5-1 and Tables 5-1 and 5-2. The P-NTU$_I$ method is to be used to develop the following performance analysis calculations:

- Check of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for 10% plugged tubes
- Rating-calculation for decrease in cooling water mass-flow rate to 8.3629×10^5 lb$_m$/h
- Rating-calculation for increase in shell-side mass-flow rate to 9.7385×10^5 lb$_m$/h

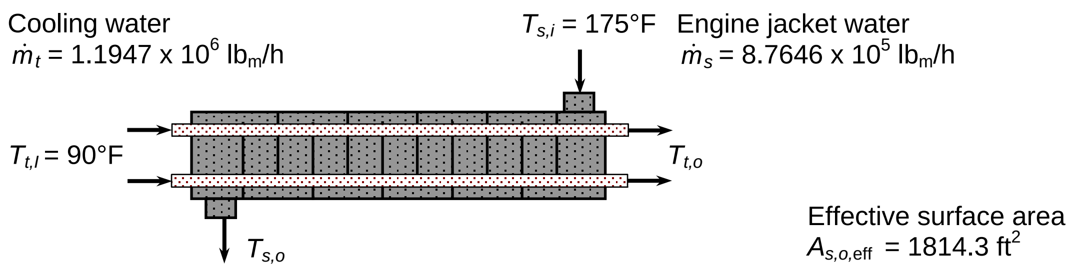


Figure 5-1
Diesel Generator Cooler Design Conditions

Table 5-1
Design Specifications: 1-1 E Shell and Tube

Tubing	Shell-Side Characteristics
Admiralty brass	Triangular arrangement
k_w = 63.1 Btu/(h ft °F)	N = 420
L_{eff} = 22 ft	S_t = 0.9375 in.
d_i = 0.652 in.	$D_{i,s}$ = 23.25 in., L_b = 24 in.
d_o = 0.75 in.	N_b = 10, B_c = 20%, single-segmental baffles

Table 5-2
Thermal Performance Specifications: 1-1 E Shell and Tube

Thermal Performance Specifications	
$F_{f,t}$ = 0.002 h ft^2 °F/Btu	$T_{t,i,\text{des}}$ = 90°F
$F_{f,s}$ = 0.001 h ft^2 °F/Btu	$T_{t,o,\text{des}}$ = 110°F
q_{des} = 23.748 MBtu/h	$T_{t,\text{ave}}$ = 100°F
U_{des} = 212 Btu/(h ft^2 °F)	$T_{s,i,\text{des}}$ = 175°F
$EMTD_{\text{des}}$ = 61.7°F	$T_{s,o,\text{des}}$ = 148.6°F
	$T_{s,ave}$ = 161.8°F

5.1.1 Check of Thermal Performance Specifications

- Reference surface: The outer surface is selected for reference.

  $A_s = A_{s,o} = N(\pi d_o L_{\text{eff}}) = 420[\pi(0.75\text{ ft}/12)](22\text{ ft}) = 1814.3\text{ ft}^2$

- Properties: k_w = 63.1 Btu/(h ft °F) for admiralty brass.

  Fluid properties are specified at average design specification temperatures:

Tube-side at $T_{t,ave}$ = 100°F	Shell-side at $T_{s,ave}$ = 161.8°F
ρ_t = 61.905 lb$_\text{m}$/ft^3	ρ_s = 60.92 lb$_\text{m}$/ft^3
$c_{P,t}$ = 0.998 Btu/(lb$_\text{m}$ °F)	$c_{P,s}$ = 1.003 Btu/(lb$_\text{m}$ °F)
k_t = 0.3587 Btu/(h ft °F)	k_s = 0.3860 Btu/(h ft °F)
μ_t = 1.648 lb$_\text{m}$/(ft h)	μ_s = 0.9067 lb$_\text{m}$/(ft h)
Pr_t = 4.585	Pr_s = 2.356

- Inlet temperature difference and capacity rates:

  $\delta T_i = T_{s,i} - T_{t,i} = 175°\text{F} - 90°\text{F} = 85°\text{F}$

  $C_t = (\dot{m}\, c_P)_t = (1.1947 \times 10^6\text{ lb}_\text{m}/\text{h})[0.9980\text{ Btu}/(\text{lb}_\text{m}\text{ °F})] = 1.1923\text{ MBtu}/(\text{h °F})$

  $C_s = (\dot{m}\, c_P)_s = (8.7646 \times 10^5\text{ lb}_\text{m}/\text{h})[1.003\text{ Btu}/(\text{lb}_\text{m}\text{ °F})] = 0.8791\text{ MBtu}/(\text{h °F})$

- Basic check of thermal performance specifications:

$\delta T_{t,\text{des}} = T_{t,o,\text{des}} - T_{t,i,\text{des}} = 20°\text{F}$ $\quad q_{t,\text{des}} = C_t\, \delta T_{t,\text{des}} = 23.85$ MBtu/h

$\delta T_{s,\text{des}} = T_{s,i,\text{des}} - T_{s,o,\text{des}} = 26.4°\text{F}$ $\quad q_{s,\text{des}} = C_s\, \delta T_{s,\text{des}} = 23.21$ MBtu/h

$q_{\text{ave}} = (q_{t,\text{des}} + q_{s,\text{des}})/2 = 23.53$ MBtu/h

$q_{EMTD} = U_{\text{des}} A_s\, EMTD_{\text{des}} = [212 \text{ Btu/(h ft}^2\ °\text{F)}]\,(1814.3 \text{ ft}^2)(61.7°\text{F}) = 23.73$ MBtu/h

The calculations differ by 0.94% for q_{des} and q_{ave} and by 0.07% for q_{EMTD} and q_{des}, which indicates consistency in the specifications for q_{des}, U_{des}, $EMTD_{\text{des}}$, $T_{t,o,\text{des}}$, and $T_{s,o,\text{des}}$.

- Overall coefficient of heat transfer:

Set $U = U_{\text{des}} = 212 \text{ Btu/(h ft}^2\ °\text{F)}$

- Classical approach: Shell-side fluid is selected for reference (see note 5-1 in Section 5.1.6).

$C_I = C_s = 0.8791$ MBtu/(h °F) $\quad C_{II} = C_t = 1.1923$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_s = \frac{UA_s}{C_s} = \frac{[212 \text{ Btu/(h ft}^2\ °\text{F)}](1814.3 \text{ ft}^2)}{0.8791 \text{ MBtu/(h °F)}} = 0.43753$$

$$R = \frac{C_I}{C_{II}} = \frac{C_s}{C_t} = \frac{0.8791 \text{ MBtu/(h °F)}}{1.1923 \text{ MBtu/(h °F)}} = 0.73729$$

- Effectiveness: 1-1 E exchanger from Table 2-1.

$$P = \frac{1 - \exp\,[-NTU_I\,(1-R)]}{1 - R\,\exp\,[-NTU_I\,(1-R)]} = 0.31678$$

- Heat transfer rate, P-NTU_I method:[7]

$$q = PC_I\,\delta T_i = 0.31678\,[0.8791 \text{ MBtu/(h °F)}]\,(85°\text{F}) = 23.67 \text{ MBtu/h}$$

[7] EMTD method: $q = UA_s\,EMTD = [212 \text{ Btu/(h ft}^2\ °\text{F)}](1814.3 \text{ ft}^2)(61.54°\text{F}) = 23.67$ MBtu/h

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = T_{t,i} + \frac{q}{C_t} = 90°\text{F} + \frac{23.67\ \text{MBtu/h}}{1.1923\ \text{MBtu/(h °F)}} = 109.85°\text{F} \qquad T_{t,\text{ave}} = 99.95°\text{F}$$

$$T_{s,o} = T_{s,i} - \frac{q}{C_s} = 175°\text{F} - \frac{23.67\ \text{MBtu/h}}{0.8791\ \text{MBtu/(h °F)}} = 148.07°\text{F} \qquad T_{s,\text{ave}} = 161.5°\text{F}$$

- Effective mean temperature difference:[8]

$$EMTD = \frac{P\ \delta T_i}{NTU_I} = \frac{(0.31678)(85°\text{F})}{0.43753} = 61.54°\text{F}$$

The calculations differ from the thermal performance specifications by 0.34% for q, 0.26% for *EMTD*, 0.14% for $T_{t,o}$, and 0.36% for $T_{s,o}$.

Conclusion: The classical calculations for design conditions are consistent with the thermal performance specifications provided by the manufacturer.

[8] F correction factor: $\Delta T_1 = 175°\text{F} - 109.85°\text{F} = 65.15°\text{F}$, $\Delta T_2 = 148.07°\text{F} - 90°\text{F} = 58.07°\text{F}$.
$NTU_{I,cf} = (1-R)^{-1} \ln[(1-PR)/(1-P)] = 0.43755$ $\qquad F = NTU_{I,cf}/NTU_I = 1.0$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 61.54°\text{F}$ $\qquad F\ LMTD_{cf} = 61.54°\text{F} \equiv EMTD$

5.1.2 Back-Calculation of $h_{s,des}$

The shell-side convection coefficient $h_{s,\mathrm{des}}$ is back-calculated based on the specified value of U_{des}:

- Wall and fouling resistances: $s_t = d_o/d_i = 0.75/0.652 = 1.1503$, $s_s = 1$.

$$r_w = \frac{d_o}{2k_w} \ln\frac{d_o}{d_i} = \frac{0.75 \text{ ft}/12}{2\ [63.1 \text{ Btu/(h ft °F)}]} \ln\frac{0.75}{0.652} = 0.00006935 \text{ h ft}^2 \text{ °F/Btu}$$

$$r_f = s_t F_{f,t} + s_s F_{f,s} = 1.1503(0.002) + 0.001 = 0.0033006 \text{ h ft}^2 \text{ °F/Btu}$$

- Clean overall coefficient of heat transfer:

$$\frac{1}{U_{C,\mathrm{des}}} = \frac{1}{U_{\mathrm{des}}} - r_f = \frac{1}{212} - 0.0033006 = 0.0014164 \text{ h ft}^2 \text{ °F/Btu}$$

$$U_{C,\mathrm{des}} = 706.0 \text{ Btu/(h ft}^2 \text{ °F)}$$

- Tube-side convection coefficient $h_{t,\mathrm{des}}$:

$$A_t = \frac{\pi d_i^2}{4}\frac{N}{N_{tp}} = \frac{\pi(0.652 \text{ ft}/12)^2}{4}\frac{420}{1} = 0.9738 \text{ ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{1.1947\times10^6 \text{ lb}_\mathrm{m}/\text{h}}{0.9738 \text{ ft}^2} = 1.227\times10^6 \text{ lb}_\mathrm{m}/(\text{ft}^2 \text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 5.505 \text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.227\times10^6 \text{ lb}_\mathrm{m}/(\text{ft}^2 \text{ h})](0.652 \text{ ft}/12)}{1.648 \text{ lb}_\mathrm{m}/(\text{ft h})} = 40{,}450$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} = 0.005503$$

$$Nu_t = \frac{(f_t/2)\, Re_t\, Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = \frac{(0.005503/2)(40{,}450)(4.585)}{1.07 + 12.7\sqrt{0.005503/2}\,(4.585^{2/3} - 1)} = 227.6$$

$$h_{t,\mathrm{des}} = Nu_t \frac{k_t}{d_i} = 227.6\, \frac{0.3587 \text{ Btu/(h ft °F)}}{0.652 \text{ ft}/12} = 1502.5 \text{ Btu/(h ft}^2 \text{ °F)}$$

- Shell-side convection coefficient $h_{s,des}$ (see Note 5-2 in Section 5.1.6):

$$\frac{1}{h_{s,\mathrm{des}}} = \frac{1}{s_s}\left(\frac{1}{U_{C,\mathrm{des}}} - \frac{s_t}{h_{t,\mathrm{des}}} - r_w\right) = \frac{1}{U_{C,\mathrm{des}}} - \frac{s_t}{h_{t,\mathrm{des}}} - r_w$$

$$= 0.0014164 - \frac{1.150}{1502.5} - 0.00006935 = 0.0005814 \text{ h ft}^2\ {}^\circ\text{F/Btu}$$

$h_{s,\mathrm{des}} = 1720$ Btu/(h ft^2 °F)

5.1.3 Rating Calculation for 10% Plugged Tubes

The properties are assumed to be unchanged as a first approximation, so that the capacity rates, shell-side convection coefficient, and wall/fouling resistances are specified by the design-point values.

- Number of active tubes:

$$N_{at} = N(1 - PLUG) = 420(1 - 0.1) = 378 \qquad \text{(10\% decrease)}$$

- Reference surface area:

$$A_s = A_{s,o} = N_{ta}(\pi d_o L) = 378[\pi(0.75 \text{ ft}/12)](22 \text{ ft}) = 1632.8 \text{ ft}^2 \qquad \text{(10\% decrease)}$$

- Tube-side convection coefficient h_t:

$$A_t = \frac{\pi d_i^2}{4}\frac{N_{at}}{N_{tp}} = \frac{\pi(0.652 \text{ ft}/12)^2}{4}\frac{378}{1} = 0.8764 \text{ ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{1.195\times10^6 \text{ lb}_\text{m}/\text{h}}{0.8764 \text{ ft}^2} = 1.363 \times 10^6 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 6.117 \text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.363\times10^6 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})](0.652 \text{ ft}/12)}{1.648 \text{ lb}_\text{m}/(\text{ft h})} = 44{,}940$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} = 0.005370$$

$$Nu_t = \frac{(f_t/2)\, Re_t\, Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 248.3$$

$$h_t = Nu_t \frac{k_t}{d_i} = 248.3 \frac{0.3587 \text{ Btu/(h ft °F)}}{0.652 \text{ ft}/12} = 1639 \text{ Btu/(h ft}^2 \text{ °F)} \qquad \text{(9.08\% increase)}$$

- Shell-side convection coefficient h_s:

$$h_s = h_{s,\text{des}} = 1720 \text{ Btu/(h ft}^2 \text{ °F)} \qquad \text{(essentially unchanged)}$$

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.150}{1639} + 0.00006935 + \frac{1}{1720} = 0.0013524 \text{ h ft}^2 \text{ °F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.0013524 + 0.00330 = 0.0046524 \text{ h ft}^2 \text{ °F/Btu}$$

$U_C = 739.4$ Btu/(h ft^2 °F) $\quad$ $U = 214.9$ Btu/(h ft^2 °F) $\quad$ (1.37% increase)

- Classical approach: Shell-side fluid is retained as reference.

$C_I = C_s = 0.8791$ MBtu/(h °F) $\quad$ $C_{II} = C_t = 1.1923$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_s = \frac{UA_s}{C_s} = \frac{[214.9 \text{ Btu/(h ft}^2 \text{ °F)}](1632.8 \text{ ft}^2)}{0.87901 \text{ MBtu/(h °F)}} = 0.39918$$

$$R = \frac{C_I}{C_{II}} = \frac{C_s}{C_t} = \frac{0.8791 \text{ MBtu/(h °F)}}{1.1923 \text{ MBtu/(h °F)}} = 0.73729 \qquad \text{(essentially unchanged)}$$

- Effectiveness: 1-1 E heat exchanger from Table 2-1.

$$P = \frac{1 - \exp[-NTU_I(1-R)]}{1 - R\exp[-NTU_I(1-R)]} = 0.2962$$

- Heat transfer rate: P-NTU$_I$ method.[9]

$$q = PC_I\,\delta T_i = 0.2962\,[0.8791 \text{ MBtu/(h °F)}](85\text{°F}) = 22.13 \text{ MBtu/h} \qquad \text{(6.51\% decrease)}$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 90\text{°F} + \frac{22.13 \text{ MBtu/h}}{1.1923 \text{ MBtu/(h °F)}} = 108.56\text{°F} \qquad T_{t,ave} = 99.28\text{°F}$$

$$T_{s,o} = 175\text{°F} - \frac{22.13 \text{ MBtu/h}}{0.8791 \text{ MBtu/(h °F)}} = 149.82\text{°F} \qquad T_{s,\mathrm{ave}} = 162.41\text{°F}$$

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[9] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.2962(85\text{°F})/0.39918 = 63.07\text{°F}$ (2.49% increase)
$q = UA_s\,EMTD = [214.9 \text{ Btu/(h ft}^2 \text{ °F)}](1632.8 \text{ ft}^2)(63.07\text{°F}) = 22.13$ MBtu/h

F correction factor: $\Delta T_1 = 175\text{°F} - 108.56\text{°F} = 66.44\text{°F}$, $\Delta T_2 = 149.82\text{°F} - 90\text{°F} = 59.82\text{°F}$.
$NTU_{I,cf} = (1-R)^{-1} \ln[(1-PR)/(1-P)] = 0.399177$ $\quad$ $F = NTU_{I,cf}/NTU_I = 1.0$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 63.07\text{°F}$ $\quad$ $F\,LMTD_{cf} = 63.07\text{°F} \equiv EMTD$

5.1.4 Rating-Calculation for Decrease in Cooling Water Mass-Flow Rate to 8.3629×10^5 lb_m/h

The properties are assumed to be unchanged as a first approximation, so that the shell-side mass-flow rate, capacity rate, convection coefficient, reference surface area, and wall/fouling resistances are specified by the design-point values.

