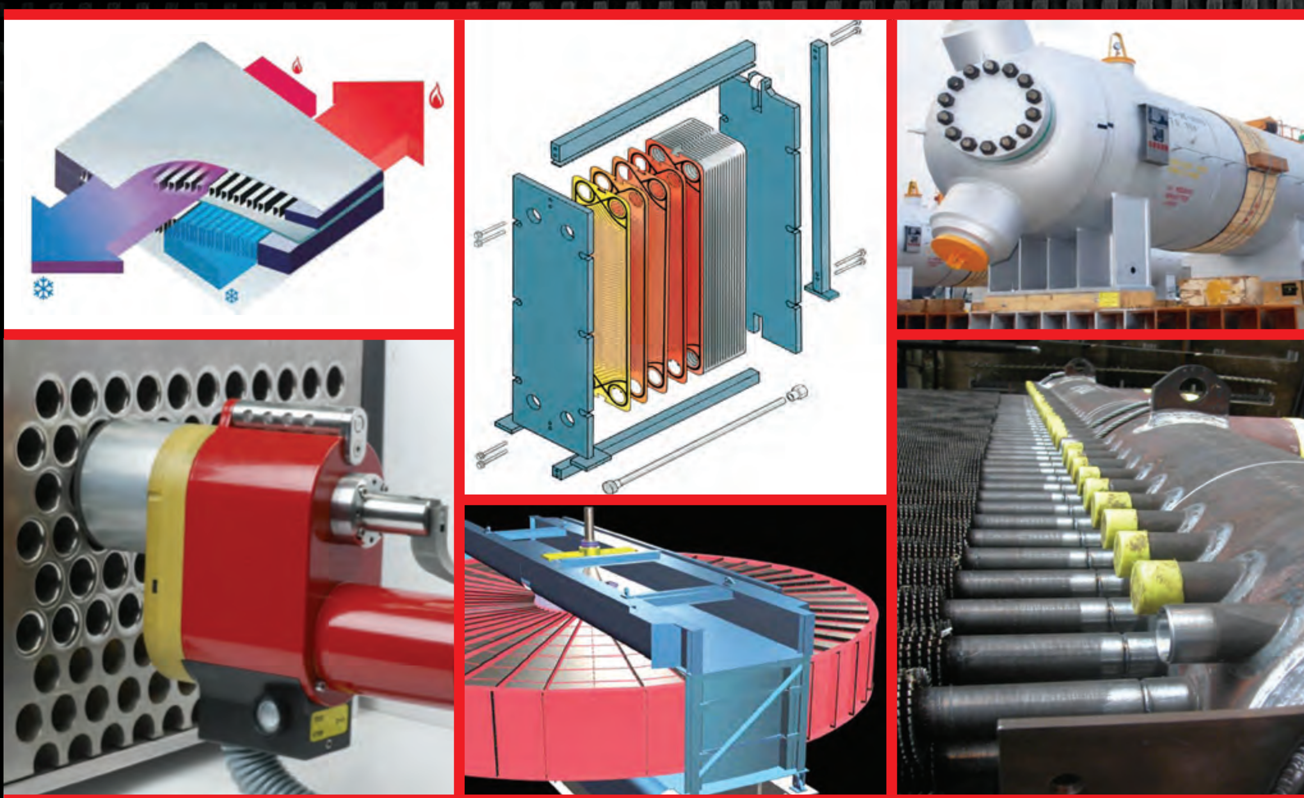
Heat Exchanger Design Handbook

SECOND EDITION



Kuppan Thulukkanam

CRC Press
Taylor & Francis Group

Heat Exchanger Design Handbook

SECOND EDITION

MECHANICAL ENGINEERING
A Series of Textbooks and Reference Books

Founding Editor

L. L. Faulkner
Columbus Division, Battelle Memorial Institute
and Department of Mechanical Engineering
The Ohio State University
Columbus, Ohio

RECENTLY PUBLISHED TITLES

Heat Exchanger Design Handbook, Second Edition,
Kuppan Thulukkanam

Vehicle Dynamics, Stability, and Control, Second Edition,
Dean Karnopp

HVAC Water Chillers and Cooling Towers: Fundamentals, Application, and Operation, Second Edition,
Herbert W. Stanford III

Ultrasonics: Fundamentals, Technologies, and Applications, Third Edition,
Dale Ensminger and Leonard J. Bond

Mechanical Tolerance Stackup and Analysis, Second Edition,
Bryan R. Fischer

Asset Management Excellence,
John D. Campbell, Andrew K. S. Jardine, and Joel McGlynn

Solid Fuels Combustion and Gasification: Modeling, Simulation, and Equipment Operations, Second Edition, Third Edition,
Marcio L. de Souza-Santos

Mechanical Vibration Analysis, Uncertainties, and Control, Third Edition,
Haym Benaroya and Mark L. Nagurka

Principles of Biomechanics,
Ronald L. Huston

Practical Stress Analysis in Engineering Design, Third Edition,
Ronald L. Huston and Harold Josephs

Practical Guide to the Packaging of Electronics, Second Edition: Thermal and Mechanical Design and Analysis,
Ali Jamnia

Friction Science and Technology: From Concepts to Applications, Second Edition,
Peter J. Blau

Design and Optimization of Thermal Systems, Second Edition,
Yogesh Jaluria

Heat Exchanger Design Handbook

SECOND EDITION

Kuppan Thulukkanam

CRC Press
Taylor & Francis Group
Boca Raton London New York

CRC Press is an imprint of the
Taylor & Francis Group, an **informa** business

CRC Press
Taylor & Francis Group
6000 Broken Sound Parkway NW, Suite 300
Boca Raton, FL 33487-2742

Version Date: 20130204

International Standard Book Number-13: 978-1-4398-4213-3 (eBook - PDF)

Visit the Taylor & Francis Web site at
http://www.taylorandfrancis.com

and the CRC Press Web site at
http://www.crcpress.com

Dedicated to

my parents, S. Thulukkanam and T. Senthamarai,

my wife, Tamizselvi Kuppan,

and my mentor, Dr. Ramesh K. Shah

Contents

Preface

INTRODUCTION

The advances in heat exchanger technology since the publication of the first edition and the topics that had been missed have necessitated a second edition of this book. This edition showcases recent advances in the selection, design, construction, operation, and maintenance of heat exchangers. The errors in the previous edition have been corrected, and the quality of figures including thermal effectiveness charts has been improved. This book provides up-to-date information on the single-phase heat transfer problems encountered by engineers in their daily work. It will continue to be a centerpiece of information for practicing engineers, research engineers, academicians, designers, and manufacturers involved in heat exchange between two or more fluids. Permission was sought from leading heat exchanger manufacturers and research organizations to include figures of practical importance, and these have been added in this edition. Care has been taken to minimize errors.

COVERAGE

In the chapter on the classification of heat exchangers, topics such as scrapped surface heat exchanger, graphite heat exchanger, coil wound heat exchanger, microscale heat exchanger, and printed circuit heat exchanger have been included. The construction and performance features of various types of heat exchangers have been compared.

Concepts like ALEX core for PFHE, radial flow heat exchanger for waste heat recovery, and rotary regenerator for HVAC applications have been added. Breach-Lock™ and Taper-Lok™ end closures have also been included.

Construction details and performance features of nonsegmental baffles heat exchangers such as EMbaffle®, Helixchanger®, and Twisted Tube® heat exchangers have been added. Design features of feedwater heater, steam surface condenser, and tantalum heat exchanger for pharmaceutical applications have also been included.

Information on pressure vessel codes, manufacturer's association standards, and ASME codes has been updated. ALPEMA standards for PFHE have been dealt with in depth.

Performance features of coil wound heat exchangers have been compared with brazed aluminum heat exchangers. The construction, selection, design, and concepts of manufacture of ACHE have been updated.

Recent advances in PHE concepts such as all welded, shell type, wide gap, free flow, semi-welded, and double-wall have been discussed and their construction and performance features compared.

The chapter on heat transfer augmentation has been thoroughly revised. Underlying the principle of heat transfer enhancement, devices such as hiTRAN thermal system and wire matrix turbulators have been described.

Fouling control concepts, such as back flushing, heat exchangers, such as self-cleaning, and liquid fluidized bed technology, fluidized bed units, and fouling control devices, such as Spirelf®, Fixotal®, and Turbotal®, have been added.

A new chapter on heat exchanger installation, operation, and maintenance covers the commissioning of new units, operation, their maintenance, repair practices, tube bundle removal, handling and cleaning, leak testing and plugging of tubes, condition monitoring, quality audit, and residual life assessment by NDT methods.

The tubesheet design procedure as per the latest ASME code, CODAP, PD 5000, and UPV has been discussed and compared with TEMA standards. The software program structure for design of ACHE and STHE has been updated. Recent trends in NDT methods such as ECT, UT, and leak testing have been included.

The chapter on fabrication of heat exchangers has now been revised, covering the recent advances in tube expansion and tube-to-tubesheet welding practices, rolling equipment, accessories, adequacy of rolling, cold working principle, configuration of tube-to-tubesheet joints for welding, modern equipment for tube hole preparation, and internal bore welding tubes. The section on heat exchanger heads has now been updated by incorporating various hot/cold working methods, and manufacturing procedures and PWHT have been discussed. New topics like CAB brazing of compact heat exchanger, cupro-braze radiators, and flat tube versus round tube concept for radiator tubings have also been added.

Due to their content and coverage, chapters 2, 5, 13 and 15 can be treated as individually self-contained units, as they do not require other chapters to be understood. This edition is abundantly illustrated with over 600 drawings, diagrams, photos, and tables. The *Heat Exchanger Design Handbook*, Second Edition is an excellent resource for mechanical, chemical, and petrochemical engineers; process equipment and pressure vessel designers, consultants, and heat exchanger manufacturers; and upper-level undergraduate and graduate students in these disciplines.

Acknowledgments

A large number of my colleagues from Indian Railways, well-wishers, and family members had contributed immensely toward the preparation of the book. Though I could not acknowledge them individually in the first edition, I mention a few of them here, as follows: Jothimani Gunasekaran, V. R. Ventakaraman, P. Subramani, Rajasri Anandan, and P. Gajapathy of ICF; Amitab Chakraborty (ADG), O. P. Agarwal (ED), M. Vijayakumar (Director), R. S. Madhukar, Suresh Prakash, Ashok Krishan, P. A. Rehman, and V. P. Gupta of RDSO, Lucknow; S. Bangaru Lakshman (then MOSR) and S. Dasarathy (then member mechanical, Railway Board, Ministry of Railways); Lilly Ravi, Renuka Devi Balasubramanian, Sukanya Balakrishnan, Sardha Balasubramaniam, and T. Adikesavan of Southern Railway; Amarnath, GM (retd.), and Radhey Shyam, GM, CLW; Cittaranjan, Ministry of Railways; Shanka Sinha of CLW; K. Kamaraj of IIT-Madras Library; M. Krishnasamy, Member of Parliament; my sisters, Anjala Manoharan, Maya Pannerselvam, and Indira Mari; my family members, Dr. K. Kalai (alias Vasanth), Dr. K. Kumudhini, and Er. K. Praveen; and Arunkumar Aranganathan, project manager, and Anithajohny Mariasusai, assistant director at SPi Global, Puducherry, India, who oversaw the production of the second edition on behalf of Taylor & Francis Group. I have immensely benefited from the contributions of scholars such as Dr. K. P. Singh, Dr. J. P. Gupta, and Dr. Ramesh K. Shah, the Donald Q. Kern awardee for the year 2005 by the American Society of Mechanical Engineers and the American Institute of Chemical Engineers. I also acknowledge the computer facilities of the Engine Development Directorate of RDSO, Lucknow, Ministry of Railways, and the library facilities of IIT-M, IIT-K, IIT-D, and RDSO, Lucknow. A large number of heat exchanger manufacturers and research organizations had spared photos and figures, and their names are acknowledged in the respective figure captions.

Author

Thulukkanam Kuppan, Indian Railway Service of Mechanical Engineers (IRSME), Ministry of Railways, is the chief mechanical engineer (senior administrative grade) in a rolling stock production unit. He has authored an article in the *ASME Journal of Pressure Vessel Technology*. His various roles have included being an experienced administrator, staff recruitment board chairman for a zonal railway, and joint director of the Engine Development Directorate of RDSO, Lucknow. He was also involved in design and performance evaluation of various types of heat exchangers used in diesel electric locomotives and has served as chief workshop engineer for the production of rolling stocks and as Director of Public Grievances (DPG) to the Minister of State for Railways, Railway Board, Government of India. Kuppan received his BE (hons) in 1980 from the PSG College of Technology, Coimbatore, Madras University, and his MTech in production engineering in 1982 from the Indian Institute of Technology, Madras, India.

1 Heat Exchangers
Introduction, Classification, and Selection

1.1 INTRODUCTION

A heat exchanger is a heat transfer device that is used for transfer of internal thermal energy between two or more fluids available at different temperatures. In most heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix. Heat exchangers are used in the process, power, petroleum, transportation, air-conditioning, refrigeration, cryogenic, heat recovery, alternate fuels, and other industries. Common examples of heat exchangers familiar to us in day-to-day use are automobile radiators, condensers, evaporators, air preheaters, and oil coolers. Heat exchangers can be classified into many different ways.

1.2 CONSTRUCTION OF HEAT EXCHANGERS

A heat exchanger consists of heat-exchanging elements such as a core or matrix containing the heat transfer surface, and fluid distribution elements such as headers or tanks, inlet and outlet nozzles or pipes, etc. Usually, there are no moving parts in the heat exchanger; however, there are exceptions, such as a rotary regenerator in which the matrix is driven to rotate at some design speed and a scraped surface heat exchanger in which a rotary element with scraper blades continuously rotates inside the heat transfer tube. The heat transfer surface is in direct contact with fluids through which heat is transferred by conduction. The portion of the surface that separates the fluids is referred to as the primary or direct contact surface. To increase heat transfer area, secondary surfaces known as fins may be attached to the primary surface. Figure 1.1 shows a collection of few types of heat exchangers.

1.3 CLASSIFICATION OF HEAT EXCHANGERS

In general, industrial heat exchangers have been classified according to (1) construction, (2) transfer processes, (3) degrees of surface compactness, (4) flow arrangements, (5) pass arrangements, (6) phase of the process fluids, and (7) heat transfer mechanisms. These classifications are briefly discussed here. For more details on heat exchanger classification and construction, refer to Shah [1,2], Gupta [3], and Graham Walker [4]. For classification and systematic procedure for selection of heat exchangers, refer to Larowski et al. [5a,5b]. Table 1.1 shows some types of heat exchangers, their construction details, and performance parameters.

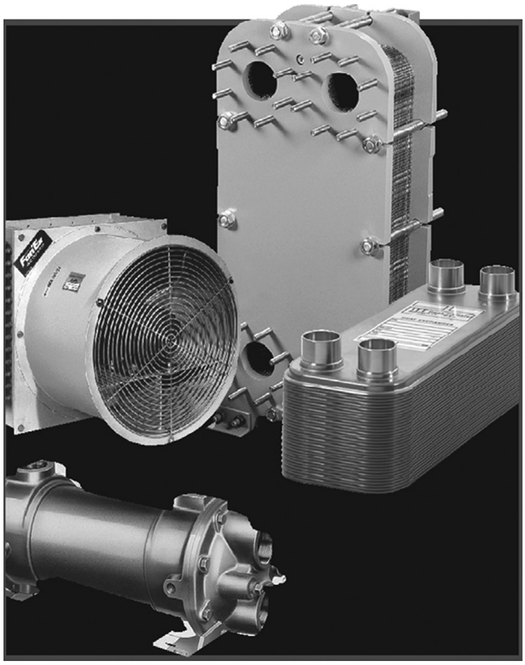


FIGURE 1.1 Collection of few types of heat exchangers. (Courtesy of ITT STANDARD, Cheektowaga, NY.)

1.3.1 Classification According to Construction

According to constructional details, heat exchangers are classified as [1] follows:

Tubular heat exchangers—double pipe, shell and tube, coiled tube
Plate heat exchangers (PHEs)—gasketed, brazed, welded, spiral, panel coil, lamella
Extended surface heat exchangers—tube-fin, plate-fin
Regenerators—fixed matrix, rotary matrix

1.3.1.1 Tubular Heat Exchanger

1.3.1.1.1 Double-Pipe Exchangers

A double-pipe heat exchanger has two concentric pipes, usually in the form of a U-bend design. Double-pipe heat changers with U-bend design are known as hairpin heat exchangers. The flow arrangement is pure countercurrent. A number of double-pipe heat exchangers can be connected in series or parallel as necessary. Their usual application is for small duties requiring, typically, less than 300 ft^2 and they are suitable for high pressures and temperatures and thermally long duties [5]. This has the advantage of flexibility since units can be added or removed as required, and the design is easy to service and requires low inventory of spares because of its standardization. Either longitudinal fins or circumferential fins within the annulus on the inner pipe wall are required to enhance the heat transfer from the inner pipe fluid to the annulus fluid. Design pressures and temperatures are broadly similar to shell and tube heat exchangers (STHEs). The design is straightforward and is carried out using the method of Kern [6] or proprietary programs. The Koch Heat Transfer Company LP, USA, is the pioneer in the design of hairpin heat exchangers. Figures 1.2 through 1.4 show double-pipe heat exchangers.