- Tube-side capacity rate:

$$C_t = (\dot{m}\, c_P)_t = (8.3629 \times 10^5\ \mathrm{lb_m/h})[9980\ \mathrm{Btu/(lb_m\ °F)}]$$

$$= 0.83463\ \mathrm{MBtu/(h\ °F)} \qquad \text{(30\% decrease)}$$

- Tube-side convection coefficient h_t: $A_t = 9738\ \mathrm{ft}^2$.

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{8.3629 \times 10^5\ \mathrm{lb_m/h}}{0.9738\ \mathrm{ft}^2} = 8.588 \times 10^5\ \mathrm{lb_m/(ft^2\ h)}$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 3.854\ \mathrm{ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[8.588 \times 10^5\ \mathrm{lb_m/(ft^2\ h)}](0.652\ \mathrm{ft}/12)}{1.648\ \mathrm{lb_m/(ft\ h)}} = 28{,}310$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} = 0.005994$$

$$Nu_t = \frac{(f_t/2)\, Re_t\, Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 169.64$$

$$h_t = Nu_t \frac{k_t}{d_i} = 169.64\ \frac{0.3587\ \mathrm{Btu/(h\ ft\ °F)}}{0.652\ \mathrm{ft}/12} = 1120\ \mathrm{Btu/(h\ ft^2\ °F)} \qquad \text{(25.5\% decrease)}$$

- Shell-side convection coefficient $h_s = h_{s,des}$:

$$h_s = h_{s,\mathrm{des}} = 1720\ \mathrm{Btu/(h\ ft^2\ °F)} \qquad \text{(essentially unchanged)}$$

- Overall coefficient of heat transfer: $s_t = d_o/d_i = 1.150$, $s_s = 1$.

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.150}{1120} + 0.00006935 + \frac{1}{1720} = 0.001678 \text{ h ft}^2 \text{ °F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.001678 + 0.00330 = 0.004978 \text{ h ft}^2 \text{ °F/Btu}$$

$U_C = 595.95$ Btu/(h ft^2 °F)

$U = 200.85$ Btu/(h ft^2 °F) (5.26% decrease)

- Classical approach: Shell-side fluid is retained as reference for consistency (see Note 5-3 in Section 5.1.6).

$C_I = C_s = 0.8791$ MBtu/(h °F) $C_{II} = C_t = 0.83463$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_s = \frac{UA_s}{C_s} = \frac{[200.85 \text{ Btu/(h ft}^2 \text{ °F)}](1814.3 \text{ ft}^2)}{0.8791 \text{ MBtu/(h °F)}} = 0.41453$$

$$R = \frac{C_I}{C_{II}} = \frac{C_s}{C_t} = \frac{0.8791 \text{ MBtu/(h °F)}}{0.83463 \text{ MBtu/(h °F)}} = 1.0533$$

- Effectiveness: 1-1 E heat exchanger from Table 2-1.

$$P = \frac{1 - \exp[-NTU_I(1-R)]}{1 - R\exp[-NTU_I(1-R)]} = 0.29077$$

- Heat transfer rate: P-NTU$_I$ method.[10]

$q = PC_I\,\delta T_i = 0.29077\,[0.8791 \text{ MBtu/(h °F)}](85\text{°F}) = 21.73$ MBtu/h (8.20% decrease)

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[10] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.29077(85\text{°F})/0.41453 = 59.62\text{°F}$ (3.12% decrease)
$q = UA_s\,EMTD = [200.85 \text{ Btu/(h ft}^2 \text{ °F)}](1814.3 \text{ ft}^2)(59.62\text{°F}) = 21.73$ MBtu/h

F correction factor: $\Delta T_1 = 175\text{°F} - 116.03\text{°F} = 58.97\text{°F}$, $\Delta T_2 = 150.28\text{°F} - 90\text{°F} = 60.28\text{°F}$.
$NTU_{I,cf} = (1-R)^{-1} \ln[(1-PR)/(1-P)] = 0.41453$ $F = NTU_{I,cf}/NTU_I = 1.0$

- Tube-and shell-side outlet temperatures:

$$T_{t,o} = 90°\text{F} + \frac{21.73\ \text{MBtu/h}}{0.8791\ \text{MBtu/(h °F)}} = 116.03°\text{F} \qquad T_{t,\text{ave}} = 103.02°\text{F}$$

$$T_{s,o} = 175°\text{F} - \frac{21.73\ \text{MBtu/h}}{0.8791\ \text{MBtu/(h °F)}} = 150.28°\text{F} \qquad T_{s,\text{ave}} = 162.64°\text{F}$$

- Property refinement: With the properties evaluated at $T_{t,\text{ave}}$ = 103.02°F and $T_{s,\text{ave}}$ = 162.64°F, the analysis indicates a 0.218% increase in heat transfer rate to 21.78 MBtu/h.

5.1.5 Rating Calculation for Increase in Shell-Side Mass-Flow Rate to 9.7385 x 10^5 lb$_m$/h

The properties are assumed to be unchanged as a first approximation, so that the tube-side mass-flow rate, capacity rate, convection coefficient, reference surface area, and wall/fouling resistances are specified by the design-point values.

- Shell-side capacity rate:

$$C_s = (\dot{m}\, c_P)_s = (9.7385 \times 10^5 \text{ lb}_m/\text{h})[1.003 \text{ Btu}/(\text{lb}_m\ ^\circ\text{F})]$$

$$= 0.97677 \text{ MBtu}/(\text{h}\ ^\circ\text{F}) \qquad (11.1\% \text{ increase})$$

- Shell-side convection coefficient h_s: Equations 4-11 and 4-12 reduce to.[11]

$$\psi_{\text{prop}} = \left(\frac{\mu_{s,\text{ref}}}{\mu_s}\right)^{0.6}\left(\frac{Pr_s}{Pr_{s,\text{ref}}}\right)^{1/3}\left(\frac{k_s}{k_{s,\text{ref}}}\right) = 1$$

for a negligible change in properties.

$$h_s = \psi_{\text{prop}}\, (\dot{m}_s/\dot{m}_{s,\text{ref}})^{0.6}\, h_{s,\text{ref}} = 1(9.7385/8.7646)^{0.6}(1720)$$

$$= 1832.2 \text{ Btu}/(\text{h ft}^2\ ^\circ\text{F}) \qquad (6.52\% \text{ increase})$$

- Tube-side convection coefficient h_t:

$$h_t = h_{t,\text{des}} = 1502.5 \text{ Btu}/(\text{h ft}^2\ ^\circ\text{F}) \qquad (\text{essentially unchanged})$$

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.150}{1502.5} + 0.00006935 + \frac{1}{1832.2} = 0.001381 \text{ h ft}^2\ ^\circ\text{F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.001381 + 0.00330 = 0.004681 \text{ h ft}^2\ ^\circ\text{F/Btu}$$

$$U_C = 724.3 \text{ Btu}/(\text{h ft}^2\ ^\circ\text{F}) \qquad U = 213.63 \text{ Btu}/(\text{h ft}^2\ ^\circ\text{F}) \qquad (0.764\% \text{ increase})$$

[11] Calculation of the shell-side Reynolds number indicates that the flow is turbulent.

- Classical approach: Shell-side fluid is retained as reference.

$C_I = C_s = 0.97677$ MBtu/(h °F) $\qquad C_{II} = C_t = 1.1923$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_s = \frac{UA_s}{C_s} = \frac{[213.63 \text{ Btu/(h ft}^2 \text{ °F)}](1814.3 \text{ ft}^2)}{0.97677 \text{ MBtu/(h °F)}} = 0.39681$$

$$R = \frac{C_I}{C_{II}} = \frac{C_s}{C_t} = \frac{0.97677 \text{ MBtu/(h °F)}}{1.1923 \text{ MBtu/(h °F)}} = 0.81922$$

- Effectiveness: 1-1 E heat exchanger from Table 2-1.

$$P = \frac{1 - \exp[-NTU_s(1-R)]}{1 - R\exp[-NTU_s(1-R)]} = 0.29148$$

- Heat transfer rate: P-NTU_I method.[12]

$$q = PC_I\,\delta T_i = 0.29148[0.97677 \text{ MBtu/(h °F)}](85\text{°F}) = 24.20 \text{ MBtu/h} \qquad (2.24\% \text{ increase})$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 90\text{°F} + \frac{24.20 \text{ MBtu/h}}{1.1923 \text{ MBtu/(h °F)}} = 110.29\text{°F} \qquad T_{t,ave} = 100.15\text{°F}$$

$$T_{s,o} = 175\text{°F} - \frac{24.20 \text{ MBtu/h}}{0.97677 \text{ MBtu/(h °F)}} = 150.23\text{°F} \qquad T_{s,ave} = 162.61\text{°F}$$

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[12] EMTD method:
$EMTD = P\,\delta T_i/NTU_I$ = 0.29148(85°F)/0.39681 = 62.437°F (1.46% increase)
$q = UA_s\,EMTD$ = [213.63 Btu/(h ft^2 °F)](1814.3 ft^2)(62.44°F) = 24.20 MBtu/h

F correction factor: ΔT_1 = 175°F – 110.29°F = 64.71°F, ΔT_2 = 150.23°F – 90°F = 60.23°F.
$NTU_{I,cf} = (1-R)^{-1} \ln[(1-PR)/(1-P)] = 0.39681$ $\qquad F = NTU_{I,cf}/NTU_I = 1.0$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$ = 62.44°F $\qquad F\,LMTD_{cf} = 62.44\text{°F} \equiv EMTD$

5.1.6 Notes for Example 1

Note 5-1 Classical calculations for Section 5.1.1: Tube-Side Reference Stream.

$C_I = C_t = 1.1923$ MBtu/(h °F) $\qquad C_{II} = C_s = 0.8791$ MBtu/(h °F)

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = \frac{[212 \text{ Btu/(h ft}^2 \text{ °F)}](1814.3 \text{ ft}^2)}{1.1923 \text{ MBtu/(h °F)}} = 0.32258$$

$$R = \frac{C_I}{C_{II}} = \frac{C_t}{C_s} = \frac{1.1923 \text{ MBtu/(h °F)}}{0.8791 \text{ MBtu/(h °F)}} = 1.3563$$

$$P = \frac{1 - \exp[-NTU_I(1-R)]}{1 - R\exp[-NTU_I(1-R)]} = 0.23356$$

$$q = PC_I\,\delta T_i = 0.23356\,[1.1923 \text{ MBtu/(h °F)}](85\text{°F}) = 23.67 \text{ MBtu/h}$$

$$EMTD = P\,\delta T_i/NTU_t = 61.54\text{°F}$$

Using the notation $R_s = 0.73729$ and $P_s = 0.31678$, we are also able to write:

$$R_t = 1/R_s = 1/0.73729 = 1.3563$$

$$NTU_t = R_s\,NTU_s = 0.73729(0.43753) = 0.32258$$

$$P_t = R_sP_s = 0.73729(0.31678) = 0.23356$$

$$EMTD = P_t\,\delta T_i/NTU_t = 61.54\text{°F}$$

Note 5-2. Effect of tube-side convection coefficient on back-calculation of $h_{s,\mathrm{des}}$.

The back-calculation for shell-side convection coefficient $h_{s,\mathrm{des}}$ based on design or other reference conditions depends on the tube-side convection correlation. To illustrate, using the IHS ESDU correlation, Equation 4-6(a):

$$Nu_t = 0.0225\, Re_t^{0.795}\, Pr_t^{0.495} \exp\left[-0.0225\, (\ln Pr_t)^2\right]$$

gives rise to

$$h_{t,\mathrm{des}} = 1477.4 \text{ Btu/(h ft}^2\ \text{°F)}$$

$$h_{s,\mathrm{des}} = 1953.5 \text{ Btu/(h ft}^2\ \text{°F)}$$

representing a decrease in $h_{t,\mathrm{des}}$ of 8.33% and an increase in $h_{s,\mathrm{des}}$ of 13.58%.

Note 5-3. Classical calculations for Section 5.1.4: Tube-Side Reference Stream.

$C_I = C_t = 0.83463$ MBtu/(h °F) $\qquad$ $C_{II} = C_s = 0.8791$ MBtu/(h °F)

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = \frac{[200.85 \text{ Btu/(h ft}^2\ \text{°F)}](1814.3 \text{ ft}^2)}{0.83463 \text{ MBtu/(h °F)}} = 0.43661$$

$$R = \frac{C_I}{C_{II}} = \frac{C_t}{C_s} = \frac{0.83463 \text{ MBtu/(h °F)}}{0.8791 \text{ MBtu/(h °F)}} = 0.94942$$

$$P = \frac{1 - \exp\left[-NTU_s (1 - R)\right]}{1 - R \exp\left[-NTU_s (1 - R)\right]} = 0.30626$$

$$q = PC_I\, \delta T_i = 0.30626\, [0.83463 \text{ MBtu/(h °F)}]\, (85\text{°F}) = 21.73 \text{ MBtu/h}$$

$$EMTD = P\, \delta T_i / NTU_t = 0.30626(85\text{°F})/0.43661 = 59.62\text{°F}$$

5.2 Example 2. Diesel Generator Cooler: TEMA 1-2 E

A TEMA 1-2 E shell-and-tube heat exchanger is used to cool a diesel generator. Design and thermal performance specifications for the heat exchanger are listed in Figure 5-2 and Tables 5-3 and 5-4. The P-NTU_I method is to be used to develop the following performance analysis calculations:

- Check of thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for 10% plugged tubes
- Rating-calculation for increase in cooling water mass-flow rate to 1.991×10^5 lb_m/h

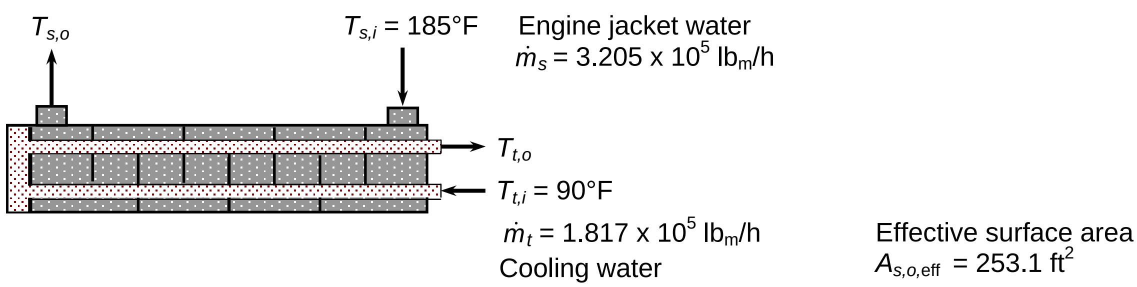

Figure 5-2
Diesel Generator Cooler Design Conditions

Table 5-3
Design Specifications: 1-2 E Shell and Tube

Tubing	Shell-side characteristics
Admiralty brass	Triangular arrangement
k_w = 63.1 Btu/(h ft °F)	N = 182
L_{eff} = 8.5 ft	S_t = 0.75 in.
d_i = 0.527 in.	$D_{i,s}$ = 11.95 in., L_b = 11.3 in.
d_o = 0.625 in.	N_b = 7, B_c = 25%, single-segmental baffles

Table 5-4
Thermal Performance Specifications: 1-2 E Shell and Tube

Thermal Performance Specifications	
$F_{f,t}$ = 0.002 h ft^2 °F/Btu	$T_{t,i,\text{des}}$ = 90°F
$F_{f,s}$ = 0	$T_{t,o,\text{des}}$ = 117.49°F
q_{des} = 4.989 MBtu/h	$T_{t,\text{ave}}$ = 103.75°F
U_{des} = 272.41 Btu/(h ft^2 °F)	$T_{s,i,\text{des}}$ = 185°F
$EMTD_{\text{des}}$ = 72.36°F	$T_{s,o,\text{des}}$ = 169.5°F
	$T_{s,\text{ave}}$ = 177.2°F

5.2.1 Check of Thermal Performance Specifications

- Reference surface: The outer surface is selected for reference.

  $A_s = A_{s,o} = N(\pi d_o L_{\text{eff}}) = 182[\pi(0.625 \text{ ft}/12)](8.5 \text{ ft}) = 253.1 \text{ ft}^2$

- Properties: k_w = 63.1 Btu/(h ft °F) for admiralty brass.

  Fluid properties are specified at average terminal temperatures.

  Tube-side at $T_{t,\text{ave}}$ = 103.75°F

  $\rho_t = 61.85 \text{ lb}_\text{m}/\text{ft}^3$

  $c_{P,t} = 0.9983 \text{ Btu}/(\text{lb}_\text{m}\ °\text{F})$

  $k_t = 0.3604 \text{ Btu}/(\text{h ft}\ °\text{F})$

  $\mu_t = 1.580 \text{ lb}_\text{m}/(\text{ft h})$

  $Pr_t = 4.377$

  Shell-side at $T_{s,\text{ave}}$ = 177.25°F

  $\rho_s = 60.67 \text{ lb}_\text{m}/\text{ft}^3$

  $c_{P,s} = 1.005 \text{ Btu}/(\text{lb}_\text{m}\ °\text{F})$

  $k_s = 0.3871 \text{ Btu}/(\text{h ft}\ °\text{F})$

  $\mu_s = 0.8093 \text{ lb}_\text{m}/(\text{ft h})$

  $Pr_s = 2.101$

- Inlet temperature difference and capacity rates:

  $\delta T_i = T_{s,i} - T_{t,i} = 185°\text{F} - 90°\text{F} = 95°\text{F}$

  $C_t = (\dot{m}\, c_P)_t = (1.817 \times 10^5 \text{ lb}_\text{m}/\text{h})[0.9983 \text{ Btu}/(\text{lb}_\text{m}\ °\text{F})] = 0.1814 \text{ MBtu}/(\text{h}\ °\text{F})$

  $C_s = (\dot{m}\, c_P)_s = (3.205 \times 10^5 \text{ lb}_\text{m}/\text{h})[1.005 \text{ Btu}/(\text{lb}_\text{m}\ °\text{F})] = 0.3221 \text{ MBtu}/(\text{h}\ °\text{F})$

- Basic check of thermal performance specifications:

  $\delta T_{t,\text{des}} = T_{t,o,\text{des}} - T_{t,i,\text{des}} = 27.49°\text{F}$ $q_{t,\text{des}} = C_t\, \delta T_{t,\text{des}} = 4.987$ MBtu/h

  $\delta T_{s,\text{des}} = T_{s,i,\text{des}} - T_{s,o,\text{des}} = 15.5°\text{F}$ $q_{s,\text{des}} = C_s\, \delta T_{s,\text{des}} = 4.9926$ MBtu/h

  $q_{\text{ave}} = (q_{t,\text{des}} + q_{s,\text{des}})/2 = 4.9898$ MBtu/h

  $q_{EMTD} = U_{\text{des}} A_s\, EMTD_{\text{des}} = [272.41 \text{ Btu/(h ft}^2\ °\text{F)}](253.1 \text{ ft}^2)(72.36°\text{F}) = 4.989$ MBtu/h

The difference in the calculations is negligible (less than 0.02%) for q_{des} and q_{ave} as well as for q_{EMTD} and q_{des}, which indicates consistency in the specifications for q_{des}, U_{des}, $EMTD_{\text{des}}$, $T_{t,o}$, and $T_{s,o}$.

- Overall coefficient of heat transfer: Set $U = U_{\text{des}} = 272.41$ Btu/(h ft^2 °F)
- Classical approach: Tube-side fluid is selected for reference.

  $C_I = C_t = 0.1814$ MBtu/(h °F) $C_{II} = C_s = 0.3221$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = \frac{[272.41 \text{ Btu/(h ft}^2\ °\text{F)}](253.1 \text{ ft}^2)}{0.1814 \text{ MBtu/(h °F)}} = 0.3801$$

$$R = \frac{C_I}{C_{II}} = \frac{C_t}{C_s} = \frac{0.1814 \text{ MBtu/(h °F)}}{0.3221 \text{ MBtu/(h °F)}} = 0.5631$$

- Effectiveness: 1-2 E heat exchanger from Table 2-1.

$$P = \frac{2}{1 + R + \sqrt{1+R^2}\,(1+e^{-\Gamma_1})/(1-e^{-\Gamma_1})} = 0.2895 \qquad \Gamma_1 = NTU_I\sqrt{1+R^2}$$

- Heat transfer rate: P-NTU$_I$ method.[13]

$$q = PC_I\, \delta T_i = 0.2895\,[0.1814 \text{ MBtu/(h °F)}](95°\text{F}) = 4.989 \text{ MBtu/h}$$

[13] EMTD method: $q = UA_s\, EMTD = [272.41 \text{ Btu/(h ft}^2\ °\text{F)}](253.1 \text{ ft}^2)(72.36°\text{F}) = 4.989$ MBtu/h.

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 90°\text{F} + \frac{4.989 \text{ MBtu/h}}{0.1814 \text{ MBtu/(h °F)}} = 117.50°\text{F} \qquad T_{t,\text{ave}} = 103.75°\text{F}$$

$$T_{s,o} = 185°\text{F} - \frac{4.989 \text{ MBtu/h}}{0.3221 \text{ MBtu/(h °F)}} = 169.51°\text{F} \qquad T_{s,\text{ave}} = 177.26°\text{F}$$

- Effective mean temperature difference:[14]

$$EMTD = \frac{P\,\delta T_i}{NTU_I} = \frac{(0.2895)(95°\text{F})}{0.3801} = 72.36°\text{F}$$

The calculations for q, $EMTD$, $T_{t,o}$, and $T_{s,o}$ differ negligibly from the thermal performance specifications.

Conclusion: The classical calculations for design conditions are consistent with the thermal performance specifications provided by the manufacturer.

[14] F correction factor: $\Delta T_1 = 185°\text{F} - 117.50°\text{F} = 67.5°\text{F}$, $\Delta T_2 = 169.51°\text{F} - 90°\text{F} = 79.51°\text{F}$.
$NTU_{I,cf} = (1-R)^{-1} \ln [(1-PR)/(1-P)] = 0.3801$ $\quad F = NTU_{I,cf}/NTU_I = 0.98656$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln (\Delta T_1/\Delta T_2) = 73.34°\text{F}$ $\quad F\,LMTD_{cf} = 72.36°\text{F} \cong EMTD$

5.2.2 Back-Calculation of $h_{s,des}$

The shell-side convection coefficient $h_{s,\mathrm{des}}$ is back-calculated based on the specified value of U_{des}.

- Wall and fouling resistances: $s_t = d_o/d_i = 0.625/0.527 = 1.186$, $s_s = 1$.

$$r_w = \frac{d_o}{2k_w} \ln \frac{d_o}{d_i} = \frac{0.625\ \text{ft}/12}{2\ [63.1\ \text{Btu}/(\text{h ft °F})]} \ln \frac{0.625}{0.527} = 0.00007039\ \text{h ft}^2\ \text{°F/Btu}$$

$$r_f = s_t F_{f,t} + s_s F_{f,s} = 1.186(0.002) + 1(0) = 0.0023719\ \text{h ft}^2\ \text{°F/Btu}$$

- Clean overall coefficient of heat transfer:

$$\frac{1}{U_{C,\mathrm{des}}} = \frac{1}{U_{\mathrm{des}}} - r_f = \frac{1}{272.41} - 0.0023719 = 0.001299\ \text{h ft}^2\ \text{°F/Btu}$$

$$U_{C,\mathrm{des}} = 769.8\ \text{Btu/(h ft}^2\ \text{°F)}$$

- Tube-side convection coefficient $h_{t,\mathrm{des}}$:

$$A_t = \frac{\pi d_i^2}{4} \frac{N}{N_{tp}} = \frac{\pi(0.527\ \text{ft}/12)^2}{4} \frac{182}{2} = 0.13784\ \text{ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{1.817 \times 10^5\ \text{lb}_\text{m}/\text{h}}{0.13784\ \text{ft}^2} = 1.318 \times 10^6\ \text{lb}_\text{m}/(\text{ft}^2\ \text{h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 5.920\ \text{ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.318 \times 10^6\ \text{lb}_\text{m}/(\text{ft}^2\ \text{h})](0.527\ \text{ft}/12)}{1.580\ \text{lb}_\text{m}/(\text{ft h})} = 36{,}640$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} = 0.005633$$

$$Nu_t = \frac{(f_t/2)\ Re_t\ Pr_t}{1.07 + 12.7\sqrt{f_t/2}\ (Pr_t^{2/3} - 1)} = 205.3$$

$$h_{t,\mathrm{des}} = Nu_t \frac{k_t}{d_i} = 205.3\ \frac{0.3604\ \text{Btu/(h ft °F)}}{0.527\ \text{ft}/12} = 1685\ \text{Btu/(h ft}^2\ \text{°F)}$$

- Shell-side convection coefficient $h_{s,\text{des}}$:

$$\frac{1}{h_{s,\text{des}}} = \frac{1}{s_s}\left(\frac{1}{U_{C,\text{des}}} - \frac{s_t}{h_{t,\text{des}}} - r_w\right) = \frac{1}{U_{C,\text{des}}} - \frac{s_t}{h_{t,\text{des}}} - r_w$$

$$= 0.001299 - \frac{1.186}{1685} - 0.00007039 = 0.00052475 \text{ h ft}^2\ °\text{F/Btu}$$

$h_{s,\text{des}} = 1905.3$ Btu/(h ft^2 °F)

5.2.3 Rating Calculation for 10% Plugged Tubes

The properties are assumed to be unchanged; the capacity rates, shell-side convection coefficient, and wall/fouling resistances are specified by the design-point values.

- Number of active tubes:

$N_{at} = N\,(1 - PLUG) = 182(1 - 0.1) = 163.8$ (10% decrease)

- Reference surface area:

$A_s = A_{s,o} = N_{ta}(\pi d_o L) = 163.8[\pi(0.625\text{ ft}/12)](8.5\text{ ft}) = 227.8\text{ ft}^2$ (10% decrease)

- Tube-side convection coefficient h_t:

$$A_t = \frac{\pi d_i^2}{4}\frac{N_{at}}{N_{tp}} = \frac{\pi(0.527\text{ ft}/12)^2}{4}\frac{163.8}{2} = 0.1241\text{ ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{1.817\times 10^6\text{ lb}_\text{m}/\text{h}}{0.1241\text{ ft}^2} = 1.465 \times 106\text{ lbm/(ft}^2\text{ h)}$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 6.578\text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.465\times 10^6\text{ lb}_\text{m}/(\text{ft}^2\text{ h})](0.527\text{ ft}/12)}{1.580\text{ lb}_\text{m}/(\text{ft h})} = 40{,}710\text{ ft}$$

$f_t = (1.58 \ln \text{Re}_\text{t} - 3.28)–2 = 0.005495$

$$Nu_t = \frac{(f_t/2)\,Re_t\,Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 224.0$$

$$h_t = Nu_t\frac{k_t}{d_i} = 224.0\,\frac{0.3604\text{ Btu/(h ft °F)}}{0.527\text{ ft}/12} = 1838\text{ Btu/(h ft}^2\text{ °F)}$$ (9.08% increase)

- Shell-side convection coefficient h_s:

$h_s = h_{s,\text{des}} = 1905.3\text{ Btu/(h ft}^2\text{ °F)}$ (essentially unchanged)

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.186}{1838} + 0.00007039 + \frac{1}{1905.3} = 0.0012405 \text{ h ft}^2\ \text{°F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.0012405 + 0.0023719 = 0.0036124 \text{ h ft}^2\ \text{°F/Btu}$$

$U_C = 806.1$ Btu/(h ft^2 °F) $U = 276.8$ Btu/(h ft^2 °F) (1.61% increase)

- Classical approach: Tube-side fluid is retained as reference.

$C_I = C_t = 0.1814$ MBtu/(h °F) $C_{II} = C_s = 0.3221$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = \frac{[276.8 \text{ Btu/(h ft}^2\text{ °F)}](227.8 \text{ ft}^2)}{0.1814 \text{ MBtu/(h °F)}} = 0.3476$$

$$R = \frac{C_I}{C_{II}} = \frac{C_t}{C_s} = \frac{0.1814 \text{ MBtu/(h °F)}}{0.3221 \text{ MBtu/(h °F)}} = 0.5631$$

(essentially unchanged)

- Effectiveness: 1-2 E heat exchanger from Table 2-1.

$$P = \frac{2}{1 + R + \sqrt{1 + R^2}\,(1 + e^{-\Gamma_1})/(1 - e^{-\Gamma_1})} = 0.2705$$

$$\Gamma_1 = NTU_I\sqrt{1 + R^2}$$

- Heat transfer rate: P-NTU_I method.[15]

$$q = PC_I\,\delta T_i = 0.2705\,[0.1814\ \text{MBtu/(h °F)}]\,(95\,°\text{F}) = 4.662\ \text{MBtu/h} \qquad (6.55\%\ \text{decrease})$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 90°\text{F} + \frac{4.662\ \text{MBtu/h}}{0.1814\ \text{MBtu/(h °F)}} = 115.70\,°\text{F} \qquad T_{t,ave} = 102.85°\text{F}$$

$$T_{s,o} = 185°\text{F} - \frac{4.662\ \text{MBtu/h}}{0.3221\ \text{MBtu/(h °F)}} = 170.53\,°\text{F} \qquad T_{s,ave} = 177.76°\text{F}$$

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[15] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.2705(95°\text{F})/0.3476 = 73.93°\text{F}$ (2.17% increase)
$q = UA_s\,EMTD = [276.8\ \text{Btu/(h ft}^2\ °\text{F)}](227.8\ \text{ft}^2)(73.93°\text{F}) = 4.662\ \text{MBtu/h}$

F correction factor: $\Delta T_1 = 185°\text{F} – 115.70°\text{F} = 69.3°\text{F}$, $\Delta T_2 = 170.53°\text{F} – 90°\text{F} = 80.53°\text{F}$.
$NTU_{I,cf} = (1–R)^{-1} \ln [(1–PR)/(1–P)] = 0.34366$ $\quad F = NTU_{I,cf}/NTU_I = 0.98867$
$LMTD_{cf} = (\Delta T_1–\Delta T_2)/\ln (\Delta T_1/\Delta T_2) = 74.774°\text{F}$ $\quad F\,LMTD_{cf} = 73.93°\text{F} \equiv EMTD$

5.2.4 Rating-Calculation for Increase in Cooling Water Mass-Flow Rate to 1.991 x 10^5 lb_m/h

The properties are assumed to be unchanged; the shell-side mass-flow rate, capacity rate, convection coefficient, reference surface area, and wall/ fouling resistances are specified by the design-point values.