1.3.1.1.1.1 Application When the process calls for a temperature cross (when the hot fluid outlet temperature is below the cold fluid outlet temperature), a hairpin heat exchanger is the most efficient design and will result in fewer sections and less surface area. Also, they are commonly used for high-fouling services such as slurries and for smaller heat duties. Multitube heat

TABLE 1.1
Heat Exchanger Types: Construction and Performance Features

Type of Heat Exchanger	Constructional Features	Performance Features
Double pipe(hair pin) heat exchanger	A double pipe heat exchanger has two concentric pipes, usually in the form of a U-bend design. U-bend design is known as hairpin heat exchangers. The flow arrangement is pure countercurrent. The surface area ranges from 300 to 6000 ft^2 (finned tubes). Pressure capabilities are full vacuum to over 14,000 psi (limited by size, material, and design condition) and temperature from −100°C to 600°C (−150°F to 1100°F).	Applicable services: The process results in a temperature cross, high-pressure stream on tubeside, a low allowable pressure drop is required on one side, when the exchanger is subject to thermal shocks, when flow-induced vibration may be a problem.
Shell and tube heat exchanger (STHE)	The most commonly used heat exchanger. It is the “workhorse” of industrial process heat transfer. They are used as oil cooler, surface condenser, feed water heater, etc. The major components of a shell and tube exchanger are tubes, baffles, shell, front head, rear head, and nozzles. Shell diameter: 60 up to 2000 mm. Operating temperature: −20°C up to 500°C. Operating pressure max. 600 bar.	Advantages: Extremely flexible and robust design, easy to maintain and repair. Disadvantages 1. Require large site (footprint) area for installation and often need extra space to remove the bundle. 2. Construction is heavy. 3. PHE may be cheaper for pressure below 16 bar (230 psi) and temperature below 200°C (392°F).
Coiled tube heat exchanger (CTHE)	Construction of these heat exchangers involves winding a large number of small-bore ductile tubes in helix fashion around a central core tube, with each exchanger containing many layers of tubes along both the principal and radial axes. Different fluids may be passed in counterflow to the single shellside fluid.	Advantages, especially when dealing with low-temperature applications where simultaneous heat transfer between more than two streams is desired. Because of small bore tubes on both sides, CTHEs do not permit mechanical cleaning and therefore are used to handle clean, solid-free fluids or fluids whose fouling deposits can be cleaned by chemicals. Materials are usually aluminum alloys for cryogenics, and stainless steels for high-temperature applications.
Finned-tube heat exchanger	Construction 1. Normal fins on individual tubes referred to as individually finned tubes. 2. Longitudinal fins on individual tubes, which are generally used in condensing applications and for viscous fluids in double-pipe heat exchangers. 3. Flat or continuous (plain, wavy, or interrupted) external fins on an array of tubes (either circular or flat tube). 4. The tube layout pattern is mostly staggered.	Merits: small inventory, low weight, easier transport, less foundation, better temperature control Applications Condensers and evaporators of air conditioners, radiators for internal combustion engines, charge air coolers and intercoolers for cooling supercharged engine intake air of diesel engines, etc.

(*continued*)

TABLE 1.1 (continued)
Heat Exchanger Types: Construction and Performance Features

Type of Heat Exchanger	Constructional Features	Performance Features
Air cooled heat exchanger (ACHE)	Construction 1. Individually finned tube bundle. The tube bundle consists of a series of finned tubes set between side frames, passing between header boxes at either end. 2. An air-pumping device (such as an axial flow fan or blower) across the tube bundle which may be either forced draft or induced draft. 3. A support structure high enough to allow air to enter beneath the ACHE.	Merits: Design of ACHE is simpler compared to STHE, since the airside pressure and temperature pertain to ambient conditions. Tubeside design is same as STHE. Maintenance cost is normally less than that for water-cooled systems. The fouling on the air side can be cleaned easily. Disadvantages of ACHEs ACHEs require large heat transfer surfaces because of the low heat transfer coefficient on the air side and the low specific heat of air. Noise is a factor with ACHEs.
Plate-fin heat exchanger (PFHE)	Plate fin heat exchangers (PFHEs) are a form of compact heat exchanger consisting of a stack of alternate flat plates called “parting sheets” and fin corrugations, brazed together as a block. Different fins (such as the plain triangular, louver, perforated, or wavy fin) can be used between plates for different applications. Plate-fin surfaces are commonly used in gas-to-gas exchanger applications. They offer high area densities (up to about 6000 m^2/m^3 or 1800 ft^2/ft^3). Designed for low-pressure applications, with operating pressures limited to about 1000 kPa g (150 psig) and operating temperature from cryogenic to 150°C (all-aluminum PFHE) and about 700°C–800°C (1300°F–1500°F) (made of heat-resistant alloys).	1. PFHE offers superior in thermal performance compared to extended surface heat exchangers. 2. PFHE can achieve temperature approaches as low as 1°C between single-phase streams and 3°C between multiphase streams. 3. With their high surface compactness, ability to handle multiple streams, and with aluminum’s highly desirable low-temperature properties, brazed aluminum plate fins are an obvious choice for cryogenic applications. 4. Very high thermal effectiveness can be achieved; for cryogenic applications, effectiveness of the order of 95% and above is common. Limitations: 1. Narrow passages in plate-fin exchangers make them susceptible for fouling and they cannot be cleaned by mechanical means. This limits their use to clean applications like handling air, light hydrocarbons, and refrigerants.
Regenerator	The heat exchanger used to preheat combustion air is called either a recuperator or a regenerator. A recuperator is a convective heat transfer type heat exchanger like tubular, plate-fin and extended surface heat exchangers. The regenerator is classified as (1) fixed matrix or fixed bed and (2) rotary regenerators. The matrix is alternatively heated by hot fluid and cooled by the cold fluid. Features: 1. A more compact size (β = 8800 m^2/m^3 for rotating type and 1600 m^2/m^3 for fixed matrix type). 2. Application to both high temperatures (800°C–1100°C) for metal matrix, and 2000°C for ceramic regenerators for services like gas turbine applications, melting furnaces or steam power plant heat recovery, and low-temperature applications like space heating (HVAC).	Usage 1. Reheating process feedstock. 2. Waste heat boiler and feed water heating for generating steam (low-temperature recovery system). 3. Air preheater—preheating the combustion air (high temperature heat recovery system). 4. Space heating—rotary heat exchanger (wheel) is mainly used in building ventilation or in the air supply/discharge system of air conditioning equipment.

TABLE 1.1 (continued)
Heat Exchanger Types: Construction and Performance Features

Type of Heat Exchanger	Constructional Features	Performance Features
	3. Operating pressure of 5–7 bar for gas turbine applications and low pressure of 1–1.5 bar for air dehumidifier and waste heat recovery applications. 4. The absence of a separate flow path like tubes or plate walls but the presence of seals to separate the gas stream in order to avoid mixing due to pressure differential.	
Plate heat exchanger (PHE)	A plate heat exchanger is usually comprised of a stack of corrugated or embossed metal plates in mutual contact, each plate having four apertures serving as inlet and outlet ports, and seals designed so as to direct the fluids in alternate flow passages. Standard performance limits Maximum operating pressure 25 bar (360 psi) Maximum temperature 160°C (320°F) With special gaskets 200°C (390°F) Maximum flow rate 3600 m^3/h (950,000 USG/min) Temperature approach As low as 1°C Heat recovery As high as 93% Heat transfer coefficient (water–water duties with normal fouling resistance) 3000–7000 W/$m^2 \cdot$°C Other varieties include, brazed plate heat exchanger (BPHE), shell and plate heat exchanger, welded plate heat exchanger, wide-gap plate heat exchanger, free-flow plate heat exchanger, semi-welded or twin-plate heat exchanger, double-wall plate heat exchanger, biabon F graphite plate heat exchanger, etc.	Merits: True counterflow, high turbulence and high heat transfer performance. Close approach temperature. Reduced fouling: Cross-contamination eliminated. Multiple duties with a single unit. Expandable. Easy to inspect and clean, and less maintenance. Low liquid volume and quick process control. Lower cost. Disadvantages 1. The maximum operating temperature and pressure are limited by gasket materials. The gaskets cannot handle corrosive or aggressive media. 2. Gasketed plate heat exchangers cannot handle particulates that are larger than 0.5 mm. 3. Gaskets always increase the leakage risk. Exchanger
Spiral plate heat exchanger (SPHE)	SPHE is fabricated by rolling a pair of relatively long strips of plate to form a pair of spiral passages. Channel spacing is maintained uniformly along the length of the spiral passages by means of spacer studs welded to the plate strips prior to rolling.	Advantages: To handle slurries and liquids with suspended fibers, and mineral ore treatment where the solid content is up to 50%. The SPHE is the first choice for extremely high viscosities, say up to 500,000 cp, especially in cooling duties. Applications: SPHEs are finding applications in reboiling, condensing, heating or cooling of viscous fluids, slurries, and sludge.
Printed circuit heat exchangers (PCHEs)	HEATRIC printed circuit heat exchangers consist of diffusion-bonded heat exchanger core that are constructed from flat metal plates into which fluid flow channels are either chemically etched or pressed. They can withstand pressure of 600 bar (9000 psi) with extreme temperatures, ranging from cryogenic to 700°C (1650°F).	Merits: fluid flow can be parallelflow, counterflow, crossflow, or a combination of these to suit the process requirements. Thermal effectiveness is of the order of 98% in a single unit. They can incorporate more than two process streams into a single unit.

(*continued*)

TABLE 1.1 (continued)
Heat Exchanger Types: Construction and Performance Features

Type of Heat Exchanger	Constructional Features	Performance Features
Lamella heat exchanger (LHE)	A lamella heat exchanger normally consists of a cylindrical shell surrounding a number of heat transferring lamellas. The design can be compared to a tube heat exchanger but with the circular tubes replaced by thin and wide channels, lamellas. The lamella heat exchanger works with the media in full counter current flow. The absence of baffle plates minimizes the pressure drop and makes handling of most media possible.	Merits: Since the lamella bundle can be easily dismantled from the shell, inspection and cleaning is easy. Applications Cooking fluid heating in pulp mills. Liquor preheaters. Coolers and condensers of flue gas. Oil coolers.
Heat pipe heat exchanger	The heat-pipe heat exchanger used for gas–gas heat recovery is essentially bundle of finned tubes assembled like a conventional air-cooled heat exchanger. The heat pipe consists of three elements: (1) a working fluid inside the tubes, (2) a wick lining inside the wall, and (3) vacuum sealed finned tube. The heat-pipe heat exchanger consists of an evaporative section through which the hot exhaust gas flows and a condensation section through which the cold air flows. These two sections are separated by a separating wall.	Application: The heat pipes are used for (i) heat recovery from process fluid to preheating of air for space heating, (ii) HVAC application-waste heat recovery from the exhaust air to heat the incoming process air It virtually does not need mechanical maintenance, as there are no moving parts. The heat pipe heat recovery systems are capable of operating at a temperature of 300°C–315°C with 60%–80% heat recovery capability.
Plate coil heat exchanger (PCHE)	Fabricated from two metal sheets, one or both of which are embossed. When welded together, the embossings form a series of well-defined passages through which the heat transfer media flows.	A variety of standard PLATECOIL® fabrications, such as pipe coil, half pipe, jacketed tanks and vessels, clamp-on upgrades, immersion heaters and coolers, heat recovery banks, storage tank heaters, etc., are available. Easy access to panels and robust cleaning surfaces reduce maintenance burdens.
Scraped surface heat exchanger	Scraped surface heat exchangers are essentially double pipe construction with the process fluid in the inner pipe and the cooling (water) or heating medium (steam) in the annulus. A rotating element is contained within the tube and is equipped with spring-loaded blades. In operation the rotating shaft scraper blades continuously scrape product film from the heat transfer tube wall, thereby enhancing heat transfer and agitating the product to produce a homogenous mixture.	Scraped surface heat exchangers are used for processes likely to result in the substantial deposition of suspended solids on the heat transfer surface. Scraped surface heat exchangers can be employed in the continuous, closed processing of virtually any pumpable fluid or slurry involving cooking, slush freezing, cooling, crystallizing, mixing, plasticizing, gelling, polymerizing, heating, aseptic processing, etc. Use of a scraped surface exchanger prevents the accumulation of significant buildup of solid deposits.

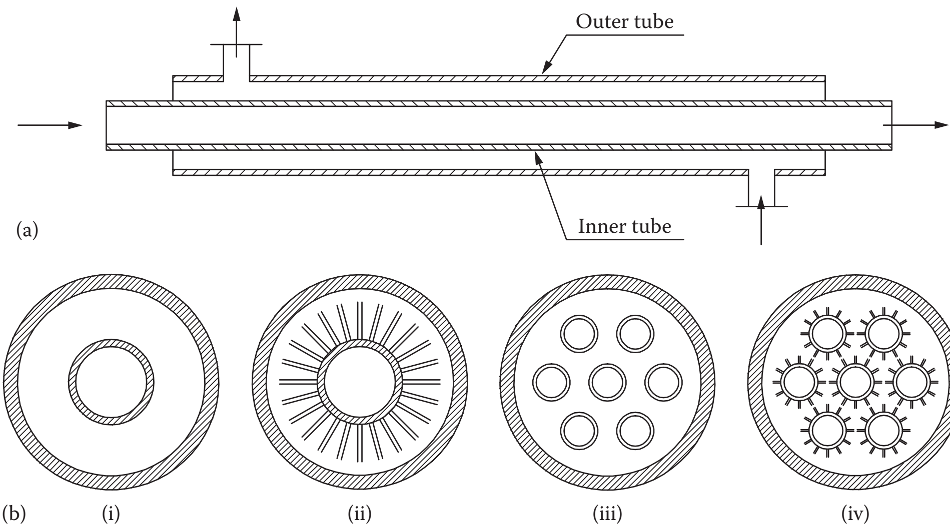


FIGURE 1.2 Double pipe/twin pipe hairpin heat exchanger. (a) Schematic of the unit, (b): (i) double pipe with bare internal tube, (ii) double pipe with finned internal tube, (iii) double pipe with multibare internal tubes, and (iv) double pipe with multifinned internal tubes. (Courtesy of Peerless Mfg. Co., Dallas, TX, Makers of Alco and Bos-Hatten brands of heat exchangers.)

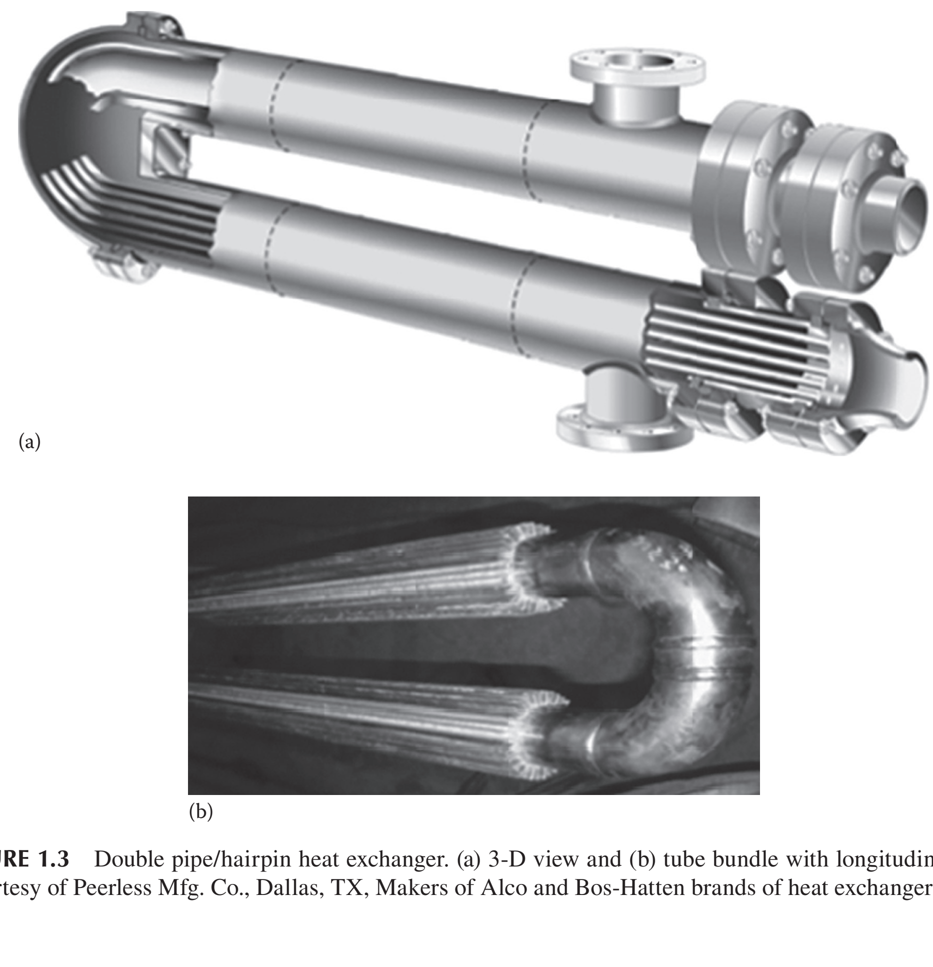
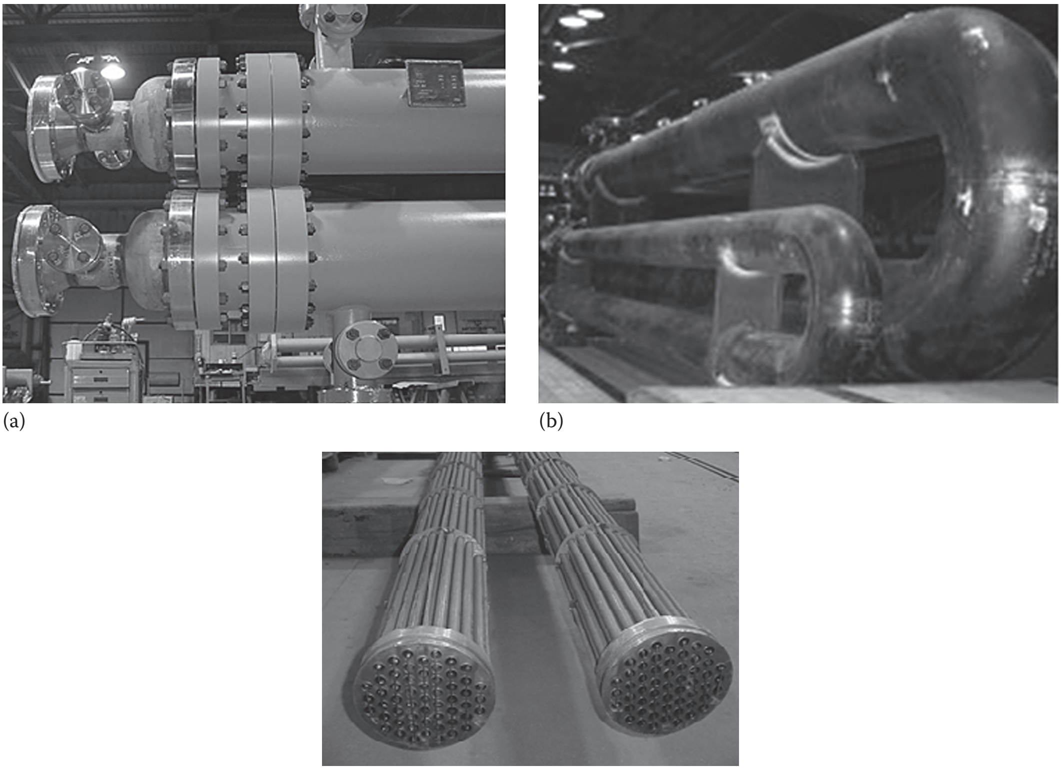


FIGURE 1.3 Double pipe/hairpin heat exchanger. (a) 3-D view and (b) tube bundle with longitudinal fins. (Courtesy of Peerless Mfg. Co., Dallas, TX, Makers of Alco and Bos-Hatten brands of heat exchangers.)