- Tube-side capacity rate:

$$C_t = (\dot{m}\, c_P)_t = (1.991 \times 10^5 \text{ lb}_\text{m}/\text{h})[0.9983 \text{ Btu}/(\text{lb}_\text{m}\ °\text{F})]$$

$$= 0.1988 \text{ MBtu}/(\text{h °F}) \qquad (8.75\% \text{ increase})$$

- Tube-side convection coefficien h_t: $A_t = 0.13784$ ft^2.

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{1.991 \times 10^5 \text{ lb}_\text{m}/\text{h}}{0.13784 \text{ ft}^2} = 1.444 \times 10^6 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 6.487 \text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.444 \times 10^5 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})](0.527 \text{ ft}/12)}{1.580 \text{ lb}_\text{m}/(\text{ft h})} = 40{,}150$$

$$ft = (1.58 \ln Re_t - 3.28)–2 = 0.005513$$

$$Nu_t = \frac{(f_t/2)\, Re_t\ Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 221.4$$

$$h_t = Nu_t \frac{k_t}{d_i} = 221.4 \frac{0.3604 \text{ Btu}/(\text{h ft °F})}{0.527 \text{ ft}/12} = 1817 \text{ Btu}/(\text{h ft}^2 \text{ °F}) \qquad (7.83\% \text{ increase})$$

- Shell-side convection coefficient h_s:

$$h_s = h_{s,\text{des}} = 1905.3 \text{ Btu}/(\text{h ft}^2 \text{ °F}) \qquad (\text{essentially unchanged})$$

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.186}{1817} + 0.00007039 + \frac{1}{1905.3} = 0.001248 \text{ h ft}^2\ \text{°F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.001248 + 0.0023719 = 0.003620 \text{ h ft}^2\ \text{°F/Btu}$$

U_C = 801.3 Btu/(h ft^2 °F) $\qquad$ U = 276.25 Btu/(h ft^2 °F) $\qquad$ (1.41% increase)

- Classical approach: Tube-side fluid is selected for reference.

$C_I = C_t$ = 0.1988 MBtu/(h °F) $\qquad$ $C_{II} = C_s$ = 0.3221 MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = \frac{[276.25\ \text{Btu/(h ft}^2\ \text{°F)}](253.1\ \text{ft}^2)}{0.1988\ \text{MBtu/(h °F)}} = 0.3517$$

$$R = \frac{C_I}{C_{II}} = \frac{C_t}{C_s} = \frac{0.1988\ \text{MBtu/(h °F)}}{0.3221\ \text{MBtu/(h °F)}} = 0.6172$$

- Effectiveness: 1-2 E heat exchanger from Table 2-1.

$$P = \frac{2}{1 + R + \sqrt{1+R^2}\,(1+e^{-\Gamma_1})/(1-e^{-\Gamma_1})} = 0.27084 \qquad \Gamma_1 = NTU_I\sqrt{1+R^2}$$

- Heat transfer rate: P-NTU$_I$ method.[16]

$$q = PC_I\ \delta T_i = 0.27084\,[0.1988\ \text{MBtu/(h °F)}](95\text{°F}) = 5.115\ \text{MBtu/h} \qquad (2.52\%\ \text{increase})$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 90\text{°F} + \frac{5.115\ \text{MBtu/h}}{0.1988\ \text{MBtu/(h °F)}} = 115.73\text{°F} \qquad T_{t,\text{ave}} = 102.86\text{°F}$$

$$T_{s,o} = 185\text{°F} - \frac{5.115\ \text{MBtu/h}}{0.3221\ \text{MBtu/(h °F)}} = 169.12\text{°F} \qquad T_{s,\text{ave}} = 177.05\text{°F}$$

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[16] EMTD method:
$EMTD = P\ \delta T_i / NTU_I$ = 0.27084(95°F)/0.3517 = 73.16°F (1.11% increase)
$q = UA_s\ EMTD$ = [276.25 Btu/(h ft^2 °F)](253.1 ft^2)(73.16°F) = 5.115 MBtu/h

F correction factor: ΔT_1 = 185°F – 115.73°F = 69.27°F, ΔT_2 = 169.12°F – 90°F = 79.12°F.
$NTU_{I,cf} = (1-R)^{-1} \ln[(1-PR)/(1-P)]$ = 0.3473 $\qquad$ $F = NTU_{I,cf}/NTU_I$ = 0.98746
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$ = 74.086°F $\qquad$ $F\ LMTD_{cf}$ = 73.16°F $\equiv EMTD$

5.3 Example 3. Component Cooling Water Heat Exchanger: TEMA 1-2 F

A TEMA 1-2 F shell-and-tube heat exchanger is used for component cooling. Design and thermal performance specifications provided by the manufacturer are shown in Figure 5-3 and Tables 5-5 and 5-6. The P-NTU$_I$ method is to be used to develop the following performance analysis calculations:

- Calculation of actual design thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for 10% plugged tubes
- Rating-calculation for increase in cooling water temperature to 80°F

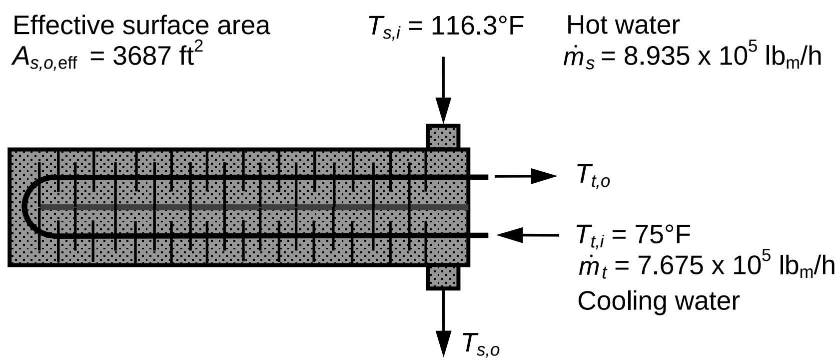

Figure 5-3
Component Cooling Water Heat Exchanger Design Conditions

Table 5-5
Design Specifications: 1-2 F Shell and Tube

Tubing	Shell-Side Characteristics
Admiralty brass	Triangular arrangement
k_w = 63.1 Btu/(h ft °F)	N = 444
L_{eff} = 36.25 ft	S_t = 1.375 in.
d_i = 0.777 in.	$D_{i,s}$ = 35 in., L_b = 19.22 in.
d_o = 0.875 in.	N_b = 22, B_c = 25%, single-segmental baffles

Table 5-6
Thermal Performance Specifications: 1-2 F Shell and Tube

Thermal Performance Specifications	
$F_{f,t}$ = 0.00075 h ft^2 °F/Btu	$T_{t,i,\mathrm{des}}$ = 75°F
$F_{f,s}$ = 0.0005 h ft^2 °F/Btu	$T_{t,o,\mathrm{des}}$ = 100°F
q_{des} = 19.22 MBtu/h	$T_{t,\mathrm{ave}}$ = 87.5°F
U_{des} = 288 Btu/(h ft^2 °F)	$T_{s,i,\mathrm{des}}$ = 116.3°F
$EMTD_{\mathrm{des}}$ = 18.1°F	$T_{s,o,\mathrm{des}}$ = 95°F
	$T_{s,\mathrm{ave}}$ = 105.65°F

Heat transfer overdesign: 7.6%; q_{DES} = 20.68 MBtu/h.

5.3.1 Calculation of Actual Thermal Performance Specifications

- Reference surface: The outer surface is selected for reference.

  $A_s = A_{s,o,\mathrm{eff}} = N(\pi d_o L_{\mathrm{eff}}) = 444[\pi(0.875\text{ ft}/12)](36.25\text{ ft}) = 3687\text{ ft}^2$

- Properties: k_w = 63.1 Btu/(h ft °F) for admiralty brass.

  Fluid properties are specified at average temperatures.

  Tube-side at $T_{t,\mathrm{ave}}$ = 87.5°F

  $\rho_t = 62.10\ \mathrm{lb_m/ft^3}$

  $c_{P,t} = 0.9965\ \mathrm{Btu/(lb_m\ °F)}$

  $k_t = 0.3532\ \mathrm{Btu/(h\ ft\ °F)}$

  $\mu_t = 1.9135\ \mathrm{lb_m/(ft\ h)}$

  $Pr_t = 5.399$

  Shell-side at $T_{s,\mathrm{ave}}$ = 105.65°F

  $\rho_s = 61.81\ \mathrm{lb_m/ft^3}$

  $c_{P,s} = 0.9984\ \mathrm{Btu/(lb_m\ °F)}$

  $k_s = 0.3612\ \mathrm{Btu/(h\ ft\ °F)}$

  $\mu_s = 1.546\ \mathrm{lb_m/(ft\ h)}$

  $Pr_s = 4.273$

- Approximate capacity rates:

  $C_t = (\dot{m}\, c_P)_t = (7.675 \times 10^5\ \mathrm{lb_m/h})[0.9965\ \mathrm{Btu/(lb_m\ °F)}] = 0.7648\ \mathrm{MBtu/(h\ °F)}$

  $C_s = (\dot{m}\, c_P)_s = (8.935 \times 10^5\ \mathrm{lb_m/h})[0.9984\ \mathrm{Btu/(lb_m\ °F)}] = 0.8921\ \mathrm{MBtu/(h\ °F)}$

- Actual design outlet temperatures and terminal temperature differences:

  Set $q = q_{\mathrm{DES}}$ = 20.68 MBtu/h

$$T_{t,o} = 75°\mathrm{F} + \frac{20.68\ \mathrm{MBtu/h}}{0.7648\ \mathrm{MBtu/(h\ °F)}} = 102.04°\mathrm{F} \qquad \delta T_t = T_{t,o} - T_{t,i} = 27.04°\mathrm{F}$$

$$T_{s,o} = 116.3°\mathrm{F} - \frac{20.68\ \mathrm{MBtu/h}}{0.8921\ \mathrm{MBtu/(h\ °F)}} = 93.12°\mathrm{F} \qquad \delta T_s = T_{s,i} - T_{s,o} = 23.18°\mathrm{F}$$

$$\delta T_i = T_{s,i} - T_{t,i} = 116.3°\mathrm{F} - 75°\mathrm{F} = 41.3°\mathrm{F}$$

- Classical approach: Tube-side fluid is selected for reference.

$$C_I = C_t = 0.7648\ \mathrm{MBtu/(h\ °F)} \qquad C_{II} = C_s = 0.8921\ \mathrm{MBtu/(h\ °F)}$$

- Independent performance parameters:

$$P = \frac{\delta \mathrm{T}_I}{\delta \mathrm{T}_i} = \frac{\delta \mathrm{T}_t}{\delta \mathrm{T}_i} = \frac{27.04°\mathrm{F}}{41.3°\mathrm{F}} = 0.6547$$

$$R = \frac{\delta \mathrm{T}_{II}}{\delta \mathrm{T}_I} = \frac{\delta \mathrm{T}_s}{\delta \mathrm{T}_t} = \frac{23.18°\mathrm{F}}{27.04°\mathrm{F}} = 0.8573$$

- Number of transfer units: 1-2 F heat exchanger (Table 2-1, approximated by ideal counterflow arrangement).[17]

$$NTU_I = NTU_t = \frac{1}{1-R} \ln \frac{1-PR}{1-P} = 1.678$$

[17] A correction factor is commonly used to account for the effect of heat transfer across the horizontal baffle.

- Actual overall coefficient of heat transfer: P-NTU$_I$ method.

$$U_{\mathrm{DES}} = NTU_I \frac{C_I}{A_s} = NTU_t \frac{C_t}{A_s} = 1.678 \ \frac{0.7648 \ \mathrm{MBtu/(h \ °F)}}{3687 \ \mathrm{ft}^2}$$

$$= 348.1 \ \mathrm{Btu/(h \ ft^2 \ °F)} \qquad \text{(20.87\% difference)}$$

- Effective mean temperature difference:[18]

$$EMTD_{\mathrm{DES}} = \frac{\delta T_I}{NTU_I} = \frac{\delta T_t}{NTU_t} = \frac{27.04°\mathrm{F}}{1.678} = 16.11°\mathrm{F} \qquad \text{(–10.99\% difference)}$$

[18] ΔT_1 = 116.3°F – 102.04°F = 14.26°F, ΔT_2 = 93.12°F – 75°F = 18.12°F.
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$ = 16.11°F
$F = EMTD/LMTD_{cf}$ = 1.0

5.3.2 Back-Calculation of $h_{s,DES}$

The shell-side convection coefficient $h_{s,\mathrm{DES}}$ is back-calculated based on the specified value of U_{DES} corresponding to q_{DES}.

- Wall and fouling resistance: $s_t = d_o/d_i = 0.875/0.777 = 1.126$, $s_s = 1$.

$$r_w = \frac{d_o}{2k_w} \ln\frac{d_o}{d_i} = 0.00006863 \text{ h ft}^2 \text{ °F/Btu}$$

$$r_f = s_t F_{f,t} + s_s F_{f,s} = 0.0013445 \text{ h ft}^2 \text{ °F/Btu}$$

- Clean overall coefficient of heat transfer:

$$\frac{1}{U_{C,\mathrm{DES}}} = \frac{1}{U_{\mathrm{DES}}} - r_f = \frac{1}{348.1} - 0.001345 = 0.001527 \text{ h ft}^2 \text{ °F/Btu}$$

$$U_{C,\mathrm{DES}} = 654.7 \text{ Btu/(h ft}^2 \text{ °F)}$$

- Tube-side convection coefficient $h_{t,\mathrm{DES}}$:

$$A_t = \frac{\pi d_i^2}{4}\frac{N}{N_{tp}} = \frac{\pi(0.777 \text{ ft}/12)^2}{4}\frac{444}{2} = 0.7310 \text{ ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{7.675\times10^5 \text{ lb}_\mathrm{m}/\text{h}}{0.7310 \text{ ft}^2} = 1.050 \times 10^6 \text{ lb}_\mathrm{m}/(\text{ft}^2 \text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 4.696 \text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.050\times10^6 \text{ lb}_\mathrm{m}/(\text{ft}^2 \text{ h})](0.777 \text{ ft}/12)}{1.9135 \text{ lb}_\mathrm{m}/(\text{ft h})} = 35{,}530$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} = 0.005674$$

$$Nu_t = \frac{(f_t/2)\, Re_t\, Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 219.9$$

$$h_{t,\mathrm{DES}} = Nu_t \frac{k_t}{d_i} = 219.9 \frac{0.3532 \text{ Btu/(h ft °F)}}{0.777 \text{ ft}/12} = 1199 \text{ Btu/(h ft}^2 \text{ °F)}$$

- Shell-side convection coefficient $h_{s,\mathrm{DES}}$:

$$\frac{1}{h_{s,\mathrm{DES}}} = \frac{1}{s_s}\left(\frac{1}{U_{C,\mathrm{DES}}} - \frac{s_t}{h_{t,\mathrm{DES}}} - r_w\right) = \frac{1}{U_{C,\mathrm{DES}}} - \frac{s_t}{h_{t,\mathrm{DES}}} - r_w$$

$$= \frac{1}{654.7} - \frac{1.126}{1199} - 0.00006863 = 0.00051967 \text{ h ft}^2\ {}^\circ\text{F/Btu}$$

$h_{s,\mathrm{DES}}$ = 1924.3 Btu/(h ft^2 °F)

5.3.3 Rating-Calculation for 10% Plugged Tubes

The properties are assumed to be unchanged as a first approximation, so that the capacity rates, shell-side convection coefficient, and wall/fouling resistances are specified by the design-point values.

- Number of active tubes, rounded to nearest even number:

$N_{at} = N(1 - PLUG) = 444(1 - 0.1) = 400$ (10% decrease)

- Reference surface area:

$A_s = A_{s,o,\mathrm{eff}}\,(1 - PLUG) = 3{,}687(1 - 0.1) = 3318\ \mathrm{ft}^2$ (10% decrease)

- Tube-side convection coefficient h_t:

$$A_t = \frac{\pi d_i^2}{4}\frac{N_{at}}{N_{tp}} = \frac{\pi(0.777\ \mathrm{ft}/12)^2}{4}\frac{400}{2} = 0.6586\ \mathrm{ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{7.6753\times10^5\ \mathrm{lb_m/h}}{0.6586\ \mathrm{ft}^2} = 1.165\times10^6\ \mathrm{lb_m/(ft^2\ h)}$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 5.213\ \mathrm{ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.165\times10^6\ \mathrm{lb_m/(ft^2\ h)}](0.777\ \mathrm{ft}/12)}{1.9135\ \mathrm{lb_m/(ft\ h)}} = 39{,}440$$

$$f_t = (1.58\ \ln Re_t - 3.28)^{-2} = 0.005536$$

$$Nu_t = \frac{(f_t/2)\ Re_t\ Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 239.8$$

$$h_t = Nu_t\frac{k_t}{d_i} = 239.8\ \frac{0.3532\ \mathrm{Btu/(h\ ft\ °F)}}{0.777\ \mathrm{ft}/12} = 1308\ \mathrm{Btu/(h\ ft^2\ °F)}$$ (9.09% increase)

- Shell-side convection coefficient h_s:

$h_s = h_{s,\mathrm{DES}} = 892.8\ \mathrm{Btu/(h\ ft^2\ °F)}$ (essentially unchanged)

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.126}{1308} + 0.00006863 + \frac{1}{1924.3} = 0.001449 \text{ h ft}^2 \text{ °F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.001449 + 0.001345 = 0.002794 \text{ h ft}^2 \text{ °F/Btu}$$

$U_C = 690.1$ Btu/(h ft^2 °F) $\quad$ $U = 357.9$ Btu/(h ft^2 °F) $\quad$ (2.82% increase)

- Classical approach: Tube-side fluid is retained as reference.

$C_I = C_t = 0.7648$ MBtu/(h °F) $\quad$ $C_{II} = C_s = 0.8921$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = \frac{[357.9 \text{ Btu/(h ft}^2 \text{ °F)}](3318 \text{ ft}^2)}{0.7648 \text{ MBtu/(h °F)}} = 1.5527$$

$$R = \frac{C_I}{C_{II}} = \frac{C_t}{C_s} = \frac{0.7648 \text{ MBtu/(h °F)}}{0.8921 \text{ MBtu/(h °F)}} = 0.8573 \qquad \text{(essentially unchanged)}$$

- Effectiveness: 1-2 F heat exchanger (Table 2-1, approximated by ideal counterflow arrangement).

$$P = \frac{1 - \exp[-NTU_I(1-R)]}{1 - R \exp[-NTU_I(1-R)]} = 0.6348$$

- Heat transfer rate: P-NTU$_I$ method.[19]

$$q = PC_I\,\delta T_i = 0.6348\,[0.7648\ \text{MBtu/(h °F)}]\,(41.3\,°\text{F}) = 20.05\ \text{MBtu/h} \quad (3.05\%\ \text{decrease})$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 75°\text{F} + \frac{20.05\ \text{MBtu/h}}{0.7648\ \text{MBtu/(h °F)}} = 101.22\,°\text{F} \qquad T_{t,\text{ave}} = 88.11°\text{F}$$

$$T_{s,o} = 116.3°\text{F} - \frac{20.05\ \text{MBtu/h}}{0.8921\ \text{MBtu/(h °F)}} = 93.82\,°\text{F} \qquad T_{s,\text{ave}} = 105.06°\text{F}$$

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[19] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.6348(41.3°\text{F})/1.5527 = 16.885°\text{F}$ (4.81% increase)
$q = UA_s\,EMTD = [357.9\ \text{Btu/(h ft}^2\ °\text{F)}](3318\ \text{ft}^2)(16.885°\text{F}) = 20.05\ \text{MBtu/h}$

F correction factor: $\Delta T_1 = 116.3°\text{F} - 101.22°\text{F} = 15.08°\text{F}$, $\Delta T_2 = 93.82°\text{F} - 75°\text{F} = 18.82°\text{F}$.
$NTU_{I,cf} = (1-R)^{-1} \ln [(1-PR)/(1-P)] = 1.5527$ $\qquad F = NTU_{I,cf}/NTU_I = 1.0$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 16.88°\text{F}$ $\qquad F\,LMTD_{cf} = 16.88°\text{F} \equiv EMTD$

5.3.4 Rating-Calculation for Increase in Cooling Water Temperature to 80°F

The properties are assumed to be unchanged as a first approximation, so that the capacity rates, convection coefficients, reference surface area, and wall/fouling resistances are specified by the design-point values.

- Terminal temperature difference: Capacity rates unchanged.

$\delta T_i = T_{s,i} - T_{t,i} = 116.3°\text{F} - 80°\text{F} = 36.3°\text{F}$ (12.1% decrease)

$C_t = 0.7648$ MBtu/(h °F) $C_s = 0.8921$ MBtu/(h °F)

- Classical approach: Tube-side fluid is retained as reference.

$C_I = C_t = 0.7648$ MBtu/(h °F) $C_{II} = C_s = 0.8921$ MBtu/(h °F)

- Independent performance parameters (essentially unchanged):

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = 1.678$$

$$R = \frac{C_I}{C_{II}} = \frac{C_t}{C_s} = 0.8573$$

- Effectiveness (essentially unchanged):

$$P = \frac{1 - \exp\left[-NTU_I(1-R)\right]}{1 - R\exp\left[-NTU_I(1-R)\right]} = 0.6547$$

- Heat transfer rate: P-NTU$_I$ method.[20]

$$q = PC_I\,\delta T_i = 0.6547\,[0.7648\text{ MBtu/(h °F)}]\,(36.3\,°\text{F}) = 18.175\text{ MBtu/h} \quad (12.11\%\text{ decrease})$$

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[20] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.6547(36.3°\text{F})/1.678 = 14.165°\text{F}$ (12.07% decrease)
$q = UA_s\,EMTD = [348.1\text{ Btu/(h ft}^2\text{ °F)}](3687\text{ ft}^2)(14.165°\text{F}) = 18.175$ MBtu/h

F correction factor: $\Delta T_1 = 116.3°\text{F} - 101.98°\text{F} = 14.32°\text{F}$, $\Delta T_2 = 97.45°\text{F} - 80°\text{F} = 17.45°\text{F}$.
$NTU_{I,cf} = (1-R)^{-1}\ln[(1-PR)/(1-P)] = 1.678$ $F = NTU_{I,cf}/NTU_I = 1.0$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 14.165°\text{F}$ $F\,LMTD_{cf} = 14.165°\text{F} \equiv EMTD$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 80°\text{F} + \frac{18.175\ \text{MBtu/h}}{0.7648\ \text{MBtu/(h °F)}} = 103.76°\text{F} \qquad T_{t,\text{ave}} = 91.88°\text{F}$$

$$T_{s,o} = 116.3°\text{F} - \frac{18.175\ \text{MBtu/h}}{0.8921\ \text{MBtu/(h °F)}} = 95.93°\text{F} \qquad T_{s,\text{ave}} = 106.11°\text{F}$$

- Refined analysis: The average terminal temperatures differ from the values used to estimate the properties by 5.0% for $T_{t,\text{ave}}$ and 0.435% for $T_{s,\text{ave}}$. The analysis is readily refined by specifying the properties at $T_{t,\text{ave}} = 91.88°\text{F}$ and $T_{s,\text{ave}} = 106.11°\text{F}$, with the following results.

$\rho_t = 62.035\ \text{lb}_\text{m}/\text{ft}^3$ | $\rho_s = 61.808\ \text{lb}_\text{m}/\text{ft}^3$

$c_{P,t} = 0.9973\ \text{Btu/(lb}_\text{m}\ \text{°F)}$ | $c_{P,s} = 0.9985\ \text{Btu/(lb}_\text{m}\ \text{°F)}$

$k_t = 0.3552\ \text{Btu/(h ft °F)}$ | $k_s = 0.3615\ \text{Btu/(h ft °F)}$

$\mu_t = 1.8143\ \text{lb}_\text{m}/(\text{ft h})$ | $\mu_s = 1.538\ \text{lb}_\text{m}/(\text{ft h})$

$Pr_t = 5.0941$ | $Pr_s = 4.250$

$h_t = 1229.1\ \text{Btu/(h ft}^2\ \text{°F)}$

$h_s = 1925.3\ \text{Btu/(h ft}^2\ \text{°F)}$

$U = 351.0\ \text{Btu/(h ft}^2\ \text{°F)}$

- Classical calculations with shell-side reference stream:

  C_t = 0.76543 MBtu/(h °F)

  C_s = 0.89216 MBtu/(h °F)

  $NTU_I = NTU_s$ = 1.6908 MBtu/(h °F)

  R = 0.8579

  P = 0.6564

  q = 18.241 MBtu/h

  $EMTD$ = 14.09°F

  $T_{t,o}$ = 103.83°F

  $T_{s,o}$ = 95.85°F

  $T_{t,\mathrm{ave}}$ = 91.915°F

  $T_{s,\mathrm{ave}}$ = 106.08°F

These calculations represent changes relative to the first approximation of 2.50% for h_t, 0.052% for h_s, 0.83% for U, and 0.36% for q. The resulting differences in average terminal temperatures are only 0.038% for $T_{t,ave}$ and –0.028% for $T_{s,ave}$. Further refinement based on use of the property correction factors $\phi_{h,t}$ and $\phi_{h,s}$ result in a negligibly small change in q.

5.4 Example 4. Turbine Building Cooling Water Heat Exchanger: TEMA 1-4 G

A TEMA 1-4 G shell-and-tube heat exchanger is used in a turbine building cooling water system. Design and thermal performance specifications for the heat exchanger are shown in Figure 5-4 and Tables 5-7 and 5-8. The P-NTU$_I$ method is to be used to develop the following performance analysis calculations:

- Calculation of actual design thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for 10% plugged tubes

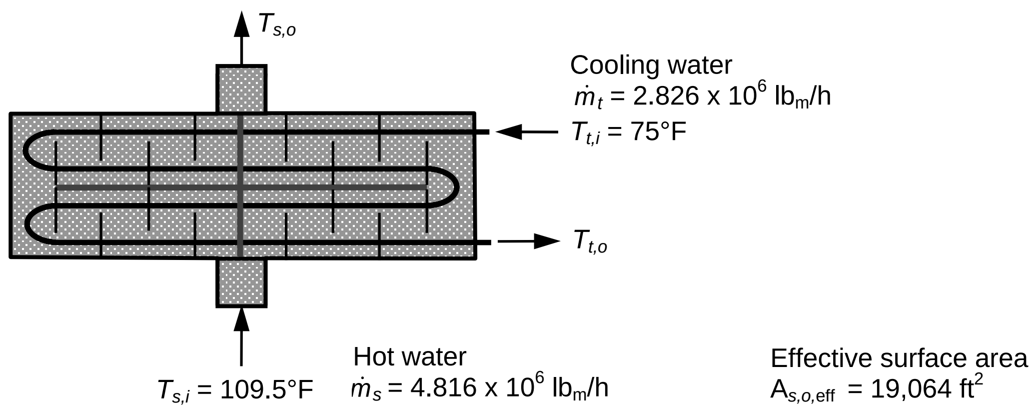

Figure 5-4
Turbine Building Cooling Water Heat Exchanger Design Conditions

Table 5-7
Design Specifications: 1-4 G Shell and Tube

Tubing	Shell-Side Characteristics
Admiralty brass	Triangular arrangement
k_w = 63.1 Btu/(h ft °F)	N = 2248
L_{eff} = 37.02 ft	S_t = 1.09375 in.
d_i = 0.777 in.	$D_{i,s}$ = 65 in., L_b = 45.2 in.
d_o = 0.875 in.	N_b = 8, B_c = 25%, single-segmental baffles

Heat transfer overdesign: 6.6% qDES = 75.37 MBtu/h

Table 5-8
Thermal Performance Specifications: 1-4 G Shell and Tube

Thermal Performance Specifications	
$F_{f,t}$ = 0.00075 h ft^2 °F/Btu	$T_{t,i,\text{des}}$ = 75°F
$F_{f,s}$ = 0.0005 h ft^2 °F/Btu	$T_{t,o,\text{des}}$ = 100°F
q_{des} = 70.7 MBtu/h	$T_{t,\text{ave}}$ = 87.5°F
U_{des} = 285 Btu/(h ft^2 °F)	$T_{s,i,\text{des}}$ = 109.5°F
$EMTD_{\text{des}}$ = 13.0°F	$T_{s,o,\text{des}}$ = 95°F
	$T_{s,ave}$ = 102°F

5.4.1 Calculation of Actual Thermal Performance Specifications

- Reference surface: The outer surface is selected for reference.

$$A_s = A_{s,o,\text{eff}} = N(\pi d_o L_{\text{eff}}) = 2248[\pi(0.875\text{ ft}/12)](37.02\text{ ft}) = 19{,}064\text{ ft}^2$$

- Properties: k_w = 63.1 Btu/(h ft °F) for admiralty brass.

Fluid properties are specified at average temperatures.

Tube-side at $T_{t,\text{ave}}$ = 87.5°F

ρ_t = 62.10 lb$_\text{m}$/ft^3

$c_{P,t}$ = 0.9965 Btu/(lb$_\text{m}$ °F)

k_t = 0.3532 Btu/(h ft °F)

μ_t = 1.9135 lb$_\text{m}$/(ft h)

Pr_t = 5.399

Shell-side at $T_{s,\text{ave}}$ = 102.25°F

ρ_s = 61.87 lb$_\text{m}$/ft^3

$c_{P,s}$ = 0.9982 Btu/(lb$_\text{m}$ °F)

k_s = 0.3597 Btu/(h ft °F)

μ_s = 1.607 lb$_\text{m}$/(ft h)

Pr_s = 4.460

- Approximate capacity rates:

$$C_t = (\dot{m}\, c_P)_t = (2.826 \times 10^6\text{ lb}_\text{m}/\text{h})[0.9965\text{ Btu}/(\text{lb}_\text{m}\text{ °F})] = 2.816\text{ MBtu}/(\text{h °F})$$

$$C_s = (\dot{m}\, c_P)_s = (4.816 \times 10^6\text{ lb}_\text{m}/\text{h})[0.9982\text{ Btu}/(\text{lb}_\text{m}\text{ °F})]$$

$$= 4.807\text{ MBtu}/(\text{h °F})\text{, based on total flow rate}$$

- Actual design outlet temperatures and terminal temperature differences:

  Set $q = q_{\mathrm{DES}} = 75.37$ MBtu/h

  $$T_{t,o} = 75°\mathrm{F} + \frac{75.37\ \mathrm{MBtu/h}}{2.816\ \mathrm{MBtu/(h\ °F)}} = 101.76°\mathrm{F} \qquad \delta T_t = T_{t,o} - T_{t,i} = 26.765°\mathrm{F}$$

  $$T_{s,o} = 109.5°\mathrm{F} - \frac{75.37\ \mathrm{MBtu/h}}{4.807\ \mathrm{MBtu/(h\ °F)}} = 93.82°\mathrm{F} \qquad \delta T_s = T_{s,i} - T_{s,o} = 15.68°\mathrm{F}$$

  $\delta T_i = T_{s,i} - T_{t,i} = 109.5°\mathrm{F} - 75°\mathrm{F} = 34.5°\mathrm{F}$

- Classical approach: Tube-side fluid is selected for reference.