(a) (b) (c)

FIGURE 1.4 Hairpin heat exchanger. (a) Separated head closure using separate bolting on shellside and tubeside and (b) Hairpin exchangers for high-pressure and high-temperature applications and (c) multitubes (bare) bundle. (Photo courtesy of Heat Exchanger Design, Inc., Indianapolis, IN.)

exchangers are used for larger heat duties. A hairpin heat exchanger should be considered when one or more of the following conditions exist:

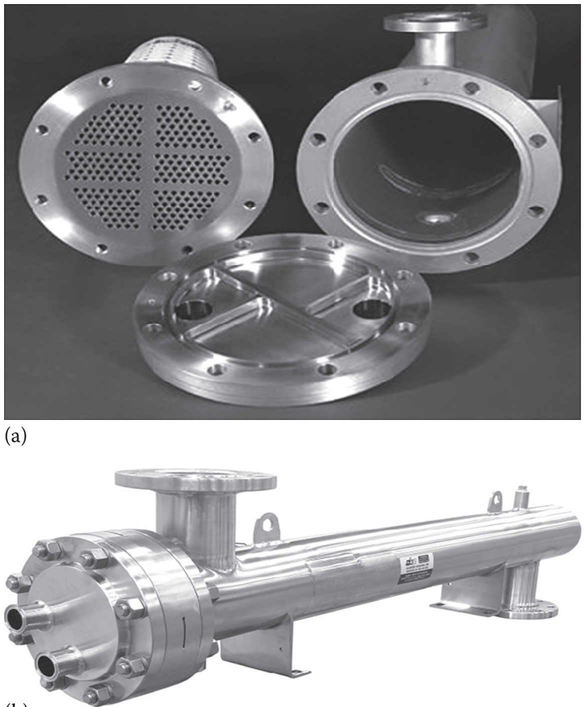
- The process results in a temperature cross
- High pressure on tubeside application
- A low allowable pressure drop is required on one side
- When an augmentation device to enhance the heat transfer coefficient is desired
- When the exchanger is subject to thermal shocks
- When flow-induced vibration may be a problem
- When solid particulates or slurries are present in the process stream

1.3.1.1.2 Shell and Tube Heat Exchanger

In process industries, shell and tube heat exchangers are used in great numbers, far more than any other type of exchanger. More than 90% of heat exchangers used in industry are of the shell and tube type [7]. STHEs are the "workhorses" of industrial process heat transfer [8]. They are the first choice because of well-established procedures for design and manufacture from a wide variety of materials, many years of satisfactory service, and availability of codes and standards for design and fabrication. They are produced in the widest variety of sizes and styles. There is virtually no limit on the operating temperature and pressure. Figure 1.5 shows STHEs.

1.3.1.1.3 Coiled Tube Heat Exchanger

Construction of these heat exchangers involves winding a large number of small-bore ductile tubes in helix fashion around a central core tube, with each exchanger containing many layers of tubes



(a)

(b)

FIGURE 1.5 Shell and tube heat exchanger. (a) Components and (b) heat exchanger. (Courtesy of Allegheny Bradford Corporation, Bradford, PA.)

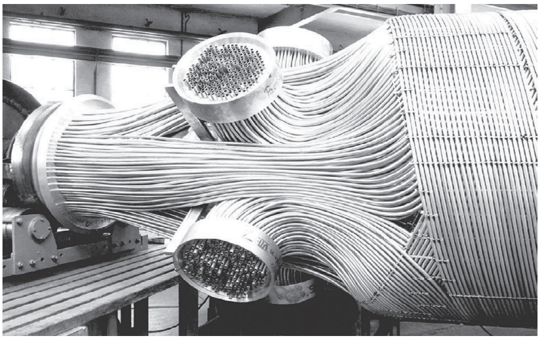
along both the principal and radial axes. The tubes in individual layers or groups of layers may be brought together into one or more tube plates through which different fluids may be passed in counterflow to the single shellside fluid. The construction details have been explained in Refs. [5,9]. The high-pressure stream flows through the small-diameter tubes, while the low-pressure return stream flows across outside of the small-diameter tubes in the annular space between the inner central core tube and the outer shell. Pressure drops in the coiled tubes are equalized for each high-pressure stream by using tubes of equal length and varying the spacing of these in the different layers. Because of small-bore tubes on both sides, CTHEs do not permit mechanical cleaning and therefore are used to handle clean, solid-free fluids or fluids whose fouling deposits can be cleaned by chemicals. The materials used are usually aluminum alloys for cryogenics and stainless steel for high-temperature applications.

CTHE offers unique advantages, especially when dealing with low-temperature applications for the following cases [9]:

- Simultaneous heat transfer between more than two streams is desired. One of the three classical heat exchangers used today for large-scale liquefaction systems is CTHE.
- A large number of heat transfer units are required.
- High-operating pressures are involved.

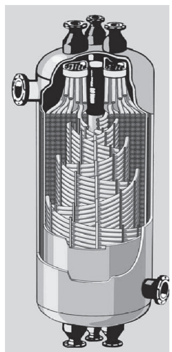
CTHE is not cheap because of the material costs, high labor input in winding the tubes, and the central mandrel, which is not useful for heat transfer but increases the shell diameter [5].

1.3.1.1.3.1 Linde Coil-Wound Heat Exchangers Linde coil-wound heat exchangers are compact and reliable with a broad temperature and pressure range and suitable for both single- and two-phase streams. Multiple streams can be accommodated in one exchanger. They are known for their



(a)

(b)



(c)

FIGURE 1.6 Coiled tube heat exchanger. (a) End section of a tube bundle, (b) tube bundle under fabrication, and (c) construction details. (From Linde AG, Engineering Division. With permission.)

robustness in particularly during start-up and shut-down or plant-trip conditions. Both the brazed aluminum PFHEs and CTHEs find application in liquefication processes. A comparison of salient features of these two types of heat exchangers is shown in Chapter 4. Figure 1.6 shows Linde coil-wound heat exchangers.

Glass coil heat exchangers: Two basic types of glass coil heat exchangers are (i) coil type and (ii) STHE with glass or MS shells in combination with glass tube as standard material for tube. Glass coil exchangers have a coil fused to the shell to make a one-piece unit. This prohibits leakage between the coil and shellside fluids [10]. The reduced heat transfer coefficient of boro silicate glass equipment compares favorably with many alternate tube materials. This is due to the smooth surface of the glass that improves the film coefficient and reduces the tendency for fouling. More details on glass heat exchangers are furnished in Chapter 13.

1.3.1.2 Plate Heat Exchangers

PHEs are less widely used than tubular heat exchangers but offer certain important advantages. PHEs can be classified into three principal groups:

1. Plate and frame or gasketed PHEs used as an alternative to tube and shell exchangers for low- and medium-pressure liquid–liquid heat transfer applications
2. Spiral heat exchanger used as an alternative to shell and tube exchangers where low maintenance is required, particularly with fluids tending to sludge or containing slurries or solids in suspension
3. Panel heat exchangers made from embossed plates to form a conduit or coil for liquids coupled with fins

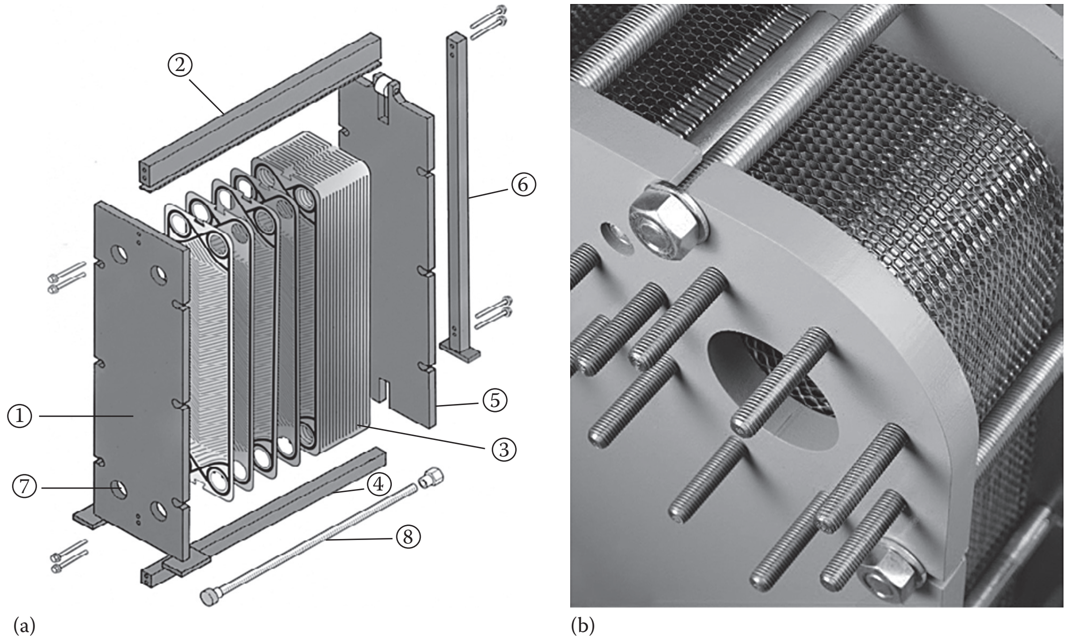


FIGURE 1.7 Plate heat exchanger. (a) Construction details—schematic (Parts details: 1, Fixed frame plate; 2, Top carrying bar; 3, Plate pack; 4, Bottom carrying bar; 5, Movable pressure plate; 6, Support column; 7, Fluids port; and 8, Tightening bolts.) and (b) closer view of assembled plates. (Courtesy of ITT STANDARD, Cheektowaga, NY.)

1.3.1.2.1 Plate and Frame or Gasketed Plate Heat Exchangers

A PHE essentially consists of a number of corrugated metal plates in mutual contact, each plate having four apertures serving as inlet and outlet ports, and seals designed to direct the fluids in alternate flow passages. The plates are clamped together in a frame that includes connections for the fluids. Since each plate is generally provided with peripheral gaskets to provide sealing arrangements, PHEs are called gasketed PHEs. PHEs are shown in Figure 1.7 and are covered in detail in Chapter 7.

1.3.1.2.2 Spiral Plate Heat Exchanger

SPHEs have been used since the 1930s, when they were originally developed in Sweden for heat recovery in pulp mills. They are classified as a type of welded PHE. An SPHE is fabricated by rolling a pair of relatively long strips of plate around a split mandrel to form a pair of spiral passages. Channel spacing is maintained uniformly along the length of the spiral passages by means of spacer studs welded to the plate strips prior to rolling. Figure 1.8 shows an SPHE. For most applications, both flow channels are closed by alternate channels welded at both sides of the spiral plate. In some services, one of the channels is left open, whereas the other closed at both sides of the plate. These two types of construction prevent the fluids from mixing.

The SPHE is intended especially for the following applications [5]:

- To handle slurries and liquids with suspended fibers and mineral ore treatment where the solid content is up to 50%.
- SPHE is the first choice for extremely high viscosities, say up to 500,000 cp, especially in cooling duties, because of maldistribution, and hence partial blockage by local overcooling is less likely to occur in a single-channel exchanger.
- SPHEs are finding applications in reboiling, condensing, heating, or cooling of viscous fluids, slurries, and sludge [11].

More details on SPHE are furnished in Chapter 7.

FIGURE 1.8 Spiral plate heat exchanger. (Courtesy of Tranter, Inc., Wichita Falls, TX.)

1.3.1.2.3 Plate or Panel Coil Heat Exchanger

These exchangers are called panel coils, plate coils, or embossed panel or jacketing. The panel coil serves as a heat sink or a heat source, depending upon whether the fluid within the coil is being cooled or heated. Panel coil heat exchangers are relatively inexpensive and can be made into any desired shape and thickness for heat sinks and heat sources under varied operating conditions. Hence, they have been used in many industrial applications such as cryogenics, chemicals, fibers, food, paints, pharmaceuticals, and solar absorbers.

Construction details of a panel coil: A few types of panel coil designs are shown in Figure 1.9. The panel coil is used in such industries as plating, metal finishing, chemical, textile, brewery, pharmaceutical, dairy, pulp and paper, food, nuclear, beverage, waste treatment, and many others. Construction details of panel coils are discussed next. M/s Paul Muller Company, Springfield, MO, and Tranter, Inc., TX, are the leading manufacturers of panel coil/plate coil heat exchangers.

Single embossed surface: The single embossed heat transfer surface is an economical type to utilize for interior tank walls, conveyor beds, and when a flat side is required. The single embossed design uses two sheets of material of different thickness and is available in stainless steel, other alloys, carbon steel, and in many material gages and working pressures.

Double embossed surface: Inflated on both sides using two sheets of material and the same thickness, the double embossed construction maximizes the heating and cooling process by utilizing both sides of the heat transfer plate. The double embossed design is commonly used in immersion applications and is available in stainless steel, other alloys, carbon steel, and in many material gages and working pressures.

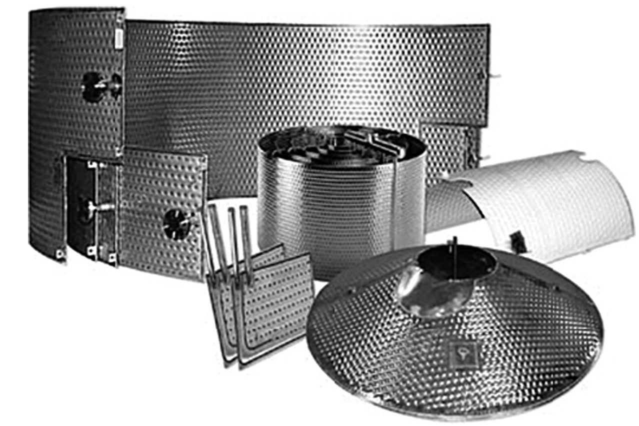


FIGURE 1.9 Temp-Plate® heat transfer surface. (Courtesy of Mueller, Heat Transfer Products, Springfield, MO.)

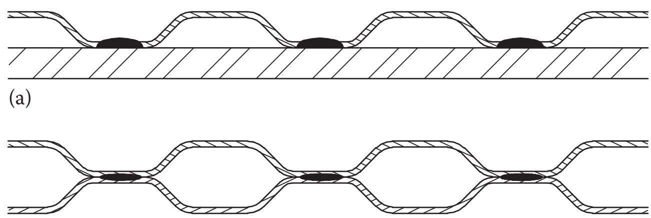


FIGURE 1.10 Welded dimpled jacket template. (a) Gas metal arc welded and (b) resistance welded.

Dimpled surface: This surface is machine punched and swaged, prior to welding, to increase the flow area in the passages. It is available in stainless steel, other alloys, carbon steel, in many material gages and working pressures, and in both MIG plug-welded and resistance spot-welded forms.

Methods of manufacture of panel coils: Basically, three different methods have been used to manufacture the panel coils: (1) they are usually welded by the resistance spot-welding or seam-welding process. An alternate method now available offers the ability to resistance spot-weld the dimpled jacket-style panel coil with a perimeter weldment made with the GMAW or resistance welding. Figure 1.10 shows a vessel jacket welded by GMAW and resistance-welding process. Other methods are (2) the die-stamping process and (3) the roll-bond process. In the die-stamping process, flow channels are die-stamped on either one or two metal sheets. When one sheet is embossed and joined to a flat (unembossed sheet), it forms a single-sided embossed panel coil. When both sheets are stamped, it forms a double-sided embossed panel coil.

Types of jackets: Jacketing of process vessels is usually accomplished by using one of the three main available types: conventional jackets, dimple jackets, and half-pipe coil jackets [12].

Advantages of panel coils: Panel coils provide the optimum method of heating and cooling process vessels in terms of control, efficiency, and product quality. Using a panel as a means of heat transfer offers the following advantages [12]:

- All liquids can be handled, as well as steam and other high-temperature vapors.
- Circulation, temperature, and velocity of heat transfer media can be accurately controlled.
- Panels may often be fabricated from a much less expensive metal than the vessel itself.
- Contamination, cleaning, and maintenance problems are eliminated.

- Maximum efficiency, economy, and flexibility are achieved.
- In designing reactors for specific process, this variety gives chemical engineers a great deal of flexibility in the choice of heat transfer medium.

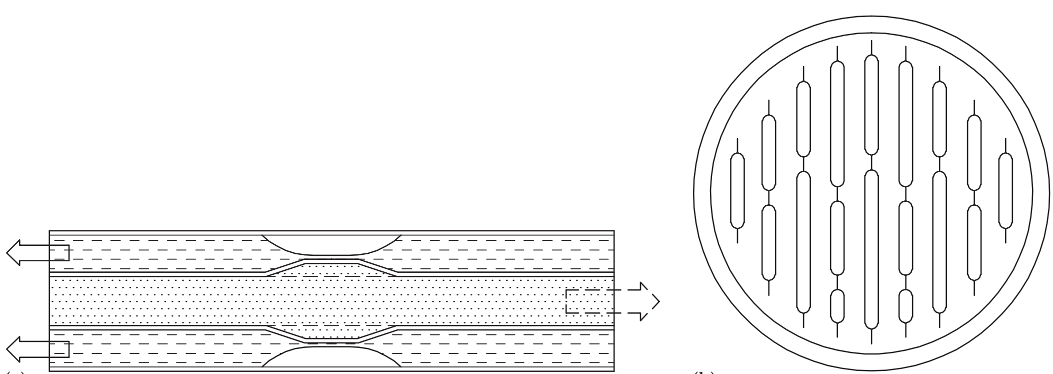
1.3.1.2.4 Lamella Heat Exchanger

The lamella heat exchanger is an efficient, and compact, heat exchanger. The principle was originally developed around 1930 by Ramens Patenter. Later Ramens Patenter was acquired by Rosenblads Patenter and the lamella heat exchanger was marketed under the Rosenblad name. In 1988, Berglunds acquired the product and continued to develop it. A lamella heat exchanger normally consists of a cylindrical shell surrounding a number of heat-transferring lamellas. The design can be compared to a tube heat exchanger but with the circular tubes replaced by thin and wide channels, lamellas. Sondex Tapiro Oy Ab Pikkupurontie 11, FIN-00810 Helsinki, Finland, markets lamella heat exchangers worldwide.