  $C_I = C_t = 2.816$ MBtu/(h °F) $\qquad C_{II} = C_s = 4.807$ MBtu/(h °F)

- Independent performance parameters:

  $$P = \frac{\delta \mathrm{T}_I}{\delta \mathrm{T}_i} = \frac{\delta \mathrm{T}_t}{\delta \mathrm{T}_i} = \frac{26.765°\mathrm{F}}{34.5°\mathrm{F}} = 0.7758$$

  $$R = \frac{\delta \mathrm{T}_{II}}{\delta \mathrm{T}_I} = \frac{\delta \mathrm{T}_s}{\delta \mathrm{T}_t} = \frac{15.68°\mathrm{F}}{26.765°\mathrm{F}} = 0.5858$$

- Number of transfer units: 1-4 G heat exchanger (Table 2-2, $C_I = C_t$, approximated by implicit solution relation for 1-2 G arrangement).[21]

$$\alpha = \exp\left[-NTU_t(2R+1)/4\right]$$

$$\beta = \exp\left[-NTU_t(2R-1)/2\right]$$

$$A = -2(1-\alpha)^2/(2R+1)$$

$$B = [4-\beta(2+1/R)]/(2-1/R)$$

$$P = \frac{1}{R}\left(\frac{B-\alpha^2}{A+B/R+2}\right)$$

These relations are solved by simple iteration to obtain $\alpha = 0.26834$, $\beta = 0.81227$, $A = -0.49307$, $B = 3.3757$, and $NTU_t = 2.4233$.

- Actual overall coefficient of heat transfer: P-NTU_I method.[22]

$$U_{\mathrm{DES}} = NTU_I \frac{C_I}{A_s} = NTU_t \frac{C_t}{A_s} = 2.4233 \ \frac{2.816 \text{ MBtu/(h °F)}}{19{,}064 \text{ ft}^2}$$

$$= 357.95 \text{ Btu/(h ft}^2 \text{ °F)} \qquad \text{(25.60\% difference)}$$

- Effective mean temperature difference:[23]

$$EMTD_{\mathrm{DES}} = \frac{\delta T_I}{NTU_I} = \frac{\delta T_t}{NTU_t} = \frac{26.765\text{°F}}{2.4233} = 11.045\text{°F} \qquad \text{(15.04\% difference)}$$

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[21] The classical solution relation for 1-4 G arrangements is not available in the literature. The practical numerical approach introduced in *Heat Transfer—Professional Version* provides a means for eliminating the analytical error associated with the use of this approximation and for accounting for the effect of heat transfer across the horizontal baffle [2].

[22] EMTD method: $q = UA_s\, EMTD$ = [357.95 Btu/(h ft^2 °F)](19,064 ft^2)(11.045°F) = 75.37 MBtu/h.

[23] F correction factor: ΔT_1 = 109.5°F – 101.76°F = 7.74°F, ΔT_2 = 93.82°F – 75°F = 18.82°F.
$NTU_{I,cf} = (1-R)^{-1} \ln[(1-PR)/(1-P)] = 2.1469$ $\quad F = NTU_{I,cf}/NTU_I = 0.8859$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$ = 12.47°F $\quad F\, LMTD_{cf}$ = 11.045°F ≡ $EMTD$

5.4.2 Back-Calculation of $h_{s,DES}$

The shell-side convection coefficient $h_{s,\mathrm{DES}}$ is back-calculated based on the specified value of U_{DES} corresponding to q_{DES}.

- Wall and fouling resistance: $s_t = d_o/d_i = 0.875/0.777 = 1.126$, $s_s = 1$.

$$r_w = \frac{d_o}{2k_w}\ln\frac{d_o}{d_i} = \frac{0.875\ \mathrm{ft}/12}{2\ [63.1\ \mathrm{Btu/(h\ ft\ °F)}]}\ln\frac{0.875}{0.777} = 0.00006863\ \mathrm{h\ ft^2\ °F/Btu}$$

$$r_f = s_t F_{f,t} + s_s F_{f,s} = 1.126(0.00075) + 0.0005 = 0.0013445\ \mathrm{h\ ft^2\ °F/Btu}$$

- Clean overall coefficient of heat transfer:

$$\frac{1}{U_{C,\mathrm{DES}}} = \frac{1}{U_{\mathrm{DES}}} - r_f = \frac{1}{357.95} - 0.0013445 = 0.0014492\ \mathrm{h\ ft^2\ °F/Btu}$$

$$U_{C,\mathrm{DES}} = 690.0\ \mathrm{Btu/(h\ ft^2\ °F)}$$

- Tube-side convection coefficient $h_{t,\mathrm{DES}}$:

$$A_t = \frac{\pi d_i^2}{4}\frac{N}{N_{tp}} = \frac{\pi(0.777\ \mathrm{ft}/12)^2}{4}\frac{2248}{4} = 1.851\ \mathrm{ft^2}$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{2.826\times10^6\ \mathrm{lb_m/h}}{1.851\ \mathrm{ft^2}} = 1.527\times10^6\ \mathrm{lb_m/(ft^2\ h)}$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 6.831\ \mathrm{ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.527\times10^6\ \mathrm{lb_m/(ft^2\ h)}](0.777\ \mathrm{ft}/12)}{1.9135\ \mathrm{lb_m/(ft\ h)}} = 51{,}680$$

$$Nu_t = \frac{(f_t/2)\ Re_t\ Pr_t}{1.07 + 12.7\sqrt{f_t/2}\ (Pr_t^{2/3} - 1)} = 300.4$$

$$f_t = (1.58\ln Re_t - 3.28)^{-2} = 0.005200$$

$$h_{t,\mathrm{DES}} = Nu_t\frac{k_t}{d_i} = 300.4\ \frac{0.3532\ \mathrm{Btu/(h\ ft\ °F)}}{0.777\ \mathrm{ft}/12} = 1639\ \mathrm{Btu/(h\ ft^2\ °F)}$$

- Shell-side convection coefficient $h_{s,\mathrm{DES}}$:

$$\frac{1}{h_{s,\mathrm{DES}}} = \frac{1}{s_s}\left(\frac{1}{U_{C,\mathrm{DES}}} - \frac{s_t}{h_{t,\mathrm{DES}}} - r_w\right) = \frac{1}{U_{C,\mathrm{DES}}} - \frac{s_t}{h_{t,\mathrm{DES}}} - r_w$$

$= 0.0069357$ h ft^2 °F/Btu

$h_{s,\mathrm{DES}} = 1441.8$ Btu/(h ft^2 °F)

5.4.3 Rating-Calculation for 10% Plugged Tubes

The properties are assumed to be unchanged as a first approximation, so that the capacity rates, shell-side convection coefficient, and wall/fouling resistances are specified by the design-point values.

- Number of active tubes (rounded to nearest even number):

$N_{at} = N(1 - PLUG) = 2248(1 - 0.1) = 2024$ (10% decrease)

- Reference surface area:

$A_s = A_{s,o,\text{eff}}\,(1 - PLUG) = 19{,}064(1 - 0.1) = 17{,}160\ \text{ft}^2$ (10% decrease)

- Tube-side convection coefficient h_t:

$$A_t = \frac{\pi d_i^2}{4}\frac{N_{at}}{N_{tp}} = \frac{\pi(0.777\ \text{ft}/12)^2}{4}\frac{2024}{4} = 1.666\ \text{ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{2.826\times10^6\ \text{lb}_\text{m}/\text{h}}{1.666\ \text{ft}^2} = 1.696\times10^6\ \text{lb}_\text{m}/(\text{ft}^2\ \text{h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 7.59\ \text{ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.696\times10^6\ \text{lb}_\text{m}/(\text{ft}^2\ \text{h})](0.777\ \text{ft}/12)}{1.9135\ \text{lb}_\text{m}/(\text{ft h})} = 57{,}400$$

$$f_t = (1.58\ \ln Re_t - 3.28)^{-2} = 0.005078$$

$$Nu_t = \frac{(f_t/2)\ Re_t\ Pr_t}{1.07 + 12.7\sqrt{f_t/2}\ (Pr_t^{2/3} - 1)} = 327.9$$

$$h_t = Nu_t\frac{k_t}{d_i} = 327.9\ \frac{0.3532\ \text{Btu/(h ft °F)}}{0.777\ \text{ft}/12} = 1789\ \text{Btu/(h ft}^2\ \text{°F)}$$ (9.15% increase)

- Shell-side convection coefficient h_s:

$h_s = h_{s,\text{DES}} = 1441.8\ \text{Btu/(h ft}^2\ \text{°F)}$ (essentially unchanged)

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.126}{1789} + 0.00006863 + \frac{1}{1441.8} = 0.0013916 \text{ h ft}^2 \text{ °F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.0013916 + 0.0013445 = 0.002736 \text{ h ft}^2 \text{ °F/Btu}$$

$U_C = 718.6$ Btu/(h ft^2 °F)

$U = 365.48$ Btu/(h ft^2 °F) (2.1% increase)

- Classical approach: Tube-side fluid is retained as reference.

$C_I = C_t = 2.816$ MBtu/(h °F) $C_{II} = C_s = 4.807$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_t = \frac{UA_s}{C_t} = \frac{[365.48 \text{ Btu/(h ft}^2 \text{ °F)}](17{,}160 \text{ ft}^2)}{2.816 \text{ MBtu/(h °F)}} = 2.227$$

$$R = \frac{C_t}{C_s} = 0.5858$$ (essentially unchanged)

- Effectiveness: 1-4 G heat exchanger (Table 2-2, $C_I = C_t$, approximated by implicit solution for 1-2 G arrangement).

$$\alpha = \exp\left[-NTU_t(2R+1)/4\right] = 0.2985$$

$$\beta = \exp\left[-NTU_t(2R-1)/2\right] = 0.8261$$

$$A = -2(1-\alpha)^2/(2R+1) = -0.4532$$

$$B = [4 - \beta(2 + 1/R)]/(2 - 1/R) = 3.201$$

$$P = \frac{1}{R}\left(\frac{B - \alpha^2}{A + B/R + 2}\right) = 0.7577$$

- Heat transfer rate: P-NTU_I method.[24]

$$q = PC_I\,\delta T_i = 0.7577\,[2.816\text{ MBtu/(h °F)}]\,(34.5\text{°F}) = 73.61\text{ MBtu/h} \qquad (2.33\%\text{ decrease})$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 75\text{°F} + \frac{73.61\text{ MBtu/h}}{2.816\text{ MBtu/(h °F)}} = 101.14\text{°F} \qquad T_{t,\mathrm{ave}} = 88.07\text{°F}$$

$$T_{s,o} = 109.5\text{°F} - \frac{73.61\text{ MBtu/h}}{4.807\text{ MBtu/(h °F)}} = 94.19\text{°F} \qquad T_{s,\mathrm{ave}} = 101.84\text{°F}$$

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[24] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.7577(34.5\text{°F})/2.227 = 11.738\text{°F}$ (6.27% increase)
$q = UA_s\,EMTD = [365.48\text{ Btu/(h ft}^2\text{ °F)}](17{,}160\text{ ft}^2)(11.738\text{°F}) = 73.615\text{ MBtu/h}$

F correction factor: $\Delta T_1 = 109.5\text{°F} - 101.14\text{°F} = 8.36\text{°F}$, $\Delta T_2 = 94.19\text{°F} - 75\text{°F} = 19.19\text{°F}$.
$NTU_{I,cf} = (1-R)^{-1}\ln[(1-PR)/(1-P)] = 2.006$ $\qquad F = NTU_{I,cf}/NTU_I = 0.9007$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 13.03\text{°F}$ $\qquad F\,LMTD_{cf} = 11.739\text{°F} \equiv EMTD$

5.5 Example 5. Decay Heat Removal Heat Exchanger: TEMA 1-2 J

A TEMA 1-2 J shell-and-tube heat exchanger is used in a decay heat removal system. Design and thermal performance specifications for the heat exchanger are shown in Figure 5-5 and Tables 5-9 and 5-10. The P-NTU$_I$ method is to be used to develop the following performance analysis calculations:

- Calculation of actual design thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for off-design inlet temperatures and mass-flow rates

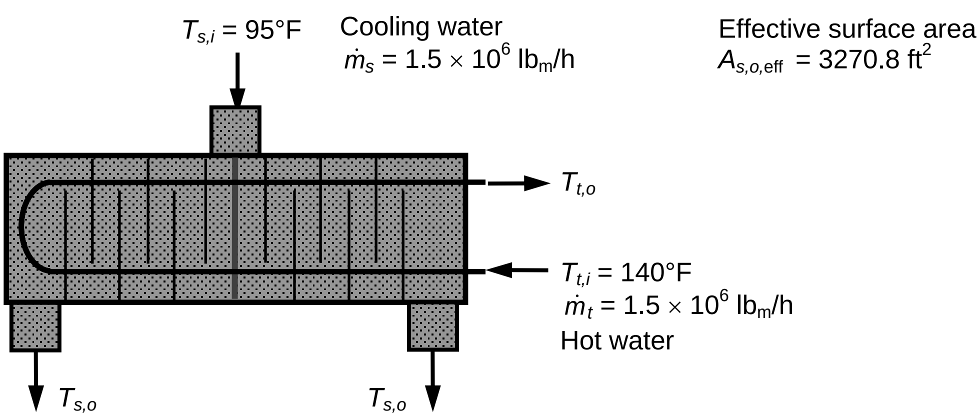

Figure 5-5
Decay Heat Removal Heat Exchanger Design Conditions

Table 5-9
Design Specifications: 1-2 J Shell and Tube

Tubing	Shell-Side Characteristics
304 Stainless Steel, 18 BWG	Triangular arrangement
k_w = 10 Btu/(h ft °F)	N = 946
L_{eff} = 17.609 ft	S_t = 0.9375 in.
d_i = 0.652 in.	$D_{i,s}$ = 37 in., L_b = 15.43 in.
d_o = 0.75 in.	N_b = 12, B_c = 20%, single-segmental baffles

Table 5-10
Thermal Performance Specifications: 1-2 J Shell and Tube

Thermal Performance Specifications	
r_f = 0.00036855 h ft^2 °F/Btu	$T_{t,i,\text{des}}$ = 140°F
q_{des} = 29.65 MBtu/h	$T_{t,o,\text{des}}$ = 120.24°F
U_{des} = 407 Btu/(h ft^2 °F)	$T_{t,\text{ave}}$ = 130.12°F
$EMTD_{\text{des}}$ = 22.2°F	$T_{s,i,\text{des}}$ = 95°F
	$T_{s,o,\text{des}}$ = 114.76°F
	$T_{s,ave}$ = 104.88°F

Heat transfer overdesign: 4.42%; q_{DES} = 30.96 MBtu/h.
Value of r_f corresponds to CF = 0.85 based on U_{des}.

5.5.1 Calculation of Actual Thermal Performance Specifications

- Reference surface: The outer surface is selected for reference.

  $A_s = A_{s,o,\text{eff}} = N(\pi d_o L_{\text{eff}}) = 946[\pi(0.75 \text{ ft}/12)](17.609 \text{ ft}) = 3270.8 \text{ ft}^2$

- Properties: k_w = 10 Btu/(h ft °F) for 304 stainless steel.

  Fluid properties are specified at average temperatures.

  Tube-side at $T_{t,\text{ave}}$ = 130.12°F — Shell-side at $T_{s,\text{ave}}$ = 104.88°F

  $\rho_t = 61.42 \text{ lb}_m/\text{ft}^3$ — $\rho_s = 61.83 \text{ lb}_m/\text{ft}^3$

  $c_{P,t} = 1.001 \text{ Btu/(lb}_m \text{ °F)}$ — $c_{P,s} = 0.9984 \text{ Btu/(lb}_m \text{ °F)}$

  $k_t = 0.3721 \text{ Btu/(h ft °F)}$ — $k_s = 0.3609 \text{ Btu/(h ft °F)}$

  $\mu_t = 1.197 \text{ lb}_m/(\text{ft h})$ — $\mu_s = 1.560 \text{ lb}_m/(\text{ft h})$

  $Pr_t = 3.220$ — $Pr_s = 4.3156$

- Approximate capacity rates:

  $C_t = (\dot{m}\, c_P)_t = (1.5 \times 10^6 \text{ lb}_m/\text{h})[1.001 \text{ Btu/(lb}_m \text{ °F)}] = 1.5015 \text{ MBtu/(h °F)}$

  $C_s = (\dot{m}\, c_P)_s = (1.5 \times 10^6 \text{ lb}_m/\text{h})[0.9984 \text{ Btu/(lb}_m \text{ °F)}]$

  $= 1.4976$ MBtu/(h °F), based on total flow rate

- Actual design outlet temperatures and terminal temperature differences:

  Set $q = q_{\mathrm{DES}} = 30.96$ MBtu/h

  $$T_{t,o} = 140°\mathrm{F} - \frac{30.96\ \mathrm{MBtu/h}}{1.5015\ \mathrm{MBtu/(h\ °F)}} = 119.38°\mathrm{F} \qquad \delta T_t = T_{t,i} - T_{t,o} = 20.62°\mathrm{F}$$

  $$T_{s,o} = 95°\mathrm{F} + \frac{30.96\ \mathrm{MBtu/h}}{1.4976\ \mathrm{MBtu/(h\ °F)}} = 115.67°\mathrm{F} \qquad \delta T_s = T_{s,o} - T_{s,i} = 20.67°\mathrm{F}$$

  $\delta T_i = T_{t,i} - T_{s,i} = 140°\mathrm{F} - 95°\mathrm{F} = 45°\mathrm{F}$

- Classical approach: Shell-side fluid is selected for reference.