The lamella is a form of welded heat exchanger that combines the construction of a PHE with that of a shell and tube exchanger without baffles. In this design, tubes are replaced by pairs of thin flat parallel metal plates, which are edge welded to provide long narrow channels, and banks of these elements of varying width are packed together to form a circular bundle and fitted within a shell. The cross section of a lamella heat exchanger is shown schematically in Figure 1.11. With this design, the flow area on the shellside is a minimum and similar in magnitude to that of the inside of the bank of elements; due to this, the velocities of the two liquid media are comparable [13]. The flow is essentially longitudinal countercurrent “tubeside” flow of both tube and shell fluids [4]. Due to this, the velocities of the two liquid media are comparable. Also, the absence of baffles minimizes the pressure drop. One end of the element pack is fixed and the other is floating to allow for thermal expansion and contraction. The connections fitted at either end of the shell, as in the normal shell and tube design, allow the bank of elements to be withdrawn, making the outside surface accessible for inspection and cleaning. Opposed from an STHE, where the whole exchanger has to be replaced in case of damage, it is possible just to replace the lamella battery and preserve the existing shell. Lamella heat exchangers can be fabricated from carbon steel, stainless steel, titanium, Incolly, and Hastelloy. They can handle most fluids, with large volume ratios between fluids. The floating nature of the bundle usually limits the working pressure to 300 psi. Lamella heat exchangers are generally used only in special cases. Design is usually done by the vendors.

Merits of lamella heat exchanger are as follows:

1. Strong turbulence in the fluid
2. High operation pressure



(a) (b)

FIGURE 1.11 Lamella heat exchanger. (a) Counterflow concept and (b) lamella tube bundle.

FIGURE 1.12 Air-cooled condenser. (Courtesy of GEA Iberica S.A., Vizcaya, Spain.)

Applications

Cooking fluid heating in pulp mills
Liquor preheaters
Coolers and condensers of flue gas
Oils coolers

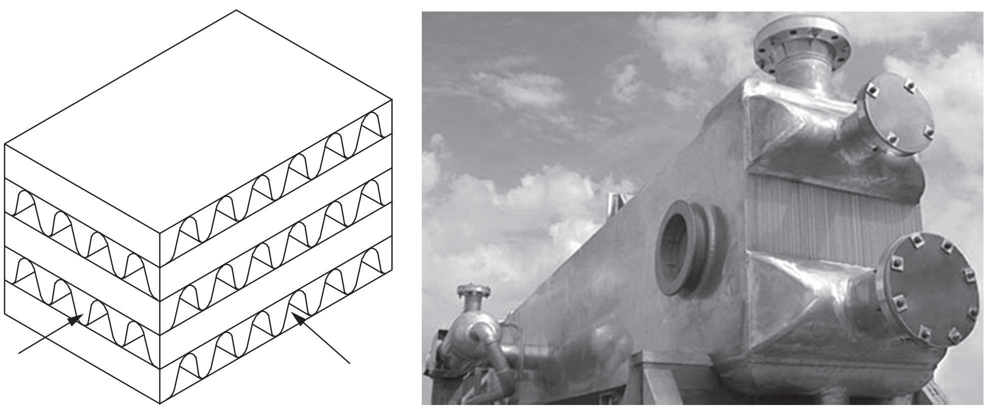
1.3.1.3 Extended Surface Exchangers

In a heat exchanger with gases or some liquids, if the heat transfer coefficient is quite low, a large heat transfer surface area is required to increase the heat transfer rate. This requirement is served by fins attached to the primary surface. Tube-fin heat exchangers (Figure 1.12) and plate-fin heat exchangers (Figure 1.13) are the most common examples of extended surface heat exchangers. Their design is covered in Chapter 4.

1.3.1.4 Regenerative Heat Exchangers

Regeneration is an old technology dating back to the first open hearths and blast furnace stoves. Manufacturing and process industries such as glass, cement, and primary and secondary metals account for a significant fraction of all energy consumed. Much of this energy is discarded in the form of high-temperature exhaust gas. Recovery of waste heat from the exhaust gas by means of heat exchangers known as regenerators can improve the overall plant efficiency [14].

Types of regenerators: Regenerators are generally classified as fixed-matrix and rotary regenerators. Further classifications of fixed and rotary regenerators are shown in Figure 1.14. In the former,



(a) (b)

FIGURE 1.13 Plate-fin heat exchanger. (a) Schematic of exchanger and (b) brazed aluminum plate-fin heat exchanger. (From Linde AG, Engineering Division. With permission.)

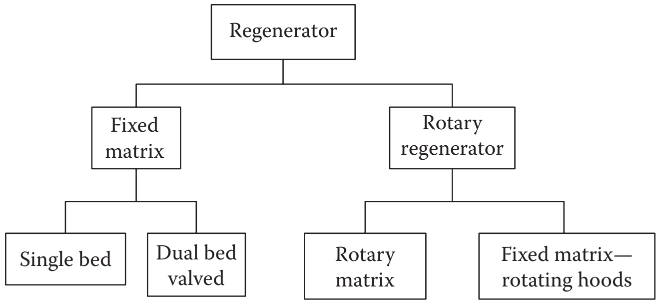


FIGURE 1.14 Classification of regenerators.

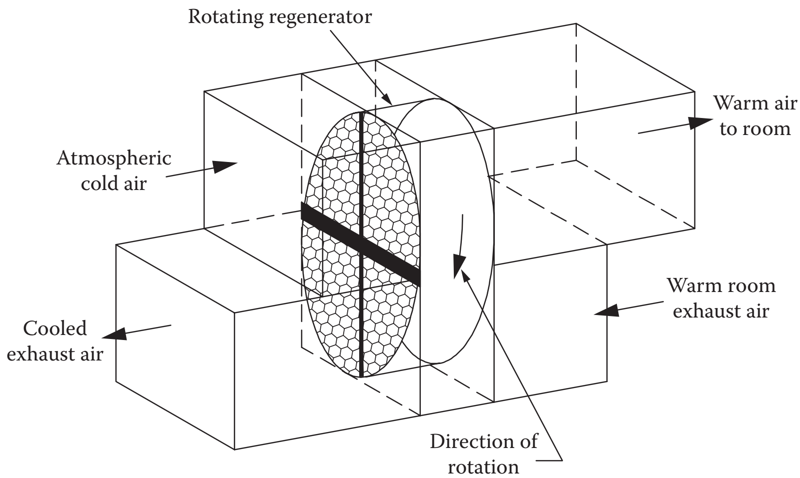


FIGURE 1.15 Rotary regenerator: working principle.

regeneration is achieved with periodic and alternate blowing of hot and the cold stream through a fixed matrix. During the hot flow period, the matrix receives thermal energy from the hot gas and transfers it to the cold stream during the cold stream flow. In the latter, the matrix revolves slowly with respect to two fluid streams. The rotary regenerator is commonly employed in gas turbine power plants where the waste heat in the hot exhaust gases is utilized for raising the temperature of compressed air before it is supplied to the combustion chamber. A rotary regenerator (rotary wheel for HVAC application) working principle is shown in Figure 1.15, and Figure 1.16 shows the Rothemuhle regenerative air preheater of Babcock and Wilcox Company. Rotary regenerators fall in the category of compact heat exchangers since the heat transfer surface area to regenerator volume ratio is very high. Regenerators are further discussed in detail in Chapter 6.

1.3.2 Classification according to Transfer Process

These classifications are as follows:

Indirect contact type—direct transfer type, storage type, fluidized bed
Direct contact type—cooling towers

1.3.2.1 Indirect Contact Heat Exchangers

In an indirect contact–type heat exchanger, the fluid streams remain separate and the heat transfer takes place continuously through a dividing impervious wall. This type of heat exchanger can be further classified into direct transfer type, storage type, and fluidized bed exchangers. Direct transfer type is dealt with next, whereas the storage type and the fluidized bed type are discussed in Chapter 6.

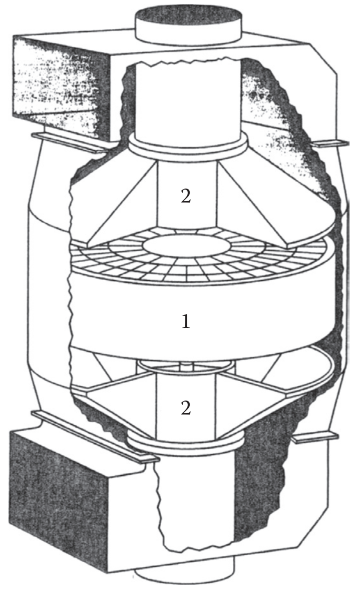


FIGURE 1.16 Rothemuhle regenerative air preheater of Babcock and Wilcox Company—stationary matrix (part 1) and revolving hoods (part 2). (Adapted from Mondt, J.R., Regenerative heat exchangers: The elements of their design, selection and use, Research Publication GMR-3396, General Motors Research Laboratories, Warren, MI, 1980.)

1.3.2.1.1 Direct Transfer–Type Exchangers

In this type, there is a continuous flow of heat from the hot fluid to the cold fluid through a separating wall. There is no direct mixing of the fluids because each fluid flows in separate fluid passages. There are no moving parts. This type of exchanger is designated as a recuperator. Some examples of direct transfer–type heat exchangers are tubular exchangers, PHEs, and extended surface exchangers. Recuperators are further subclassified as prime surface exchangers, which do not employ fins or extended surfaces on the prime surface. Plain tubular exchangers, shell and tube exchangers with plain tubes, and PHEs are examples of prime surface exchangers.

1.3.2.2 Direct Contact–Type Heat Exchangers

In direct contact–type heat exchangers, the two fluids are not separated by a wall and come into direct contact, exchange heat, and are then separated.

Owing to the absence of a wall, closer temperature approaches are attained. Very often, in the direct contact type, the process of heat transfer is also accompanied by mass transfer. Various types of direct contact heat exchangers include (a) immiscible fluid exchanger, (b) gas–liquid exchanger, and (c) liquid–vapor exchanger. The cooling towers and scrubbers are examples of a direct contact–type heat exchanger.

1.3.3 Classification according to Surface Compactness

Compact heat exchangers are important when there are restrictions on the size and weight of exchangers. A compact heat exchanger incorporates a heat transfer surface having a high area density, β, somewhat arbitrarily 700 m^2/m^3 (200 ft^2/ft^3) and higher [1]. The area density, β, is the ratio of heat transfer area A to its volume V. A compact heat exchanger employs a compact surface on one or more sides of a two-fluid or a multifluid heat exchanger. They can often achieve higher thermal effectiveness than shell and tube exchangers (95% vs. the 60%–80% typical for STHEs), which makes them particularly useful in energy-intensive industries [15]. For least capital cost, the size of the unit should be minimal. There are some additional advantages to small volume as follows:

Small inventory, making them good for handling expensive or hazardous materials [15]
Low weight
Easier transport
Less foundation
Better temperature control

Some barriers to the use of compact heat exchangers include [15] the following:

The lack of standards similar to pressure vessel codes and standards, although this is now being redressed in the areas of plate-fin exchangers [16] and air-cooled exchangers [17].
Narrow passages in plate-fin exchangers make them susceptible for fouling and they cannot be cleaned by mechanical means. This limits their use to clean applications like handling air, light hydrocarbons, and refrigerants.

1.3.4 Classification According to Flow Arrangement

The basic flow arrangements of the fluids in a heat exchanger are as follows:

Parallelflow
Counterflow
Crossflow

The choice of a particular flow arrangement is dependent upon the required exchanger effectiveness, fluid flow paths, packaging envelope, allowable thermal stresses, temperature levels, and other design criteria. These basic flow arrangements are discussed next.

1.3.4.1 Parallelflow Exchanger

In this type, both the fluid streams enter at the same end, flow parallel to each other in the same direction, and leave at the other end (Figure 1.17). (For fluid temperature variations, idealized as one-dimensional, refer Figure 2.2 of Chapter 2.) This arrangement has the lowest exchanger effectiveness among the single-pass exchangers for the same flow rates, capacity rate (mass × specific heat) ratio, and surface area. Moreover, the existence of large temperature differences at the inlet end may induce high thermal stresses in the exchanger wall at inlet. Parallelflows are advantageous. (a) In heating very viscous fluids, parallelflow provides for rapid heating. The quick change in viscosity results in reduced pumping power requirements through the heat exchanger, (b) where the more moderate mean metal temperatures of the tube walls are required, and (c) where the improvements in heat transfer rates compensate for the lower LMTD. Although this flow arrangement is not used widely, it is preferred for the following reasons [2]:

1. When there is a possibility that the temperature of the warmer fluid may reach its freezing point.
2. It provides early initiation of nucleate boiling for boiling applications.
3. For a balanced exchanger (i.e., heat capacity rate ratio $C^* = 1$), the desired exchanger effectiveness is low and is to be maintained approximately constant over a range of NTU values.
4. The application allows piping only suited to parallelflow.

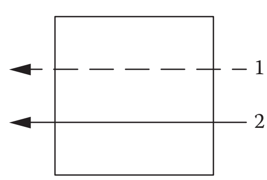


FIGURE 1.17 Parallelflow arrangement.

5. Temperature-sensitive fluids such as food products, pharmaceuticals, and biological products are less likely to be "thermally damaged" in a parallelflow heat exchanger.
6. Certain types of fouling such as chemical reaction fouling, scaling, corrosion fouling, and freezing fouling are sensitive to temperature. Where control of temperature-sensitive fouling is a major concern, it is advantageous to use parallelflow.

1.3.4.2 Counterflow Exchanger

In this type, as shown in Figure 1.18a, the two fluids flow parallel to each other but in opposite directions, and its temperature distribution may be idealized as shown in Figure 1.18b. Ideally, this is the most efficient of all flow arrangements for single-pass arrangements under the same parameters. Since the temperature difference across the exchanger wall at a given cross section is the lowest, it produces minimum thermal stresses in the wall for equivalent performance compared to other flow arrangements. In certain types of heat exchangers, counterflow arrangement cannot be achieved easily, due to manufacturing difficulties associated with the separation of the fluids at each end, and the design of inlet and outlet header design is complex and difficult [2].

1.3.4.3 Crossflow Exchanger

In this type, as shown in Figure 1.19, the two fluids flow normal to each other. Important types of flow arrangement combinations for a single-pass crossflow exchanger include the following:

- Both fluids unmixed
- One fluid unmixed and the other fluid mixed
- Both fluids mixed

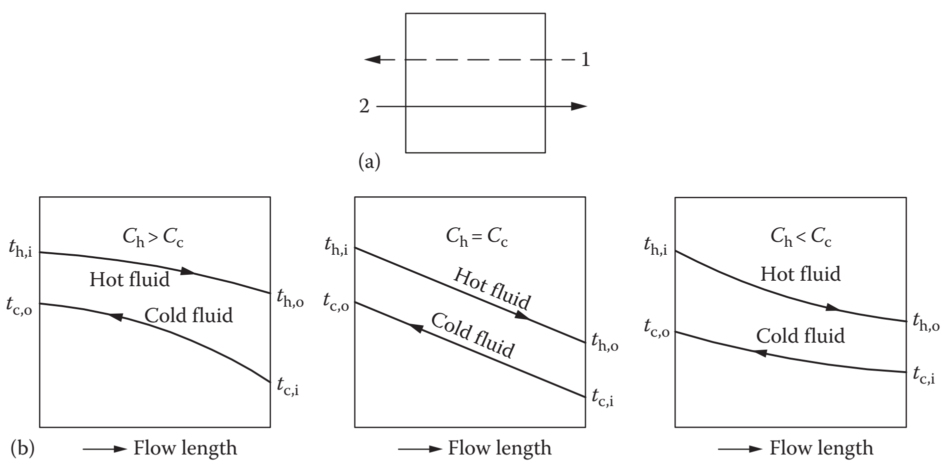


FIGURE 1.18 (a) Counterflow arrangement (schematic) and (b) temperature distribution (schematic). (*Note*: C_h and C_c are the heat capacity rate of hot fluid and cold fluid respectively, i refers to inlet, o refers to outlet conditions and t refers to fluid temperature.)

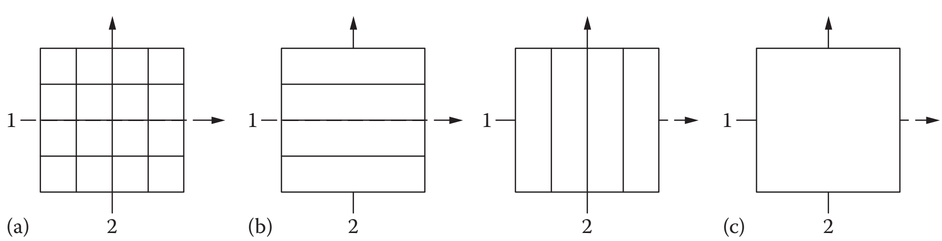


FIGURE 1.19 Crossflow arrangement: (a) unmixed–unmixed, (b) unmixed–mixed, and (c) mixed–mixed.

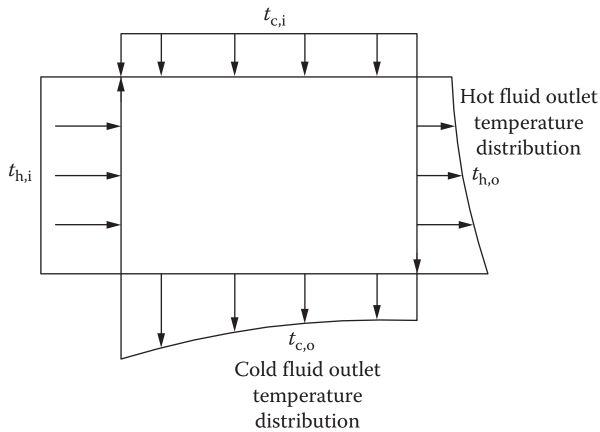


FIGURE 1.20 Temperature distribution for unmixed–unmixed crossflow arrangement.

A fluid stream is considered "unmixed" when it passes through individual flow passage without any fluid mixing between adjacent flow passages. Mixing implies that a thermal averaging process takes place at each cross section across the full width of the flow passage. A tube-fin exchanger with flat (continuous) fins and a plate-fin exchanger wherein the two fluids flow in separate passages (e.g., wavy fin, plain continuous rectangular or triangular flow passages) represent the unmixed–unmixed case. A crossflow tubular exchanger with bare tubes on the outside would be treated as the unmixed–mixed case, that is, unmixed on the tubeside and mixed on the outside. The both fluid mixed case is practically a less important case and represents a limiting case of some multipass shell and tube exchangers (TEMA E and J shell).

For the unmixed–unmixed case, fluid temperature variations are idealized as two-dimensional only for the inlet and outlet sections; this is shown in Figure 1.20. The thermal effectiveness for the crossflow exchanger falls in between those of the parallelflow and counterflow arrangements. This is the most common flow arrangement used for extended surface heat exchangers because it greatly simplifies the header design. If the desired heat exchanger effectiveness is generally more than 80%, the size penalty for crossflow may become excessive. In such a case, a counterflow unit is preferred [2]. In shell and tube exchangers, crossflow arrangement is used in the TEMA X shell having a single tube pass.

1.3.5 Classification According to Pass Arrangements

These are either single pass or multipass. A fluid is considered to have made one pass if it flows through a section of the heat exchanger through its full length once. In a multipass arrangement, a fluid is reversed and flows through the flow length two or more times.

1.3.5.1 Multipass Exchangers

When the design of a heat exchanger results in either extreme length, significantly low velocities, or low effectiveness, or due to other design criteria, a multipass heat exchanger or several single-pass exchangers in series or a combination of both is employed. Specifically, multipassing is resorted to increase the exchanger thermal effectiveness over the individual pass effectiveness. As the number of passes increases, the overall direction of the two fluids approaches that of a pure counterflow exchanger. The multipass arrangements are possible with compact, shell and tube, and plate exchangers.

1.3.6 Classification According to Phase of Fluids

1.3.6.1 Gas–Liquid

Gas–liquid heat exchangers are mostly tube-fin-type compact heat exchangers with the liquid on the tubeside. The radiator is by far the major type of liquid–gas heat exchanger, typically cooling the engine jacket water by air. Similar units are necessary for all the other water-cooled engines used in trucks, locomotives, diesel-powered equipment, and stationery diesel power plants. Other examples are air coolers, oil coolers for aircraft, intercoolers and aftercoolers in compressors, and condensers and evaporators of room air-conditioners. Normally, the liquid is pumped through the tubes, which have a very high convective heat transfer coefficient. The air flows in crossflow over the tubes. The heat transfer coefficient on the air side will be lower than that on the liquid side. Fins will be generally used on the outside of the tubes to enhance the heat transfer rate.

1.3.6.2 Liquid–Liquid

Most of the liquid–liquid heat exchangers are shell and tube type, and PHEs to a lesser extent. Both fluids are pumped through the exchanger, so the principal mode of heat transfer is forced convection. The relatively high density of liquids results in very high heat transfer rate, so normally fins or other devices are not used to enhance the heat transfer [4]. In certain applications, low-finned tubes, microfin tubes, and heat transfer augmentation devices are used to enhance the heat transfer.

1.3.6.3 Gas–Gas

This type of exchanger is found in exhaust gas–air preheating recuperators, rotary regenerators, intercoolers, and/or aftercoolers to cool supercharged engine intake air of some land-based diesel power packs and diesel locomotives, and cryogenic gas liquefaction systems. In many cases, one gas is compressed so that the density is high while the other is at low pressure and low density. Compared to liquid–liquid exchangers, the size of the gas–gas exchanger will be much larger, because the convective heat transfer coefficient on the gas side is low compared to the liquid side. Therefore, secondary surfaces are mostly employed to enhance the heat transfer rate.

1.3.7 Classification According to Heat Transfer Mechanisms

The basic heat transfer mechanisms employed for heat transfer from one fluid to the other are (1) single-phase convection, forced or free, (2) two-phase convection (condensation or evaporation) by forced or free convection, and (3) combined convection and radiation. Any of these mechanisms individually or in combination could be active on each side of the exchanger. Based on the phase change mechanisms, the heat exchangers are classified as (1) condensers and (2) evaporators.

1.3.7.1 Condensers

Condensers may be liquid (water) or gas (air) cooled. The heat from condensing streams may be used for heating fluid. Normally, the condensing fluid is routed (1) outside the tubes with a water-cooled steam condenser or (2) inside the tubes with gas cooling, that is, air-cooled condensers of refrigerators and air-conditioners. Fins are normally provided to enhance heat transfer on the gas side.

1.3.7.2 Evaporators

This important group of tubular heat exchangers can be subdivided into two classes: fired systems and unfired systems.

Fired systems: These involve the products of combustion of fossil fuels at very high temperatures but at ambient pressure (and hence low density) and generate steam under pressure. Fired systems are called boilers. A system may be a fire tube boiler (for small low-pressure applications) or a water tube boiler.

Unfired systems: These embrace a great variety of steam generators extending over a broad temperature range from high-temperature nuclear steam generators to very-low-temperature cryogenic gasifiers for liquid natural gas evaporation. Many chemical and food processing applications involve the use of steam to evaporate solvents, concentrate solutions, distill liquors, or dehydrate compounds.

1.3.8 Other Classifications

1.3.8.1 Micro Heat Exchanger

Micro- or microscale heat exchangers are heat exchangers in which at least one fluid flows in lateral confinements with typical dimensions below 1 mm and are fabricated via silicon micromachining, deep x-ray lithography, or nonlithographic micromachining [18]. The plates are stacked forming "sandwich" structures, as in the "large" plate exchangers. All flow configurations (cocurrent, countercurrent, and crossflow) are possible.

Typically, the fluid flows through a cavity called a microchannel. Microheat exchangers have been demonstrated with high convective heat transfer coefficient. Investigation of microscale thermal devices is motivated by the single-phase internal flow correlation for convective heat transfer:

$$h = \frac{\mathrm{Nu}\, k}{D_\mathrm{h}}$$

where

h is the heat transfer coefficient
Nu is the Nusselt number
k is the thermal conductivity of the fluid
D_h is the hydraulic diameter of the channel or duct

In internal laminar flows, the Nusselt number becomes a constant. As the Reynolds number is proportional to hydraulic diameter, fluid flow in channels of small hydraulic diameter will predominantly be laminar. This correlation therefore indicates that the heat transfer coefficient increases as the channel diameter decreases. Heat transfer enhancement in laminar flow is further discussed in Chapter 8 Section 8.3.3.

1.3.8.1.1 Advantages over Macroscale Heat Exchangers

- Substantially better performance
- Enhanced heat transfer coefficient with a large number of smaller channels
- Smaller size that allows for an increase in mobility and uses
- Light weight reduces the structural and support requirements
- Lower cost due to less material being used in fabrication

1.3.8.1.2 Applications of Microscale Heat Exchangers

Microscale heat exchangers are being used in the development of fuel cells. They are currently used in automotive industries, HVAC applications, aircraft, manufacturing industries, and electronics cooling.

1.3.8.1.3 Demerits of Microscale Heat Exchangers

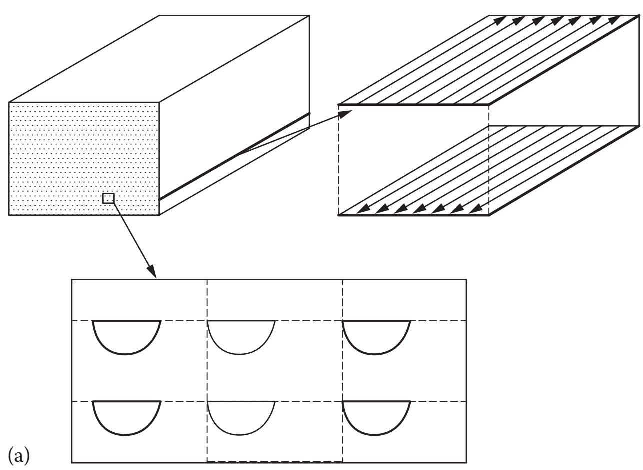
One of the main disadvantages of microchannel heat exchangers is the high-pressure loss that is associated with a small hydraulic diameter.

1.3.8.2 Printed Circuit Heat Exchanger

PCHE, developed by Heatric Division of Meggitt (UK) Ltd., is a promising heat exchanger because it is able to withstand pressures up to 50 MPa and temperatures from cryogenic to 700°C. It is extremely compact (the most common design feature to achieve compactness has been small channel size) and has high efficiency, of the order of 98%. It can handle a wide variety of clean fluids. The flow configuration can be either crossflow or counterflow. It will maintain parent material strength and can be made from stainless steel, nickel alloys, copper alloys, and titanium. Fluid flow channels are etched chemically on metal plates. It has a typical plate thickness of 1.6 mm, width 600 mm, and length 1200 mm. The channels are semicircular with 1–2 mm diameter. Etched plates are stacked and diffusion bonded together to fabricate as a block. The blocks are then welded together to form the complete heat exchanger core, as shown in Figure 1.21a.

HEATRIC PCHEs consist of diffusion-bonded heat exchanger core that are constructed from flat metal plates into which fluid flow channels are either chemically etched or pressed. The required configuration of the channels on the plates for each fluid is governed by the operating temperature and pressure-drop constraints for the heat exchange duty and the channels can be of unlimited variety and complexity. Fluid flow can be parallelflow, counterflow, crossflow, or a combination of these to suit the process requirements. Figure 1.21b through f shows HEATRIC PCHE.

The etched plates are then stacked and diffusion-bonded together to form strong, compact, all metal heat exchanger core. Diffusion bonding is a "solid-state joining" process entailing pressing metal surfaces together at temperatures below the melting point, thereby promoting grain growth between the surfaces. Under carefully controlled conditions, diffusion-bonded joints reach parent metal strength and stacks of plates are converted into solid blocks containing the fluid flow passages. The blocks are then welded together to form the complete heat exchange core. Finally, headers and nozzles are welded to the core in order to direct the fluids to the appropriate sets of passages. Welded and diffusion-bonded PCHEs employ no gaskets or braze material, resulting in superior integrity compared to other technologies that may use gaskets or brazing as part of their construction. (Gaskets or braze material can be potential sources of leakage, fluid incompatibility, and temperature limitations.) The mechanical design is normally of ASME VIII Division 1. Other design codes can be employed as required.



(a)

FIGURE 1.21 Printed circuit heat exchanger. (a) Heat exchanger block with flow channel.

(*continued*)

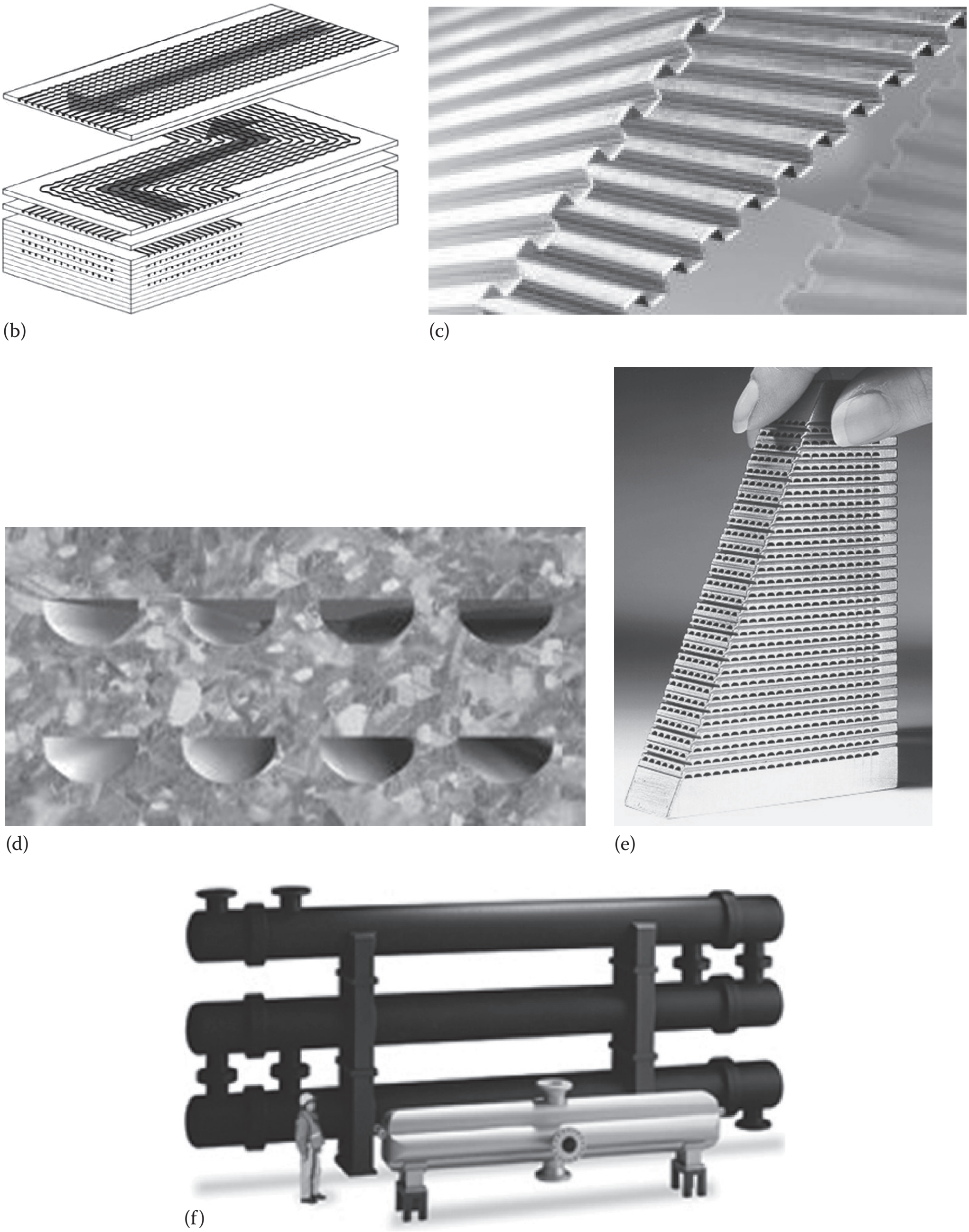


FIGURE 1.21 (continued) Printed circuit heat exchanger. (b) Flow channel, (c) and (d) section through flow channel, (e) diffusion bonded core, (f) comparison of size of PCHE shell and tube heat exchanger (smaller size) with a conventional exchanger (bigger size) for similar duty. (Courtesy of Heatric UK, Dorset, U.K.)

1.3.8.2.1 Materials of Construction

The majority of diffusion-bonded heat exchangers are constructed from 300 series austenitic stainless steel. Various other metals that are compatible with the diffusion-bonded process and have been qualified for use include 22 chromeduplex, copper–nickel, nickel alloys, and titanium.

1.3.8.2.2 Features of PCHE

Diffusion-bonded heat exchangers are highly compact and robust that are well established in the upstream hydrocarbon processing, petrochemical and refining industries. Various salient constructional and performance features are given next:

1. Compactness: Diffusion-bonded heat exchangers are 1/4–1/6th the conventional STHEs of the equivalent heat duty. This design feature has space and weight advantages, reducing exchanger size together with piping and valve requirements. The diffusion-bonded heat exchanger in the foreground of Figure 1.21e undertakes the same thermal duty, at the same pressure drop, as the stack of three shell and tube exchangers behind. PCHE might be judged as a promising compact heat exchanger for the high efficiency recuperator [19].
2. Process capability: They can withstand pressures of 600 bar (9000 psi) or excess and can cope with extreme temperatures, ranging from cryogenic to 900°C (1650°F).
3. Thermal effectiveness: Diffusion-bonded exchangers can achieve high thermal effectiveness of the order of 98% in a single unit.
4. They can incorporate more than two process streams into a single unit.
5. The compatibility of the chemical etching and diffusion-bonding process with a wide range of materials ensure that they are suitable for a range of corrosive and high purity streams.

1.3.8.3 Perforated Plate Heat Exchanger as Cryocoolers

High-efficiency compact heat exchangers are needed in cryocoolers to achieve very low temperatures. One approach to meet the requirements for compact and efficient cooling systems is the perforated PHE. Such heat exchangers are made up of a large number of parallel, perforated plates of high-thermal conductivity metal in a stacked array, with gaps between plates being provided by spacers. Gas flows longitudinally through the plates in one direction and other stream flows in the opposite direction through separated portions of the plates. Heat transfer takes place laterally across the plates from one stream to the other. The operating principles of this type of heat exchanger are described by Fleming [20]. The device employs plates of 0.81 mm thickness with holes of 1.14 mm in diameter and a resulting length-to-diameter ratio in the range of 0.5–1.0. The device is designed to operate from room temperature to 80 K. In order to improve operation of a compact cryocooler, much smaller holes, in the low-micron-diameter range, and thinner plates with high length-to-diameter ratio are needed. As per US Patent 5101894 [21], uniform, tubular perforations having diameters down to the low-micron-size range can be obtained. Various types of heat exchange devices including recuperative and regenerative heat exchangers may be constructed in accordance with the invention for use in cooling systems based on a number of refrigeration cycles such as the Linde–Hampson, Brayton, and Stirling cycles.

1.3.8.4 Scraped Surface Heat Exchanger

Scraped surface heat exchangers are used for processes likely to result in the substantial deposition of suspended solids on the heat transfer surface. Scraped surface heat exchangers can be employed in the continuous, closed processing of virtually any pumpable fluid or slurry involving cooking, slush freezing, cooling, crystallizing, mixing, plasticizing, gelling, polymerizing, heating, aseptic processing, etc. Use of a scraped surface exchanger prevents the accumulation of significant buildup of solid deposits. The construction details of scraped surface heat exchangers are explained in Ref. [4]. Scraped surface heat exchangers are essentially double pipe construction with the process fluid in the inner pipe and the cooling (water) or heating medium (steam) in the annulus. A rotating element is contained within the tube and is equipped with spring-loaded blades. In operation, the rotating shaft scraper blades continuously scrape product film from the heat transfer tube wall, thereby enhancing heat transfer and agitating the product to produce a homogenous mixture. For most applications, the shaft is mounted in the center of the heat transfer tube. An off-centered shaft mount or *eccentric* design is recommended for viscous and sticky

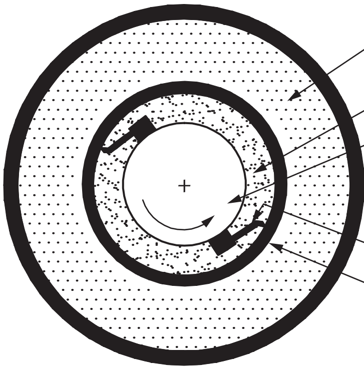


FIGURE 1.22 Scraped surface heat exchanger: principle.

products. This shaft arrangement increases product mixing and reduces the mechanical heat load. Oval tubes are used to process extremely viscous products. All pressure elements are designed in accordance with the latest ASME code requirements. The principle of working of scraped surface heat exchangers is shown in Figure 1.22. For scraped surface exchangers, operating costs are high and applications are highly specific [5]. Design is mostly done by vendors. The leading manufactures include HRS Heat Exchangers, Ltd., UK, and Waukesha Cherry-Burrell, USA.

1.3.8.4.1 Unicus Scraped Surface Heat Exchanger

Unicus™ is the trade name for scraped surface heat exchanger of HRS Heat Exchangers Ltd., UK, for high-fouling and viscous fluid applications. The design is based on STHE with scraping elements inside each interior tube. The scrapers are moved back and forth by hydraulic action. The scraping action has two very important advantages: any fouling on the tube wall is removed and the scraping movement introduces turbulence in the fluid increasing heat transfer.