  $C_I = C_s = 1.4976$ MBtu/(h °F) $\qquad C_{II} = C_t = 1.5015$ MBtu/(h °F)

- Independent performance parameters:

  $$P = \frac{\delta T_I}{\delta T_i} = \frac{\delta T_s}{\delta T_i} = \frac{20.67°\mathrm{F}}{45°\mathrm{F}} = 0.4593$$

  $$R = \frac{\delta T_{II}}{\delta T_I} = \frac{\delta T_t}{\delta T_s} = \frac{20.62°\mathrm{F}}{20.67°\mathrm{F}} = 0.9976$$

- Number of transfer units: 1-2 J exchanger (Table 2-2, $C_I = C_s$ implicit relation for NTU_s).

  $$P = \frac{1}{1 + \lambda B - 2\lambda CD + R/2}$$

  $$\lambda = \sqrt{1 + R^2/4}$$

  $$A = \exp\,(NTU_s)$$

  $$B = (A^\lambda + 1)/(A^\lambda - 1)$$

  $$C = A^{(1+\lambda)/2}/[\lambda - 1 + (\lambda + 1)A^\lambda]$$

  $$D = 1 + \lambda A^{(\lambda - 1)/2}/(A^\lambda - 1)$$

  These relations are solved by simple iteration to obtain $\lambda = 1.1175$, $A = 2.6709$, $B = 2.0012$, $C = 0.43767$, $D = 1.59264$, and $NTU_s = 0.9824$ for the specified values of P and R.

- Actual overall coefficient of heat transfer: P-NTU_I method.[25]

$$U_{\mathrm{DES}} = NTU_I \frac{C_I}{A_s} = NTU_s \frac{C_s}{A_s} = 0.9824 \ \frac{1.4976 \ \mathrm{MBtu/(h\ {}^\circ F)}}{3270.8 \ \mathrm{ft}^2}$$

$$= 449.8 \ \mathrm{Btu/(h\ ft^2\ {}^\circ F)} \qquad \text{(10.52\% difference)}$$

- Effective mean temperature difference:[26]

$$EMTD_{\mathrm{DES}} = \frac{\delta T_I}{NTU_I} = \frac{\delta T_s}{NTU_s} = \frac{27.67^\circ\mathrm{F}}{0.9824} = 21.04^\circ\mathrm{F} \qquad \text{(5.23\% difference)}$$

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[25] $CF_{\mathrm{DES}} = U_{\mathrm{DES}}/U_{C,\mathrm{DES}} = 1 - U_{\mathrm{DES}}\, r_f = 0.8342$.

[26] F correction factor: $\Delta T_1 = 140^\circ\mathrm{F} - 115.67^\circ\mathrm{F} = 24.33^\circ\mathrm{F}$, $\Delta T_2 = 119.38^\circ\mathrm{F} - 95^\circ\mathrm{F} = 24.38^\circ\mathrm{F}$.
$NTU_{I,cf} = (1-R)^{-1} \ln[(1-PR)/(1-P)] = 0.8486$ $\qquad F = NTU_{I,cf}/NTU_I = 0.8638$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 24.36^\circ\mathrm{F}$ $\qquad F\, LMTD_{cf} = 21.04^\circ\mathrm{F} \equiv EMTD$

5.5.2 Back-Calculation of $h_{s,DES}$

The shell-side convection coefficient $h_{s,\mathrm{DES}}$ is back-calculated based on the specified value of U_{DES} corresponding to q_{DES}.

- Wall and fouling resistance: $s_t = d_o/d_i = 0.75/0.652 = 1.1503$, $s_s = 1$.

$$r_w = \frac{d_o}{2k_w}\ln\frac{d_o}{d_i} = \frac{0.75\text{ ft}/12}{2\,[10\text{ Btu/(h ft °F)}]}\ln\frac{0.75}{0.652} = 0.0004376\text{ h ft}^2\text{ °F/Btu}$$

$r_{f,\mathrm{DES}} = r_{f,\mathrm{des}} = 0.00036855$ h ft^2 °F/Btu

- Clean overall coefficient of heat transfer:

$$\frac{1}{U_{C,\mathrm{DES}}} = \frac{1}{U_{\mathrm{DES}}} - r_f = \frac{1}{449.8} - 0.00036855 = 0.0018547\text{ h ft}^2\text{ °F/Btu}$$

$U_{C,\mathrm{DES}} = 539.2$ Btu/(h ft^2 °F)

- Tube-side convection coefficient $h_{t,\mathrm{DES}}$:

$$A_t = \frac{\pi d_i^2}{4}\frac{N}{N_{tp}} = \frac{\pi(0.652\text{ ft}/12)^2}{4}\frac{946}{2} = 1.097\text{ ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{1.5\times10^6\text{ lb}_\mathrm{m}/\text{h}}{1.097\text{ ft}^2} = 1.367\times10^6\text{ lb}_\mathrm{m}/(\text{ft}^2\text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 6.18\text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.367\times10^6\text{ lb}_\mathrm{m}/(\text{ft}^2\text{ h})](0.652\text{ ft}/12)}{1.197\text{ lb}_\mathrm{m}/(\text{ft h})} = 62{,}050$$

$$f_t = (1.58\ln Re_t - 3.28)^{-2} = 0.004990$$

$$Nu_t = \frac{(f_t/2)\,Re_t\,Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 274.1$$

$$h_{t,\mathrm{DES}} = Nu_t\frac{k_t}{d_i} = 274.1\,\frac{0.3721\text{ Btu/(h ft °F)}}{0.652\text{ ft}/12} = 1877\text{ Btu/(h ft}^2\text{ °F)}$$

- Shell-side convection coefficient $h_{s,\mathrm{DES}}$:

$$\frac{1}{h_{s,\mathrm{DES}}}=\frac{1}{s_s}\left(\frac{1}{U_{C,\mathrm{DES}}}-\frac{s_t}{h_{t,\mathrm{DES}}}-r_w\right)=\frac{1}{U_{C,\mathrm{DES}}}-\frac{s_t}{h_{t,\mathrm{DES}}}-r_w$$

$$=0.0018547-\frac{1.1503}{1877}-0.0004376=0.00080426\ \text{h ft}^2\ {}^\circ\text{F/Btu}$$

$h_{s,\mathrm{DES}} = 1243.4$ Btu/(h ft^2 °F)

5.5.3 Rating-Calculation for Off-Design Inlet Temperatures and Mass-Flow Rates

$T_{t,i}$ = 250°F, $T_{s,I}$ = 105°F, $\dot{m}_t = 1.415 \times 10^6$ lb$_m$/h, $\dot{m}_s = 1.49 \times 10^6$ lb$_m$/h

- Properties: Fluid properties are specified at the inlet temperatures as a first approximation.

Tube-side at $T_{t,i}$ = 250°F	Shell-side at $T_{s,i}$ = 105°F
ρ_t = 58.80 lb$_m$/ft^3	ρ_s = 61.93 lb$_m$/ft^3
$c_{P,t}$ = 1.014 Btu/(lb$_m$ °F)	$c_{P,s}$ = 0.9960 Btu/(lb$_m$ °F)
k_t = 0.3949 Btu/(h ft °F)	k_s = 0.3636 Btu/(h ft °F)
μ_t = 0.5586 lb$_m$/(ft h)	μ_s = 1.561 lb$_m$/(ft h)
Pr_t = 1.434	Pr_s = 4.285

- Inlet terminal temperature difference and capacity rates:

$\delta T_i = T_{t,i} - T_{s,i}$ = 250°F − 105°F = 145°F (222.2% increase)

$C_t = (\dot{m}\, c_P)_t = (1.415 \times 10^6$ lb$_m$/h)[1.014 Btu/(lb$_m$ °F)]

= 1.4348 MBtu/(h °F) (4.44% decrease)

$C_s = (\dot{m}\, c_P)_s = (1.49 \times 10^6$ lb$_m$/h)[0.9960 Btu/(lb$_m$ °F)]

= 1.484 MBtu/(h °F), based on total flow rate (0.91% decrease)

- Tube-side convection coefficient h_t: $A_t = 1.097$ ft^2.

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{1.415\times 10^6 \text{ lb}_\text{m}/\text{h}}{1.097 \text{ ft}^2} = 1.290 \times 10^6 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 6.09 \text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[1.290\times 10^6 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})](0.652 \text{ ft}/12)}{0.5586 \text{ lb}_\text{m}/(\text{ft h})} = 125{,}500$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} = 0.004289$$

$$Nu_t = \frac{(f_t/2)\, Re_t\, Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 313.84$$

$$h_t = Nu_t \frac{k_t}{d_i} = 313.84 \frac{0.3949 \text{ Btu/(h ft °F)}}{0.652 \text{ ft}/12} = 2281 \text{ Btu/(h ft}^2 \text{ °F)} \qquad \text{(21.52\% increase)}$$

- Shell-side convection coefficient h_s: Set $h_{s,\text{DES}}$ as first approximation.

$$h_s = h_{s,\text{DES}} = 1243.4 \text{ Btu/(h ft}^2 \text{ °F)}$$

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.1503}{2281} + 0.0004376 + \frac{1}{1243.4} = 0.0017461 \text{ h ft}^2 \text{ °F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.0017461 + 0.00036855 = 0.00211455 \text{ h ft}^2 \text{ °F/Btu}$$

$U_C = 572.7$ Btu/(h ft^2 °F) $\qquad$ $U = 472.9$ Btu/(h ft^2 °F) $\qquad$ (5.15% increase)

- Classical approach: Shell-side fluid is retained as reference.

$C_I = C_s = 1.484$ MBtu/(h °F) $\qquad$ $C_{II} = C_t = 1.4348$ MBtu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_s = \frac{UA_s}{C_s} = \frac{[472.9 \text{ Btu/(h ft}^2 \text{ °F)}](3270.8 \text{ ft}^2)}{1.484 \text{ MBtu/(h °F)}} = 1.0423$$

$$R = \frac{C_s}{C_t} = \frac{1.484 \text{ MBtu/(h °F)}}{1.4348 \text{ MBtu/(h °F)}} = 1.0343$$

- Effectiveness: 1-2 J heat exchanger (Table 2-2, $C_I = C_s$).

$$P = \frac{1}{1+\lambda B - 2\lambda CD + R/2}$$

$$\lambda = \sqrt{1+R^2/4}$$

$$A = \exp\,(NTU_s)$$

$$B = (A^\lambda + 1)/(A^\lambda - 1)$$

$$C = A^{(1+\lambda)/2}/[\lambda - 1 + (\lambda + 1)A^\lambda]$$

$$D = 1 + \lambda A^{(\lambda - 1)/2}/(A^\lambda - 1)$$

This explicit solution relation gives rise to $\lambda = 1.1258$, $A = 2.8349$, $B = 1.8961$, $C = 0.43264$, $D = 1.5386$, and $P = 0.46447$ for the specified values of NTU_s and R.

- Heat transfer rate: P-NTU_I method.[27]

$$q = PC_I\,\delta T_i = 0.46447\,[1.484\text{ MBtu/(h °F)}]\,(145\text{°F}) = 99.94\text{ MBtu/h} \quad (222.8\%\text{ increase})$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 250\text{°F} - \frac{99.94\text{ MBtu/h}}{1.4348\text{ MBtu/(h °F)}} = 180.35\text{°F} \qquad T_{t,\text{ave}} = 215.17\text{°F}$$

$$T_{s,o} = 150\text{°F} - \frac{54{,}028\text{ Btu/h}}{2160\text{ Btu/(h °F)}} = 125.0\text{°F} \qquad T_{s,\text{ave}} = 138.67\text{°F}$$

[27] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.46447(145\text{°F})/1.0423 = 64.615\text{°F}$ (207% increase)
$q = UA_s\,EMTD = [472.9\text{ Btu/(h ft}^2\text{ °F)}](3270.8\text{ ft}^2)(64.615\text{°F}) = 99.94$ MBtu/h

F correction factor: $\Delta T_1 = 250\text{°F} - 172.35\text{°F} = 77.65\text{°F}$, $\Delta T_2 = 180.35\text{°F} - 105\text{°F} = 75.35\text{°F}$.
$NTU_{I,cf} = (1-R)^{-1}\ln[(1-PR)/(1-P)] = 0.88047$ $\qquad F = NTU_{I,cf}/NTU_I = 0.84474$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 76.49\text{°F}$ $\qquad F\,LMTD_{cf} = 64.615\text{°F} \equiv EMTD$

- Refined analysis: The analysis gives rise to the following results for fluid properties specified at the average temperatures $T_{t,\text{ave}}$ = 215.17°F and $T_{s,\text{ave}}$ = 138.67°F.

  ρ_t = 60.068 lb_m/ft^3 ρ_s = 61.288 lb_m/ft^3

  $c_{P,t}$ = 1.0079 Btu/(lb_m °F) $c_{P,s}$ = 1.0013 Btu/(lb_m °F)

  k_t = 0.3934 Btu/(h ft °F) k_s = 0.3759 Btu/(h ft °F)

  μ_t = 0.6505 lb_m/(ft h) μ_s = 1.1041 lb_m/(ft h)

  Pr_t = 1.666 Pr_s = 2.941

  h_t = 2193.6 Btu/(h ft^2 °F)

  h_s = 1396.8 Btu/(h ft^2 °F)

  U = 488.6 Btu/(h ft^2 °F)

- Classical calculations with shell-side reference stream:

  C_t = 1.426 MBtu/(h °F)

  C_s = 1.492 MBtu/(h °F)

  $NTU_I = NTU_s$ = 1.0711 MBtu/(h °F)

  R = 1.0461

  P = 0.46724

  q = 101.1 MBtu/h

  $EMTD$ = 63.25°F

  $T_{t,o}$ = 179.11°F

  $T_{s,o}$ = 172.76°F

  $T_{t,\text{ave}}$ = 214.56°F

  $T_{s,\text{ave}}$ = 138.88°F

These calculations represent changes relative to the first approximation of –3.96% for h_t, 12.3% for h_s, 3.32% for U, and 1.16% for q. The resulting differences in average terminal temperatures are only –0.69% for $T_{t,\text{ave}}$ and 0.24% for $T_{s,\text{ave}}$. Further refinement based on use of the property correction factor method introduced in reference 2 result in a difference of less than 0.16% in q.