1.3.8.4.1.1 Elements of the Unicus Unicus consists of three parts: a hydraulic cylinder that moves the scraper bars, the STHE part, and a chamber that separates both the elements. The hydraulic cylinder is connected to a hydraulic power pack. The smaller models of the Unicus range can be supplied with a pneumatic cylinder. The scraping system consists of a stainless steel rod to which the scraping elements are fitted, as shown in Figure 1.23, and Figure 1.24 shows Unicus scraped surface heat exchangers. The pictures show the various types of scrapers that can be applied. For each application, the optimal scraper is selected and fitted.

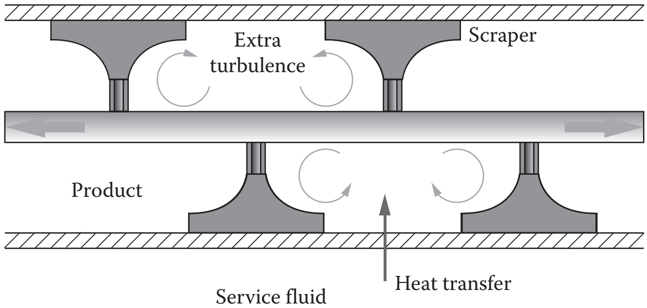
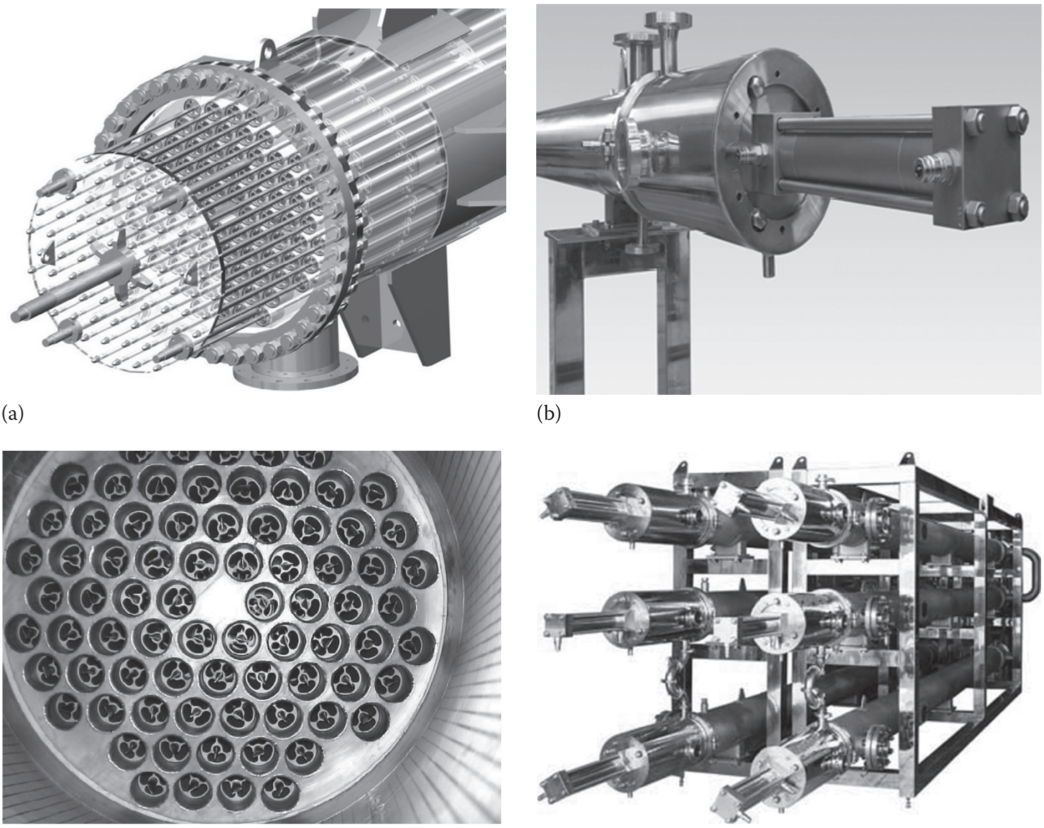


FIGURE 1.23 Principle of Unicus scraped surface heat exchanger working. (Courtesy of HRS Heat Exchangers Ltd, Herts, U.K.)



(a) (b) (c) (d)

FIGURE 1.24 Unicus dynamic scraped surface heat exchanger. (a) 3-D model, (b) hydraulic cylinder head, (c) a unit of Unicus, and (d) multiple Unicus units. (Courtesy of HRS Heat Exchangers Ltd, Herts, U.K.)

1.3.8.5 Graphite Heat Exchanger

Impervious graphite as a heat exchanger material is used for the construction of various types of heat exchangers such as STHE, cubic block heat exchanger, and plate and frame or gasketed heat exchangers (PHEs). Graphite tubes are used in STHE (refer to Chapter 13) and plates are used in PHEs (refer to Chapter 7) for special purpose applications. It resists a wide variety of inorganic and organic chemicals. Graphite heat exchangers are employed as boilers and condensers in the distillation by evaporation of hydrochloric acid and in the concentration of weak sulfuric acid and of rare earth chloride solutions. Since cubic heat exchanger cannot be treated in categorization of extended surface heat exchanger, the same is covered next.

Cubic heat exchanger: It is similar to the compact crossflow heat exchanger, consisting of drilled holes in two perpendicular planes. They are suitable when both the process streams are corrosive. With a cubic exchanger, a multipass arrangement is possible. It is manufactured by assembling of accurately machined and drilled graphite plates bonded together by synthetic resins, oven cured and sintered. Gasketed headers with nozzles are assembled on both sides to the block to form a block heat exchanger and are clamped together, as shown in Figure 1.25.

Modular-block cylindrical exchanger: In this arrangement, solid impervious graphite blocks have holes drilled in them. These blocks can be multistacked in a cylindrical steel shell that has

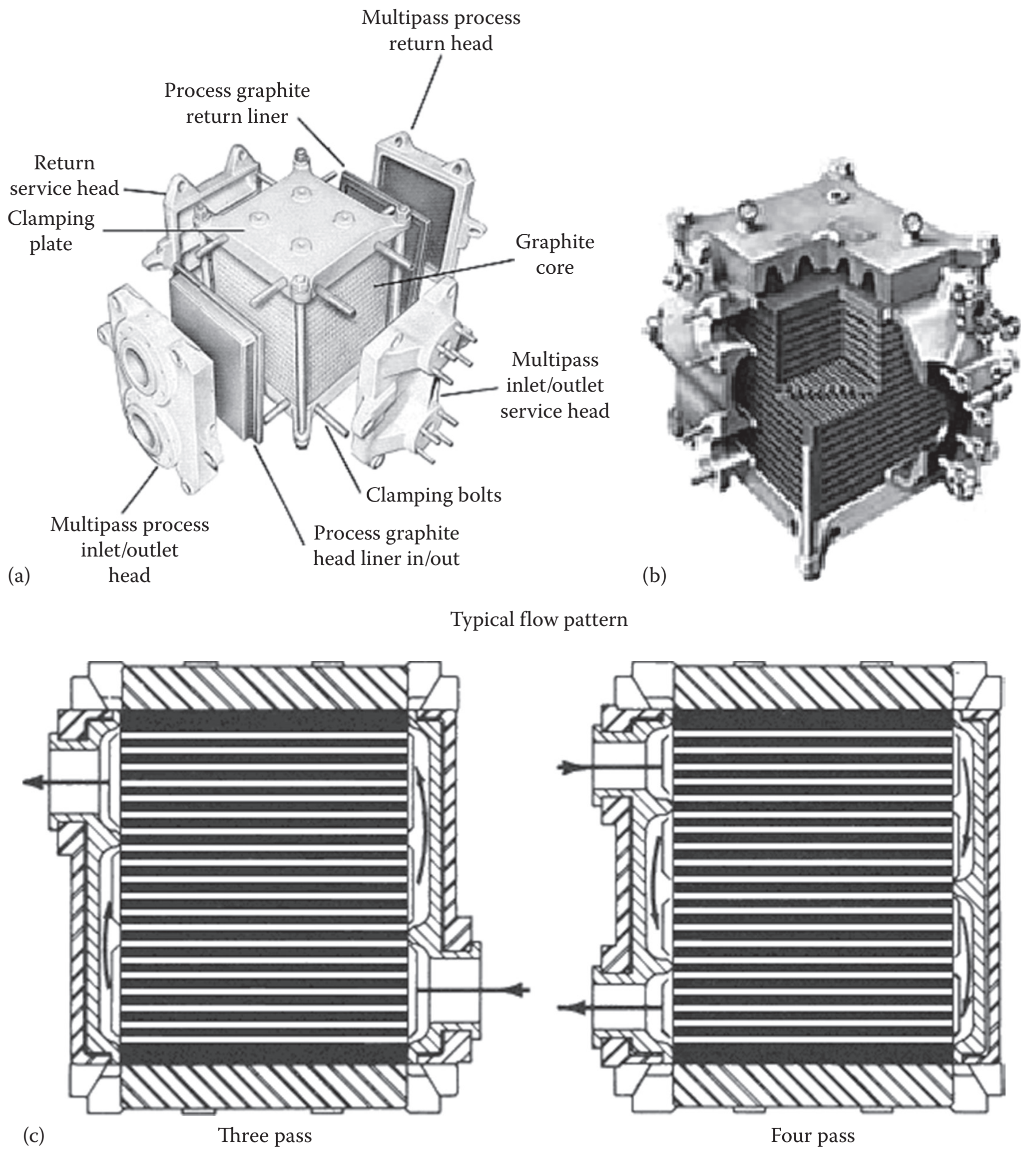


FIGURE 1.25 NK series multipass cubic graphite heat exchanger. (a) Construction details, (b) cut section, and (c) multipass flow pattern. (Courtesy of MERSEN, Paris La Défense, France.)

gland fittings. The process holes are axial and the service holes are transverse. The units are designed as evaporators and reboilers. A modular block Graphilor® exchanger is shown in Figure 1.26.

1.4 SELECTION OF HEAT EXCHANGERS

1.4.1 Introduction

Selection is the process in which the designer selects a particular type of heat exchanger for a given application from a variety of heat exchangers. There are a number of alternatives for selecting heat transfer equipment, but only one among them is the best for a given set of conditions. The heat exchanger selection criteria are discussed next.

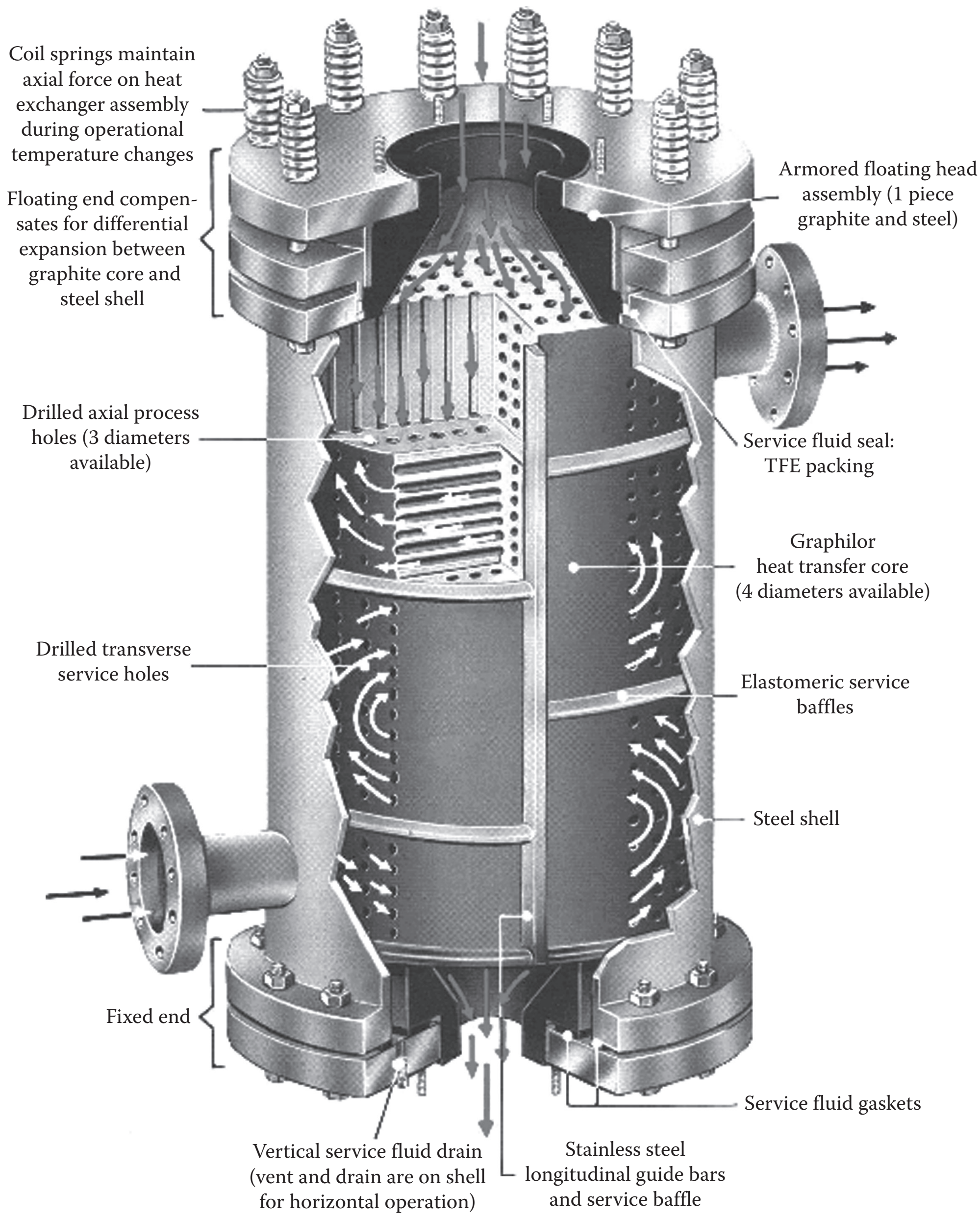


FIGURE 1.26 GRAPHILOR® cylindrical tubes (in block) heat exchanger. (Courtesy of MERSEN, Paris La Défense, France.)

1.4.2 Selection Criteria

Selection criteria are many, but primary criteria are type of fluids to be handled, operating pressures and temperatures, heat duty, and cost (see Table 1.1). The fluids involved in heat transfer can be characterized by temperature, pressure, phase, physical properties, toxicity, corrosivity, and fouling tendency. Operating conditions for heat exchangers vary over a very wide range, and a broad spectrum of demands is imposed for their design and performance. All of these must be considered when assessing the type of unit to be used [22]. When selecting a heat exchanger for a given duty, the following points must be considered:

- Materials of construction
- Operating pressure and temperature, temperature program, and temperature driving force
- Flow rates
- Flow arrangements
- Performance parameters—thermal effectiveness and pressure drops
- Fouling tendencies
- Types and phases of fluids
- Maintenance, inspection, cleaning, extension, and repair possibilities
- Overall economy
- Fabrication techniques
- Mounting arrangements: horizontal or vertical
- Intended applications

1.4.2.1 Materials of Construction

For reliable and continuous use, the construction materials for pressure vessels and heat exchangers should have a well-defined corrosion rate in the service environments. Furthermore, the material should exhibit strength to withstand the operating temperature and pressure. STHEs can be manufactured in virtually any material that may be required for corrosion resistance, for example, from nonmetals like glass, Teflon, and graphite to exotic metals like titanium, zirconium, tantalum, etc. Compact heat exchangers with extended surfaces are mostly manufactured from any metal that has drawability, formability, and malleability. Heat exchanger types like PHEs normally require a material that can be pressed or welded.

1.4.2.2 Operating Pressure and Temperature

1.4.2.2.1 Pressure

The design pressure is important to determine the thickness of the pressure-retaining components. The higher the pressure, the greater will be the required thickness of the pressure-retaining membranes and the more advantage there is to placing the high-pressure fluid on the tubeside. The pressure level of the fluids has a significant effect on the type of unit selected [22].

At low pressures, the vapor-phase volumetric flow rate is high and the low-allowable pressure drops may require a design that maximizes the area available for flow, such as crossflow or split flow with multiple nozzles.

At high pressures, the vapor-phase volumetric flow rates are lower and allowable pressure drops are greater. These lead to more compact units.

In general, higher heat transfer rates are obtained by placing the low-pressure gas on the outside of tubular surfaces.

Operating pressures of the gasketed PHEs and SPHEs are limited because of the difficulty in pressing the required plate thickness, and by the gasket materials in the case of PHEs. The floating nature of floating-head shell and tube heat exchangers and lamella heat exchangers limits the operating pressure.

1.4.2.2.2 Temperature

Design temperature: This parameter is important as it indicates whether a material at the design temperature can withstand the operating pressure and various loads imposed on the component. For low-temperature and cryogenic applications, toughness is a prime requirement, and for high-temperature applications the material has to exhibit creep resistance.

Temperature program: Temperature program in both a single-pass and multipass STHE decides (1) the mean metal temperatures of various components like shell, tube bundle, and tubesheet, and

(2) the possibility of temperature cross. The mean metal temperatures affect the integrity and capability of heat exchangers and thermal stresses induced in various components.

Temperature driving force: The effective temperature driving force is a measure of the actual potential for heat transfer that exists at the design conditions. With a counterflow arrangement, the effective temperature difference is defined by the log mean temperature difference (LMTD). For flow arrangements other than counterflow arrangement, LMTD must be corrected by a correction factor, F. The F factor can be determined analytically for each flow arrangement but is usually presented graphically in terms of the thermal effectiveness P and the heat capacity ratio R for each flow arrangement.

Influence of operating pressure and temperature on selection of some types of heat exchangers: The influence of operating pressure and temperature on selection of STHE, compact heat exchanger, gasketed PHE, and spiral exchanger is discussed next.

Shell and tube heat exchanger: STHE units can be designed for almost any combination of pressure and temperature. In extreme cases, high pressure may impose limitations by fabrication problems associated with material thickness, and by the weight of the finished unit. Differential thermal expansion under steady conditions can induce severe thermal stresses either in the tube bundle or in the shell. Damage due to flow-induced vibration on the shellside is well known. In heat-exchanger applications where high heat transfer effectiveness (close approach temperature) is required, the standard shell and tube design may require a very large amount of heat transfer surface [23]. Depending on the fluids and operating conditions, other types of heat-exchanger design should be investigated.

Compact heat exchanger: Compact heat exchangers are constructed from thinner materials; they are manufactured by mechanical bonding, soldering, brazing, welding, etc. Therefore, they are limited in operating pressures and temperatures.

Gasketed plate heat exchangers and spiral exchangers: Gasketed PHEs and spiral exchangers are limited by pressure and temperature, wherein the limitations are imposed by the capability of the gaskets.