5.6 Example 6. Lube Oil Cooler: TEMA 1-4 E, Plain Tube

A TEMA 1-4 E shell-and-tube heat exchanger is used for cooling lube oil (SAE 10). Design and thermal performance specifications for the heat exchanger are shown in Figure 5-6 and Tables 5-11 and 5-12. The P-NTU$_I$ method is to be used to develop the following performance analysis calculations:

- Calculation of actual design thermal performance specifications
- Back-calculation of shell-side convection coefficient
- Rating-calculation for 10% plugged tubes

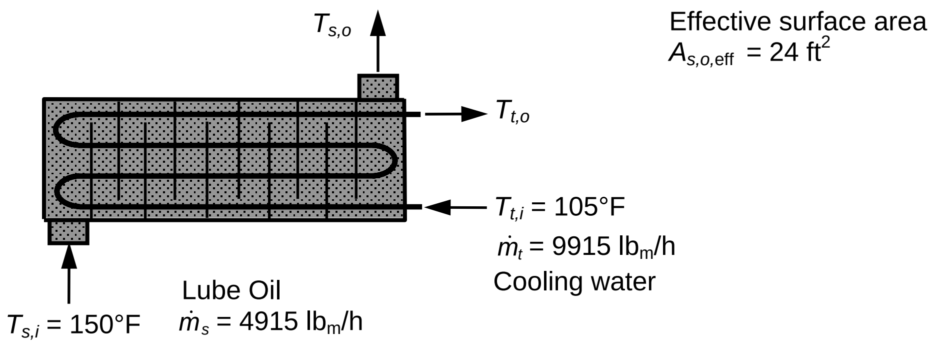

Figure 5-6
Lube Oil Cooler Design Conditions

Table 5-11
Design Specifications: 1-4 E Shell and Tube

Tubing	Shell-Side Characteristics
90/10 copper nickel (CuNi)	Triangular arrangement
k_w = 29.5 Btu/(h ft °F)	N = 80
L_{eff} = 3.056 ft	S_t = 0.45309 in.
d_i = 0.319 in.	$D_{i,s}$ = 5.125 in., L_b = 3.3 in.
d_o = 0.375 in.	N_b = 10, B_c = 20%, single-segmental baffles

Table 5-12
Thermal Performance Specifications: 1-4 E Shell and Tube

Thermal Performance Specifications	
$F_{f,t}$ = 0.001 h ft^2 °F/Btu	$T_{t,i,\text{des}}$ = 105°F
$F_{f,s}$ = 0.0005 h ft^2 °F/Btu	$T_{t,o,\text{des}}$ = 110.4°F
q_{des} = 54 kBtu/h	$T_{t,\text{ave}}$ = 107.7°F
U_{des} = 80.6 Btu/(h ft^2 °F)	$T_{s,i,\text{des}}$ = 150°F
$EMTD_{\text{des}}$ = 27.9°F	$T_{s,o,\text{des}}$ = 125°F
	$T_{s,ave}$ = 137.5°F

Heat transfer overdesign: 5.78%; q_{DES} = 57,120 Btu/h.

5.6.1 Calculation of Actual Thermal Performance Specifications

- Reference surface: The outer surface is selected for reference.

  $A_s = A_{s,o,\text{eff}} = N(\pi d_o L_{\text{eff}}) = 80[\pi(0.375 \text{ ft}/12)](3.056 \text{ ft}) = 24 \text{ ft}^2$

- Properties: k_w = 29.5 Btu/(h ft °F).

  Fluid properties are specified at average temperatures.

Tube-side at $T_{t,\text{ave}}$ = 107.7°F	Shell-side at $T_{s,\text{ave}}$ = 137.7°F
ρ_t = 61.78 lb$_\text{m}$/ft^3	ρ_s = 53.3 lb$_\text{m}$/ft^3
$c_{P,t}$ = 0.9987 Btu/(lb$_\text{m}$ °F)	$c_{P,s}$ = 0.4395 Btu/(lb$_\text{m}$ °F)
k_t = 0.3622 Btu/(h ft °F)	k_s = 0.09 Btu/(h ft °F)
μ_t = 1.509 lb$_\text{m}$/(ft h)	μ_s = 22.96 lb$_\text{m}$/(ft h)
Pr_t = 4.161	Pr_s = 112.12

- Approximate capacity rates:

  $C_t = (\dot{m}\, c_P)_t$ = (9915 lb$_\text{m}$/h)[0.9987 Btu/(lb$_\text{m}$ °F)] = 9902 Btu/(h °F)

  $C_s = (\dot{m}\, c_P)_s$ = (4915 lb$_\text{m}$/h)[0.4395 Btu/(lb$_\text{m}$ °F)] = 2160 Btu/(h °F)

- Actual design outlet temperatures and terminal temperature differences:

  Set $q = q_{\mathrm{DES}} = 57{,}120$ Btu/h.

  $$T_{t,o} = 105°\mathrm{F} + \frac{57{,}120 \text{ Btu/h}}{9902 \text{ Btu/(h °F)}} = 110.77°\mathrm{F} \qquad \delta T_t = T_{t,o} - T_{t,i} = 5.77°\mathrm{F}$$

  $$T_{s,o} = 150°\mathrm{F} - \frac{57{,}120 \text{ Btu/h}}{2160 \text{ Btu/(h °F)}} = 123.56°\mathrm{F} \qquad \delta T_s = T_{s,i} - T_{s,o} = 26.44°\mathrm{F}$$

  $\delta T_i = T_{s,i} - T_{t,i} = 150°\mathrm{F} - 105°\mathrm{F} = 45°\mathrm{F}$

- Classical approach: Shell-side fluid is selected for reference.

  $C_I = C_s = 2160$ Btu/(h °F) $\qquad C_{II} = C_t = 9902$ Btu/(h °F)

- Independent performance parameters:

  $$P = \frac{\delta T_I}{\delta T_i} = \frac{\delta T_s}{\delta T_i} = \frac{26.44°\mathrm{F}}{45°\mathrm{F}} = 0.58741$$

  $$R = \frac{\delta T_{II}}{\delta T_I} = \frac{\delta T_t}{\delta T_s} = \frac{5.77°\mathrm{F}}{26.44°\mathrm{F}} = 0.21825$$

- Number of transfer units: 1-4 E heat exchanger (Table 2-2, $C_I = C_s$ implicit solution relation for NTU_s).

  $$P = \frac{4}{2(1+R) + \sqrt{4+R^2}\coth(\Gamma_3/4) + R\tanh(R\,NTU_s/4)} \qquad \Gamma_3 = NTU_s\sqrt{4+R^2}$$

  This relation is solved by simple iteration to obtain $NTU_s = 0.99205$.[28]

[28] The error in the approximation of NTU_s by use of the 1-2 E shell solution relation proves to be negligible for this application.

- Actual overall coefficient of heat transfer: P-NTU_I method.[29]

$$U_{\mathrm{DES}} = NTU_I \frac{C_I}{A_s} = NTU_s \frac{C_s}{A_s} = 0.99205 \ \frac{2160 \text{ Btu/(h °F)}}{24 \text{ ft}^2}$$

$$= 89.33 \text{ Btu/(h ft}^2 \text{ °F)} \qquad (10.83\% \text{ difference})$$

- Effective mean temperature difference:[30]

$$EMTD_{\mathrm{DES}} = \frac{\delta T_I}{NTU_I} = \frac{\delta T_s}{NTU_s} = \frac{26.44\text{°F}}{0.99205} = 26.65\text{°F} \qquad (4.48\% \text{ difference})$$

[29] EMTD method: $q = UA_s\, EMTD$ = [89.33 Btu/(h ft^2 °F)](24 ft^2)(26.65°F) = 57,130 Btu/h.

[30] F correction factor: ΔT_1 = 150°F – 110.77°F = 39.23°F, ΔT_2 = 123.56°F – 105°F = 18.56°F.
$NTU_{I,cf} = (1-R)^{-1} \ln [(1-PR)/(1-P)] = 0.9570$ $\qquad F = NTU_{I,cf}/NTU_I = 0.9647$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln (\Delta T_1/\Delta T_2) = 27.62$°F $\qquad F\, LMTD_{cf} = 26.65$°F $\equiv EMTD$

5.6.2 Back-Calculation of $h_{s,DES}$

The shell-side convection coefficient $h_{s,\mathrm{DES}}$ is back-calculated based on the specified value of U_{DES} corresponding to q_{DES}.

- Wall and fouling resistance: $s_t = d_o/d_i = 0.375/0.319 = 1.175$, $s_s = 1$.

$$r_w = \frac{d_o}{2k_w}\ln\frac{d_o}{d_i} = \frac{0.375\text{ ft}/12}{2\,[29.5\text{ Btu/(h ft °F)}]}\ln\frac{0.375}{0.319} = 0.00008566\text{ h ft}^2\text{ °F/Btu}$$

$$r_f = s_t F_{f,t} + s_s F_{f,s} = 1.175(0.001) + 0.0005 = 0.001675\text{ h ft}^2\text{ °F/Btu}$$

$$\frac{1}{U_{C,\mathrm{DES}}} = \frac{1}{U_{\mathrm{DES}}} - r_f = \frac{1}{89.33} - 0.001675 = 0.0095194\text{ h ft}^2\text{ °F/Btu}$$

$$U_{C,\mathrm{DES}} = 105.05\text{ Btu/(h ft}^2\text{ °F)}$$

- Tube-side convection coefficient $h_{t,\mathrm{DES}}$:

$$A_t = \frac{\pi d_i^2}{4}\frac{N}{N_{tp}} = \frac{\pi(0.319\text{ ft}/12)^2}{4}\frac{80}{4} = 0.0110\text{ ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{9915\text{ lb}_\mathrm{m}/\mathrm{h}}{0.0110\text{ ft}^2} = 8.932\times 10^5\text{ lb}_\mathrm{m}/(\text{ft}^2\text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 4.016\text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[9915\text{ lb}_\mathrm{m}/(\text{ft}^2\text{ h})](0.319\text{ ft}/12)}{1.509\text{ lb}_\mathrm{m}/(\text{ft h})} = 15{,}735$$

$$f_t = (1.58\ln Re_t - 3.28)^{-2} = 0.006958$$

$$Nu_t = \frac{(f_t/2)\,Re_t\,Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 100.8$$

$$h_{t,\mathrm{DES}} = Nu_t\frac{k_t}{d_i} = 100.8\,\frac{0.3622\text{ Btu/(h ft °F)}}{0.319\text{ ft}/12} = 1373.4\text{ Btu/(h ft}^2\text{ °F)}$$

- Shell-side convection coefficient $h_{s,DES}$:

$$\frac{1}{h_{s,\text{DES}}} = \frac{1}{s_s}\left(\frac{1}{U_{C,\text{DES}}} - \frac{s_t}{h_{t,\text{DES}}} - r_w\right) = \frac{1}{U_{C,\text{DES}}} - \frac{s_t}{h_{t,\text{DES}}} - r_w$$

$$= 0.0095194 - \frac{1.175}{1373.4} - 0.00008566 = 0.008578 \text{ h ft}^2\ \text{°F/Btu}$$

$h_{s,\text{DES}} = 116.57$ Btu/(h ft^2 °F)

5.6.3 Rating-Calculation for 10% Plugged Tubes

The properties are assumed to be unchanged as a first approximation, such that the capacity rates, shell-side convection coefficient, and wall/fouling resistances are specified by the design-point values.

- Number of active tubes:

$$N_{at} = N(1 - PLUG) = 80(1 - 0.1) = 72 \qquad \text{(10\% decrease)}$$

- Reference surface area:

$$A_s = A_{s,o,\text{eff}}\,(1 - PLUG) = 24(1 - 0.1) = 21.6 \text{ ft}^2 \qquad \text{(10\% decrease)}$$

- Tube-side convection coefficient h_t:

$$A_t = \frac{\pi d_i^2}{4}\frac{N_{at}}{N_{tp}} = \frac{\pi(0.319 \text{ ft}/12)^2}{4}\frac{72}{4} = 0.009990 \text{ ft}^2$$

$$G_t = \frac{\dot{m}_t}{A_t} = \frac{9915 \text{ lb}_\text{m}/\text{h}}{0.009990 \text{ ft}^2} = 9.925 \times 10^5 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})$$

$$U_{b,t} = \frac{G_t}{\rho_t} = 4.462 \text{ ft/s}$$

$$Re_t = \frac{G_t d_i}{\mu_t} = \frac{[9.925 \times 10^5 \text{ lb}_\text{m}/(\text{ft}^2 \text{ h})](0.319 \text{ ft}/12)}{1.509 \text{ lb}_\text{m}/(\text{ft h})} = 17{,}484$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} = 0.006768$$

$$Nu_t = \frac{(f_t/2)\, Re_t\, Pr_t}{1.07 + 12.7\sqrt{f_t/2}\,(Pr_t^{2/3} - 1)} = 109.8$$

$$h_t = Nu_t \frac{k_t}{d_i} = 109.8\,\frac{0.3622 \text{ Btu/(h ft °F)}}{0.319 \text{ ft}/12} = 1496 \text{ Btu/(h ft}^2 \text{ °F)} \qquad \text{(8.93\% increase)}$$

- Shell-side convection coefficient h_s:

$h_s = h_{s,\mathrm{DES}} = 116.57$ Btu/(h ft^2 °F) (essentially unchanged)

- Overall coefficient of heat transfer:

$$\frac{1}{U_C} = \frac{s_t}{h_t} + r_w + \frac{s_s}{h_s} = \frac{1.175}{1496} + 0.00008561 + \frac{1}{116.57} = 0.009450 \text{ h ft}^2 \text{ °F/Btu}$$

$$\frac{1}{U} = \frac{1}{U_C} + r_f = 0.009450 + 0.001675 = 0.01112 \text{ h ft}^2 \text{ °F/Btu}$$

$U_C = 105.82$ Btu/(h ft^2 °F) $U = 89.89$ Btu/(h ft^2 °F) (0.63% increase)

- Classical approach: Shell-side fluid is retained as reference.

$C_I = C_s = 2160$ Btu/(h °F) $C_{II} = C_t = 9902$ Btu/(h °F)

- Independent performance parameters:

$$NTU_I = NTU_s = \frac{UA_s}{C_s} = \frac{[89.89 \text{ Btu/(h ft}^2 \text{ °F)}](21.6 \text{ ft}^2)}{2160 \text{ Btu/(h °F)}} = 0.89883$$

$$R = \frac{C_s}{C_t} = 0.21825$$ (essentially unchanged)

- Effectiveness: 1-4 E heat exchanger (Table 2–2, $C_I = C_s$).

$$\Gamma_3 = NTU_s \sqrt{4 + R^2} = 1.8083$$

$$P = \frac{4}{2(1+R) + \sqrt{4+R^2} \coth(\Gamma_3/4) + R \tanh(R\ NTU_s/4)} = 0.55584$$

- Heat transfer rate: P-NTU_I method.[31]

$$q = PC_I\,\delta T_i = 0.55584\,[2160\text{ Btu/(h °F)}]\,(45\text{°F}) = 54{,}028\text{ Btu/h} \qquad (5.41\%\text{ decrease})$$

- Tube- and shell-side outlet temperatures:

$$T_{t,o} = 105\text{°F} + \frac{54{,}028\text{ Btu/h}}{9902\text{ Btu/(h °F)}} = 110.456\text{°F} \qquad T_{t,\text{ave}} = 107.73\text{°F}$$

$$T_{s,o} = 150\text{°F} - \frac{54{,}028\text{ Btu/h}}{2160\text{ Btu/(h °F)}} = 125.0\text{°F} \qquad T_{s,\text{ave}} = 137.5\text{°F}$$

[31] EMTD method:
$EMTD = P\,\delta T_i/NTU_I = 0.55584(45\text{°F})/0.89883 = 27.83\text{°F}$ (4.43% increase)
$q = UA_s\,EMTD = [89.89\text{ Btu/(h ft}^2\text{ °F)}](21.6\text{ ft}^2)(27.83\text{°F}) = 54{,}028\text{ Btu/h}$

F correction factor: $\Delta T_1 = 150\text{°F} - 110.456\text{°F} = 39.544\text{°F}$, $\Delta T_2 = 125.0\text{°F} - 105\text{°F} = 20.0\text{°F}$.
$NTU_{I,cf} = (1-R)^{-1}\ln[(1-PR)/(1-P)] = 0.8727$ $\qquad F = NTU_{I,cf}/NTU_I = 0.9708$
$LMTD_{cf} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) = 28.67\text{°F}$ $\qquad F\,LMTD_{cf} = 27.83\text{°F} \equiv EMTD$

6
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