1.4.2.3 Flow Rate

Flow rate determines the flow area: the higher the flow rate, the higher will be the crossflow area. Higher flow area is required to limit the flow velocity through the conduits and flow passages, and the higher velocity is limited by pressure drop, impingement, erosion, and, in the case of shell and tube exchanger, by shellside flow-induced vibration. Sometimes, a minimum flow velocity is necessary to improve heat transfer to eliminate stagnant areas and to minimize fouling.

1.4.2.4 Flow Arrangement

As defined earlier, the choice of a particular flow arrangement is dependent upon the required exchanger effectiveness, exchanger construction type, upstream and downstream ducting, packaging envelope, and other design criteria.

1.4.2.5 Performance Parameters: Thermal Effectiveness and Pressure Drops

Thermal effectiveness: For high-performance service requiring high thermal effectiveness, use brazed plate-fin exchangers (e.g., cryogenic service) and regenerators (e.g., gas turbine applications), use tube-fin exchangers for slightly less thermal effectiveness in applications, and use shell and tube units for low-thermal effectiveness service.

Pressure drop: Pressure drop is an important parameter in heat exchanger design. Limitations may be imposed either by pumping cost or by process limitations or both. The heat exchanger should be designed in such a way that unproductive pressure drop is avoided to the maximum extent in areas like inlet and outlet bends, nozzles, and manifolds. At the same time, any pressure-drop limitation that is imposed must be utilized as nearly as possible for an economic design.

1.4.2.6 Fouling Tendencies

Fouling is defined as the formation on heat exchanger surfaces of undesirable deposits that impede the heat transfer and increase the resistance to fluid flow, resulting in higher pressure drop. The growth of these deposits causes the thermohydraulic performance of heat exchanger to decline with time. Fouling affects the energy consumption of industrial processes, and it also decides the amount of extra material required to provide extra heat transfer surface to compensate for the effects of fouling. Compact heat exchangers are generally preferred for nonfouling applications. In a shell and tube unit, the fluid with more fouling tendencies should be put on the tubeside for ease of cleaning. On the shellside with cross baffles, it is sometimes difficult to achieve a good flow distribution if the baffle cut is either too high or too low. Stagnation in any regions of low velocity behind the baffles is difficult to avoid if the baffles are cut more than about 20%–25%. PHEs and spiral plate exchangers are better chosen for fouling services. The flow pattern in PHE induces turbulence even at comparable low velocities; in the spiral units, the scrubbing action of the fluids on the curved surfaces minimizes fouling. Also consider Philips RODbaffle heat exchanger, TWISTED TUBE® heat exchanger, Helixchanger® heat exchanger or EMbaffle® heat exchanger to improve flow velocity on shellside, enhance heat transfer performance and reduce fouling tendencies on shellside.

1.4.2.7 Types and Phases of Fluids

The phase of the fluids within a unit is an important consideration in the selection of the heat exchanger type. Various combinations of fluid phases dealt in heat exchangers are liquid–liquid, liquid–gas, and gas–gas. Liquid-phase fluids are generally the simplest to deal with. The high density and the favorable values of many transport properties allow high heat transfer coefficients to be obtained at relatively low-pressure drops [4].

1.4.2.8 Maintenance, Inspection, Cleaning, Repair, and Extension Aspects

Consider the suitability of various heat exchangers as regards maintenance, inspection, cleaning, repair, and extension. For example, the pharmaceutical, dairy, and food industries require quick access to internal components for frequent cleaning. Since some of the heat exchanger types offer great variations in design, this must be kept in mind when designing for a certain application. For instance, consider inspection and manual cleaning. Spiral plate exchangers can be made with both sides open at one edge, or with one side open and one closed. They can be made with channels between 5 and 25 mm wide, with or without studs. STHE can be made with fixed tubesheets or with a removable tube bundle, with small- or large-diameter tubes, or small or wide pitch. A lamella heat exchanger bundle is removable and thus fairly easy to clean on the shellside. Inside, the lamella, however, cannot be drilled to remove the hard fouling deposits. Gasketed PHEs are easy to open, especially when all nozzles are located on the stationary end-plate side. The plate arrangement can be changed for other duties within the frame and nozzle capacity.

Repair of some of the shell and tube exchanger components is possible, but the repair of expansion joint is very difficult. Tubes can be renewed or plugged. Repair of compact heat exchangers of tube-fin type is very difficult except by plugging of the tube. Repair of the plate-fin exchanger is generally very difficult. For these two types of heat exchangers, extension of units for higher thermal duties is generally not possible. All these drawbacks are easily overcome in a PHE. It can be easily repaired, and plates and other parts can be easily replaced. Due to modular construction, PHEs possess the flexibility of enhancing or reducing the heat transfer surface area, modifying the pass arrangement, and addition of more than one duty according to the heat transfer requirements at a future date.

1.4.2.9 Overall Economy

There are two major costs to consider in designing a heat exchanger: the manufacturing cost and the operating costs, including maintenance costs. In general, the less the heat transfer surface area and less the complexity of the design, the lower is the manufacturing cost. The operating cost is the pumping cost due to pumping devices such as fans, blowers, and pumps. The maintenance costs include costs of spares that require frequent renewal due to corrosion, and costs due to corrosion/fouling prevention and control. Therefore, the heat exchanger design requires a proper balance between thermal sizing and pressure drop.

1.4.2.10 Fabrication Techniques

Fabrication techniques are likely to be the determining factor in the selection of a heat transfer surface matrix or core. They are the major factors in the initial cost and to a large extent influence the integrity, service life, and ease of maintenance of the finished heat exchanger [24]. For example, shell and tube units are mostly fabricated by welding, plate-fin heat exchangers and automobile aluminum radiators by brazing, copper–brass radiators by soldering, most of the circular tube-fin exchangers by mechanical assembling, etc.

1.4.2.11 Choice of Unit Type for Intended Applications

According to the intended applications, the selection of heat exchangers will follow the guidelines given in Table 1.2.

TABLE 1.2
Choice of Heat Exchanger Type for Intended Applications

Application	Remarks
Low-viscosity fluids	For high temperature/pressures, use STHE or double-pipe heat exchanger. Use PHE or LHE for low temperature/pressure applications.
Low-viscosity liquid to steam	Use STHE in carbon steel.
Medium-viscosity fluids	Use PHE or with high solids content, use SPHE.
High-viscosity fluids	PHE offers the advantages of good flow distribution. For extreme viscosities, the SPHE is preferred.
Fouling liquids	Use STHE with removable tube bundle. SPHE or PHE is preferred due to good flow distribution. Use PHE if easy access is of importance. Also consider Philips RODbaffle heat exchanger, *TWISTED TUBE® heat exchanger and* Helixchanger® heat exchanger, and EMbaffle® heat exchanger to improve flow velocity on the shellside, enhance heat transfer performance, and reduce fouling tendencies on shellside.
Slurries, suspensions, and pulps	SPHE offers the best characteristics. Also consider free flow PHE or wide gap PHE, or scraped surface heat exchanger.
Heat-sensitive liquids	PHE fulfills the requirements best. Also consider SPHE.
Cooling with air	Extended surface types like tube-fin heat exchanger or PFHE.
Gas or air under pressure	Use STHE with extended surface on the gas side or brazed plate-fin exchanger made of stainless steel or nickel alloys.
Cryogenic applications	Brazed aluminum plate-fin exchanger, coiled tube heat exchangers, or PCHE.
Vapor condensation	Surface condensers of STHE in carbon steel are preferred. Also consider SPHE or brazed plate heat exchanger.
Vapor/gas partial condensation	Choose SPHE.
Refrigeration and air conditioning applications	Finned tube heat exchangers, special types of PHEs, brazed PHE up to 200°C.
Air–air or gas–gas applications	Regenerators and plate-fin heat exchangers. Also consider STHE.
Viscous products, aseptic products, jam, food and meat processing, heat sensitive products and particulate laden products	Scraped surface heat exchanger.

Note: STHE, shell and tube heat exchanger; PHE, gasketed plate heat exchanger; SPHE, spiral plate heat exchanger; LHE, lamella heat exchanger; PCHE, printed circuit heat exchangers; CTHE, coiled tube heat exchanger; PFHE, plate-fin heat exchanger.

1.5 REQUIREMENTS OF HEAT EXCHANGERS

Heat exchangers have to fulfill the following requirements:

- High thermal effectiveness
- Pressure drop as low as possible
- Reliability and life expectancy
- High-quality product and safe operation
- Material compatibility with process fluids
- Convenient size, easy for installation, reliable in use
- Easy for maintenance and servicing
- Light in weight but strong in construction to withstand the operational pressures and vibrations especially heat exchangers for military applications
- Simplicity of manufacture
- Low cost
- Possibility of effecting repair to maintenance problems

The heat exchanger must meet normal process requirements specified through problem specification and service conditions for combinations of the clean and fouled conditions, and uncorroded and corroded conditions. The exchanger must be maintainable, which usually means choosing a configuration that permits cleaning as required and replacement of tubes, gaskets, and any other components that are damaged by corrosion, erosion, vibration, or aging. This requirement may also place limitations on space for tube bundle pulling, to carry out maintenance around it, lifting requirements for heat exchanger components, and adaptability for in-service inspection and monitoring.

REFERENCES

1. Shah, R. K., Classification of heat exchangers, in *Heat Exchangers: Thermal-Hydraulic Fundamentals and Design* (S. Kakac, A. E. Bergles, and F. Mayinger, eds.), Hemisphere, Washington, DC, 1981, pp. 9–46.
2. Shah, R. K. and Sekulic, D. P., *Fundamentals of Heat Exchanger Design*, John & Wiley, New York, 2003.
3. Gupta, J. P., *Fundamentals of Heat Exchanger and Pressure Vessel Technology*, Hemisphere, Washington, DC, 1986.
4. Walker, G., *Cryocoolers*, *Part 2: Applications*, Plenum Press, New York, 1983.

5a. Larowski, A. and Taylor, M. A., *Systematic Procedures for Selection of Heat Exchangers*, C58/82, Institution of Mechanical Engineers, London, U.K., 1982, pp. 32–56.

5b. Larowski, A. and Taylor, M. A., *Systematic Procedure for Selection of Heat Exchangers*, *Proc. Inst. Mech. Eng.*, *197A,* 51–69 (1983).

6. Kern, D. Q., *Process Heat Transfer*, McGraw-Hill, New York, 1950.
7. Chisholm, D. (ed.), *Developments in Heat Exchanger Technology—1*, Applied Science Publishers, London, England, 1980.
8. Minton, P., Process heat transfer, *Proceedings of the 9th International Heat Transfer Conference*, Heat Transfer 1990–Jerusalem, Israel, Paper No. KN–2, *1*, 355–362 (1990).
9. Timmerhaus, K. D. and Flynn, T. M., *Cryogenic Progress Engineering*, Plenum Press, New York, 1989.
10. Muoio, J. M., Glass as a material of construction for heat transfer equipment, *Industrial Heat Exchangers Conference Proceedings* (A. J. Hayes, W. W. Liang, S. L. Richlen, and E. S. Tabb, eds.), American Society for Metals, Metals Park, OH, 1985, pp. 385–390.
11. Yilmaz, S. and Samuelson, B., Vertical thermosyphon boiling in spiral plate heat exchangers, in *Heat Transfer—1983*, *Seattle, Chem. Eng. Prog. Symp. Ser.*, *79*(225), 47–53 (1983).
12. Markovitz, R. E., Picking the best vessel jacket, *Chem. Eng.*, *78*(26), 156–162 (1971).
13. Usher, J. D. and Cattell, G. S., Compact heat exchangers, in *Developments in Heat Exchanger Technology—1* (D. Chisholm, ed.), Applied Science Publishers Ltd., London, U.K., 1980, pp. 127–152.

14. Reay, D. A., Heat exchangers for waste heat recovery, International Research and Development Co., Ltd., New Castle upon Tync, U.K., in *Developments in Heat Exchanger Technology—1* (D. Chisholm, ed.), Applied Science Publishers Ltd., London, U.K., 1980, pp. 233–256.
15. Butterworth, D. and Mascone, C. F., Heat transfer heads into the 21 century, *Chem. Eng. Prog.*, September, 30–37 (1991).
16a. Taylor, M. A. (ed), *Plate-Fin Heat Exchangers*, *Guide to Their Specification and Use*, HTFS (Harwell Laboratory), Oxon, U.K., 1980.
16b. ALPEMA, The standards of the Brazed Aluminium Plate-Fin Heat Exchanger Manufacturer's Association, 3rd ed., Didcot, Oxon, U.K., 2010. (www.alpema.org).
17. API Standard 660, *Shell and Tube Exchangers for General Refinery Services*, 7th edn., American Petroleum Institute, Washington, DC, 2002.
18a. Steinke, M. E. and Kandlikar, G., Single-phase heat transfer enhancement techniques in microchannel and minichannel flows, in *Microchannels and Minichannels—2004*, ASME, Rochester, NY, June 17–19, 2004.
18b. Tuckerman, D. B. and Pease, R. F. W., High-performance heat sinking for VLSI, *IEEE Electron Device Letters*, Vol. 2 (5), May 1981, pp. 126–129.
19. Nikitin, K., Kato, Y. and Ngo, L., Thermal-Hydraulic Performance of Printed Circuit Heat Exchanger in Supercritical CO_2 Cycle, Research Laboratory for Nuclear Reactors, Tokyo Institute of Technology, Okayama, Tokyo, Japan, 2005.
20. Fleming, R. B., A Compact Perforated Plate Heat Exchanger, in *Advances in Cryogenic Engineering*, Vol. 14, Plenum Press, New York, 1969, pp. 196–204.
21. Hendricks, J. B., Perforated plate heat exchanger and method of fabrication, Patent number: 5101894, Apr 7, 1992.
22. Gollin, M., Heat exchanger design and rating, in *Handbook of Applied Thermal Design* (E. C. Guyer, ed.), Chapter 2, Part 7, McGraw-Hill, New York, 1984, pp. 7-24–7-36.
23. Caciula, L. and Rudy, T. M., Prediction of plate heat exchanger performance, *D Symp. Ser.*, *Heat Transfer—1983*, Seattle, WA, pp. 76–89.
24. Fraas, A. P. and Ozisik, M. N., *Heat Exchanger Design*, John Wiley & Sons, New York, 1965.

SUGGESTED READINGS

Bell, K. J. and Mueller, A. C., *Wolverine Heat Transfer Data Book II*, Wolverine Division of UOP Inc., Decatur, AL, 1984.

Scaccia, C. and Theoclitus, G., Types, performance and applications, in *The Chemical Engineering Guide to Heat Transfer, Volume I: Plant Principles* (K. J. McNaughton and the Staff of Chemical Engineering, eds.), Hemisphere/McGraw-Hill, New York, 1986, pp. 3–14.

Shah, R. K., Classification of heat exchangers, in *Low Reynolds Number Flow Heat Exchangers* (S. Kakac, R. K. Shah, and A. E. Bergles, eds.), Hemisphere, Washington, DC, 1983, pp. 9–19.

Shah, R. K., What's new in heat exchanger design, *ASME*, *Mech. Eng., 106,* 50–59 (1984).

Sukhatme, S. P. and Devotta, S., Classification of heat transfer equipment, in *Heat Transfer Equipment Design* (R. K. Shah, E. C. Subbarao, and R. A. Mashelikar, eds.), Hemisphere, Washington, DC, 1988, pp. 7–16.

Thome, R. T., *Wolverine Heat Transfer Engineering Data Book III*, Wolverine Division of UOP Inc., Decatur, AL, 2004.

BIBLIOGRAPHY

Afgan, N. H. and Schlunder, E. U. (eds.), *Heat Exchangers: Design and Theory Sourcebook*, McGraw-Hill, New York, 1974.

Apblett, W. R. (ed.), *Shell and Tube Heat Exchangers*, American Society for Metals, Metals Park, OH, 1982.

Bhatia, M. V. and Cheremisinoff, N. P., *Heat Transfer Equipment*, Process equipment series, Vol. 2, Technomic, Westport, CT, 1980.

Bryers, R. W. (ed.), *Fouling of Heat Exchanger Surfaces*, Engineering Foundation, New York, 1983.

Chen, S. S., *Flow Induced Vibration of Circular Cylindrical Structures*, Hemisphere, Washington, DC, 1987.

Cheremisinoff, N. (ed.), *Handbook of Heat and Mass Transfer*, *Vol. 1—Heat Transfer Operation* & *Vol. 2—Mass Transfer and Reactor Design*, Gulf Publishing Company, Houston, TX, pp. 767–805, 1986.

Cheremisinoff, N. P., *Heat Transfer Pocket Handbook*, Gulf Publishing Company, Books Division, Houston, TX, 1984.

Chisholm, D. (ed.), *Heat Exchanger Technology*, Elsevier Applied Science, New York, 1988.

Effectiveness Ntu Relationships for Design and Performance Evaluation of Multi-pass Crossflow Heat Exchangers, Engineering Sciences Data Unit Item 87020, ESDU International, McLean, VA, October 1987.

Effectiveness Ntu Relationships for Design and Performance Evaluation of Two-Stream Heat Exchangers, Engineering Sciences Data Unit Item 86018, ESDU International, McLean, VA, July 1986.

Foster, B. D. and Patton, J. B. (eds.), *Ceramic Heat Exchangers*, American Ceramic Society, Columbus, OH, 1985.

Fraas, A. P., *Heat Exchanger Design*, 2nd edn., John Wiley & Sons, New York, 1989.

Ganapathy, V., *Applied Heat Transfer*, PennWell Publishing Co., Tulsa, OK, 1982.

Garrett-Price, B. A., Smith, S. A., Watts, R. L., Knudsen, J. C., Marner, W. J., and Suitor, J. W., *Fouling of Heat Exchangers*, Noyes, Park Ridge, NJ, 1985.

Hayes, A. J., Liang, W. W., Richlen, S. L., and Tabb, E. S. (eds.), *Industrial Heat Exchangers Conference Proceedings*, American Society for Metals, Metals Park, OH, 1985.

Hewitt, G. F. (coordinating editor), *Hemisphere Handbook of Heat Exchanger Design*, Hemisphere, New York, 1990.

Hewitt, G. F., Shires, G. L., and Bott, T. R., *Process Heat Transfer*, CRC Press, Boca Raton, FL, 1994.

Hewitt, G. F. and Whalley, P. B., *Handbook of Heat Exchanger Calculations*, Hemisphere, Washington, DC, 1989.

Hryniszak, W., *Heat Exchangers: Application to Gas Turbines*, Butterworth Scientific Publications, London, U.K., 1958.

Hussain, H., *Heat Transfer in Counterflow*, *Parallel Flow and Cross Flow*, McGraw-Hill, New York, 1982.

Jakob, M., *Heat Transfer*, John Wiley & Sons, New York, 1957.

Kakac, S., Bergles, A. E., and Fernandes, E. O. (eds.), *Two-Phase Flow Heat Exchangers: Thermal Hydraulic Fundamentals and Design*, Kluwer Academic Publishers, Dordrecht, the Netherlands, 1988.

Kakac, S., Bergles, A. E., and Mayinger, F. (eds.), *Heat Exchangers: Thermal-Hydraulic Fundamentals and Design*, Hemisphere/McGraw-Hill, Washington, DC, 1981.

Kakac, S., Shah, R. K., and Aung, W. (eds.), *Handbook of Single Phase Convective Heat Transfer*, John Wiley & Sons, New York, 1987.

Kakac, S., Shah, R. K., and Bergles, A. E. (eds.), *Low Reynolds Number Flow Heat Exchangers*, Hemisphere, Washington, DC, 1983.

Kays, W. M. and London, A. L., *Compact Heat Exchangers*, 3rd edn., McGraw-Hill, New York, 1984.

Kern, D. Q. and Kraus, A. D., *Extended Surface Heat Transfer*, McGraw-Hill, New York, 1972, pp. 439–641.

Kraus, A. D., *Analysis and Evaluation of Extended Surface Thermal Systems*, Hemisphere, Washington, DC, 1982.

Levenspiel, O., *Engineering Flow and Heat Exchange*, Plenum Press, New York, 1984.

Mahajan, K. K., *Design of Process Equipments*, Pressure Vessel Handbook Publishing Company, Tulsa, OK, 1985.

Manzoor, M., *Heat Flow through Extended Surface Heat Exchangers*, Springer-Verlag, Berlin, Germany, 1984.

Martin, H., *Heat Exchangers*, Hemisphere, Washington, DC, 1992.

McNaughton, K. J. and the Staff of Chemical Engineering (eds.), *The Chemical Engineering Guide to Heat Transfer: Vol. I: Plant Principles*, *Vol. II: Equipment*, Hemisphere/McGraw-Hill, New York, 1986.

Melo, L. F., Bott, T. R., and Bernardo, C. A. (eds.), *Advances in Fouling Science and Technology*, Kluwer Academic Publishers, Dordrecht, the Netherlands, 1988.

Mondt, J. R., Regenerative Heat Exchangers: The Elements of Their Design, Selection and Use, Research Publication GMR-3396, General Motors Research Laboratories, Warren, MI, 1980.

Mori, Y., Sheindlin, A. E., and Afgan, N. H. (eds.), *High Temperature Heat Exchangers*, Hemisphere, Washington, DC, 1986.

Palen, J. W. (ed.), *Heat Exchanger Sourcebook*, Hemisphere, Washington, DC, 1987.

Perry, R. H. and Chilton, C. H. (eds.), *Chemical Engineers' Handbook*, 5th edn., McGraw-Hill, New York, 1973.

Reay, D. A., *Heat Recovery Systems*, E. and F. N. Spon Ltd., London, U.K., 1979.

Rohsenow, W. M., Hartnett, J. P., and Ganic, E. N. (eds.), *Handbook of Heat Transfer Applications*, McGraw-Hill, New York, 1985.

Saunders, E. A. D., *Heat Exchangers: Selection*, *Design and Construction*, Addison Wesley Longman, Reading, MA, 1989.

Schlunder, E. U. (editor-in-chief), *Heat Exchanger Design Handbook*, in five volumes (Vol. 1—*Heal Exchanger Theory*, Vol. 2—*Fluid Mechanics and Heat Transfer*, Vol. 3—*Thermal and Hydraulic Design of Heat Exchangers*, Vol. 4—*Mechanical Design of Heat Exchangers*, and Vol. 5—*Physical Properties*), Hemisphere, Washington, DC, 1983.

Schmidt, F. W. and Willmott, A. J., *Thermal Energy Storage and Regeneration*, Hemisphere/McGraw-Hill, Washington, DC, 1981.

Shah, R. K., Kraus, A. D., and Metzger, D. (eds.), *Compact Heat Exchangers—A Festschrift for A. L. London*, Hemisphere, Washington, DC, 1990, pp. 31–90.

Shah, R. K. and London, A. L., *Laminar Flow Forced Convection in Ducts*, Supplement 1 to advances in heat transfer series, Academic Press, New York, 1978.

Shah, R. K. and Mueller, A. C., Heat exchanger, in *Ullmann's Encyclopedia of Industrial Chemistry*, Unit Operations II, Vol. B3, Chapter 2, VCH Publishers, Weinheim, Germany, 1989.

Shah, R. K., Subbarao, E. C., and Mashelkar, R. A. (eds.), *Heat Transfer Equipment Design*, Hemisphere, Washington, DC, 1988.

Sheindlin, A. E. (ed.), *High Temperature Equipment*, Hemisphere, Washington, DC, 1986.

Singh, K. P. and Soler, A. I., *Mechanical Design of Heat Exchangers and Pressure Vessel Components*, Arcturus Publishers, Cherry Hill, NJ, 1984.

Somerscales, E. F. C. and Knudsen, J. G. (eds.), *Fouling of Heat Transfer Equipment*, Hemisphere/McGraw-Hill, Washington, DC, 1981.

Stasiulevicius, J. and Skrinska, A., *Heat Transfer of Finned Tube Bundles in Crossflow*, Hemisphere, Washington, DC, 1987.

Taborek, J., Hewitt, G. F., and Afgan, N. (eds.), *Heat Exchangers: Theory and Practice*, Hemisphere/McGraw-Hill, Washington, DC, 1983.

Walker, G., *Industrial Heat Exchangers—A Basic Guide*, Hemisphere/McGraw-Hill, New York, 1982.

Yokell, S., *A Working Guide to Shell and Tube Heat Exchangers*, McGraw-Hill, New York, 1990.

Zukauskas, A. A., *High Performance Single-Phase Heat Exchangers* (J. Karni, English edition editor), Hemisphere, Washington, DC, 1989.

Zukauskas, A. and Ulinskas, R., *Heat Transfer in Tube Banks in Crossflow*, Hemisphere, Washington, DC, 1988.

2 Heat Exchanger Thermohydraulic Fundamentals

2.1 HEAT EXCHANGER THERMAL CIRCUIT AND OVERALL CONDUCTANCE EQUATION

In order to develop relationships between the heat transfer rate q, surface area A, fluid terminal temperatures, and flow rates in a heat exchanger, the basic equations used for analysis are the energy conservation and heat transfer rate equations [1]. The energy conservation equation for an exchanger having an arbitrary flow arrangement is

$$q = C_{\mathrm{h}}(t_{\mathrm{h,i}} - t_{\mathrm{h,o}}) = C_{\mathrm{c}}(t_{\mathrm{c,o}} - t_{\mathrm{c,i}}) \tag{2.1}$$

and the heat transfer rate equation is

$$q = UA\Delta t_{\mathrm{m}} = \frac{\Delta t_{\mathrm{m}}}{R_{\mathrm{o}}} \tag{2.2}$$

where

- Δt_{m} is the true mean temperature difference (MTD), which depends upon the exchanger flow arrangement and the degree of fluid mixing within each fluid stream
- C_{c} is the capacity rate of the cold fluid, $(Mc_{\mathrm{p}})_{\mathrm{c}}$
- C_{h} is the capacity rate of the hot fluid, $(Mc_{\mathrm{p}})_{\mathrm{h}}$
- $t_{\mathrm{c,i}}$ and $t_{\mathrm{c,o}}$ are cold fluid terminal temperatures (inlet and outlet)
- $t_{\mathrm{h,i}}$ and $t_{\mathrm{h,o}}$ are hot fluid terminal temperatures (inlet and outlet)

The heat exchanger thermal circuit variables and overall conduction described here are based on Refs. [1,2].

The inverse of the overall thermal conductance UA is referred to as the overall thermal resistance R_{o}, and it is made up of component resistances in series as shown in Figure 2.1:

$$R_{\mathrm{o}} = R_{\mathrm{h}} + R_1 + R_{\mathrm{w}} + R_2 + R_{\mathrm{c}} \tag{2.3}$$

where the parameters of the right-hand side of Equation 2.3 are R_{h}, hot side film convection resistance, $1/(\eta_{\mathrm{o}}hA)_{\mathrm{h}}$; R_1, thermal resistance due to fouling on the hot side given in terms of fouling resistance $R_{\mathrm{f,h}}$ (i.e., values tabulated in standards or textbooks), $R_{\mathrm{f,h}}/(\eta_{\mathrm{o}}A)_{\mathrm{h}}$; R_{w}, thermal resistance of the separating wall, expressed for a flat wall by

$$R_{\mathrm{w}} = \frac{\delta}{A_{\mathrm{w}}k_{\mathrm{w}}} \tag{2.4a}$$

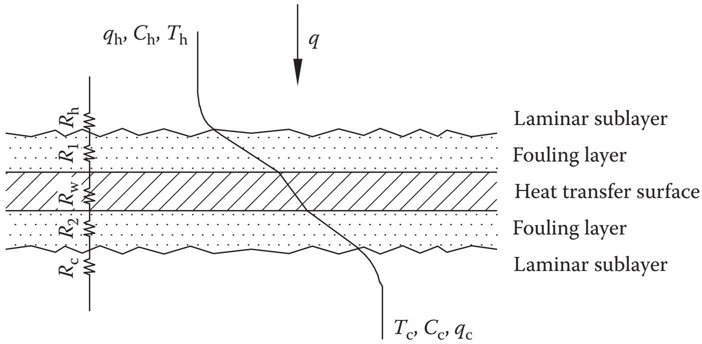


FIGURE 2.1 Elements of thermal resistance of a heat exchanger.

and for a circular wall by

$$R_w = \frac{\ln(d/d_i)}{2\pi k_w L N_t} \tag{2.4b}$$

where

- δ is the wall thickness
- A_w is the total wall area for heat conduction
- k_w is the thermal conductivity of the wall material
- d is the tube outside diameter
- d_i is the tube inside diameter
- L is the tube length
- N_t is the number of tubes, and total wall area for heat conduction is given by

$$A_w = L_1 L_2 L_p \tag{2.5}$$

where

- L_1, L_2, and N_p are the length, width, and total number of separating plates, respectively
- R_2 is the thermal resistance due to fouling on the cold side, given in terms of cold side fouling resistance $R_{f,c}$ by $R_{f,c}/(\eta_o A)_c$
- R_c is the cold side film convection resistance, $1/(\eta_o hA)_c$

In these definitions, h is the heat transfer coefficient on the side under consideration, A represents the total of the primary surface area, A_p, and the secondary (finned) surface area, A_f, on the fluid side under consideration, η_o is the overall surface effectiveness of an extended surface, and the subscripts h and c refer to the hot and cold fluid sides, respectively. The overall surface effectiveness η_o is related to the fin efficiency η_f and the ratio of fin surface area A_f to total surface area A as follows:

$$\eta_o = 1 - \frac{A_f}{A}(1 - \eta_f) \tag{2.6}$$

Note that η_o is the unity for an all prime surface exchanger without fins. Equation 2.3 can be alternately expressed as

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_h} + \frac{R_{f,h}}{(\eta_o A)_h} + R_w + \frac{1}{(\eta_o hA)_c} + \frac{R_{f,c}}{(\eta_o A)_c} \tag{2.7}$$

Since $UA = U_h A_h = U_c A_c$, the overall heat transfer coefficient U as per Equation 2.7 may be defined optionally in terms of either hot fluid surface area or cold fluid surface area. Thus, the option of A_h or A_c must be specified in evaluating U from the product, UA. For plain tubular exchangers, U_o based on tube outside surface is given by

$$\frac{1}{U_o} = \frac{1}{h_o} + R_{f,o} + \frac{d\ln(d/d_i)}{2k_w} + \frac{R_{f,i}d}{d_i} + \frac{d}{h_i d_i} \tag{2.8}$$

The knowledge of wall temperature in a heat exchanger is essential to determine the localized hot spots, freeze points, thermal stresses, local fouling characteristics, or boiling and condensing coefficients. Based on the thermal circuit of Figure 2.1, when R_w is negligible, $T_{w,h} = T_{w,c} = T_w$ is computed from [1,2] as

$$T_w = \frac{T_h + T_c\left[(R_h + R_1)/(R_c + R_2)\right]}{1 + \left[(R_h + R_1)/(R_c + R_2)\right]} \tag{2.9}$$

When $R_1 = R_2 = 0$, Equation 2.9 further simplifies to

$$T_w = \frac{T_h/R_h + T_c/R_c}{1/R_h + 1/R_c} = \frac{(\eta_o hA)_h T_h + (\eta_o hA)_c T_c}{(\eta_o hA)_h + (\eta_o hA)_c} \tag{2.10}$$

2.2 HEAT EXCHANGER HEAT TRANSFER ANALYSIS METHODS

2.2.1 Energy Balance Equation

The first law of thermodynamics must be satisfied in any heat exchanger design procedure at both the macro and microlevels. The overall energy balance for any two-fluid heat exchanger is given by

$$m_h c_{p,h}(t_{h,i} - t_{h,o}) = m_c c_{p,c}(t_{c,o} - t_{c,i}) \tag{2.11}$$

Equation 2.11 satisfies the “macro” energy balance under the usual idealizations made for the basic design theory of heat exchangers [3].

2.2.2 Heat Transfer

For any flow arrangement, heat transfer for two fluid streams is given by

$$q = C_h(t_{h,i} - t_{h,o}) = C_c(t_{c,o} - t_{c,i}) \tag{2.12}$$

and the expression for maximum possible heat transfer rate q_{max} is

$$q_{max} = C_{min}(t_{h,i} - t_{c,i}) \tag{2.13}$$

The maximum possible heat transfer rate would be obtained in a counterflow heat exchanger with very large surface area and zero longitudinal wall heat conduction, and the actual operating conditions are the same as the theoretical conditions.

2.2.3 Basic Methods to Calculate Thermal Effectiveness

There are four design methods to calculate the thermal effectiveness of heat exchangers:

1. ε-NTU method
2. P-NTU$_t$ method
3. LMTD method
4. ψ-P method

The basics of these methods are discussed next. For more details on these methods, refer to Refs. [1,2].

2.2.3.1 ε-NTU Method

The formal introduction of the ε-NTU method for the heat exchanger analysis was in 1942 by London and Seban [4]. In this method, the total heat transfer rate from the hot fluid to the cold fluid in the exchanger is expressed as

$$q = \varepsilon C_{\min}(t_{\text{h,i}} - t_{\text{c,i}}) \tag{2.14}$$

where ε is the heat exchanger effectiveness. It is nondimensional and for a direct transfer type heat exchanger, in general, it is dependent on NTU, C^*, and the flow arrangement:

$$\varepsilon = \phi\,(\text{NTU}, C^*, \text{flow arrangement}) \tag{2.15}$$

These three nondimensional parameters, C^*, NTU, and ε, are defined next.

Heat capacity rate ratio, C^*: This is simply the ratio of the smaller to larger heat capacity rate for the two fluid streams so that $C^* \leq 1$.

$$C^* = \frac{C_{\min}}{C_{\max}} = \frac{(mc_{\text{p}})_{\min}}{(mc_{\text{p}})_{\max}} \tag{2.16}$$

where
C refers to the product of mass and specific heat of the fluid
the subscripts min and max refer to the $C_{\min}$ and $C_{\max}$ sides, respectively

In a two-fluid heat exchanger, one of the streams will usually undergo a greater temperature change than the other. The first stream is said to be the "weak" stream, having a lower thermal capacity rate ($C_{\min}$), and the other with higher thermal capacity rate ($C_{\max}$) is the "strong" stream.

Number of transfer units, *NTU*: NTU designates the nondimensional "heat transfer size" or "thermal size" of the exchanger. It is defined as a ratio of the overall conductance to the smaller heat capacity rate:

$$\text{NTU} = \frac{UA}{C_{\min}} = \frac{1}{C_{\min}}\int_A U dA \tag{2.17}$$

If U is not a constant, the definition of the second equality applies. For constant U, substitution of the expression for UA results in [1,2]

$$\text{NTU} = \frac{1}{C_{\min}}\left[\frac{1}{1/(\eta_{\text{o}}hA)_{\text{h}} + R_1 + R_{\text{w}} + R_2 + 1/(\eta_{\text{o}}hA)_{\text{c}}}\right] \tag{2.18}$$

where R_1 and R_2 are the thermal resistances due to fouling on the hot side and the cold side, respectively, as defined in Equation 2.7. In the absence of the fouling resistances, NTU can be given by the expression

$$\frac{1}{\mathrm{NTU}} = \frac{1}{\mathrm{NTU_h}(C_h/C_{min})} + R_w C_{min} + \frac{1}{\mathrm{NTU_c}(C_c/C_{min})} \tag{2.19}$$

and the number of heat transfer units on the hot and cold sides of the exchanger may be defined as follows:

$$\mathrm{NUT_h} = \frac{(\eta_o hA)_h}{C_h} \quad \mathrm{NTU_c} = \frac{(\eta_o hA)_c}{C_c} \tag{2.20}$$

Heat exchanger effectiveness, ε: Heat exchanger effectiveness, ε, is defined as the ratio of the actual heat transfer rate, q, to the thermodynamically possible maximum heat transfer rate (q_{max}) by the second law of thermodynamics:

$$\varepsilon = \frac{q}{q_{max}} \tag{2.21}$$

The value of ε ranges between 0 and 1. Using the value of actual heat transfer rate q from Equation 2.12 and q_{max} from Equation 2.13, the exchanger effectiveness ε of Equation 2.21 is given by

$$\varepsilon = \frac{C_h(t_{h,i} - t_{h,o})}{C_{min}(t_{h,i} - t_{c,i})} = \frac{C_c(t_{c,o} - t_{c,i})}{C_{min}(t_{h,i} - t_{c,i})}$$
$$\text{For } C^* = 1, \quad \varepsilon_h = \varepsilon_c \tag{2.22}$$

Dependence of ε on NTU: At low NTU, the exchanger effectiveness is generally low. With increasing values of NTU, the exchanger effectiveness generally increases, and in the limit it approaches the maximum asymptotic value. However, there are exceptions such that after reaching a maximum value, the effectiveness decreases with increasing NTU.

2.2.3.2 *P*-NTU$_t$ Method

This method represents a variant of the ε-NTU method. The origin of this method is related to shell and tube exchangers. In the ε-NTU method, one has to keep track of the C_{min} fluid. In order to avoid possible errors, an alternative is to present the temperature effectiveness, P, of the fluid side under consideration as a function of NTU and heat capacity rate of that side to that of the other side, R. Somewhat arbitrarily, the side chosen is the tubeside regardless of whether it is the hot side or the cold side.

General P-NUT$_t$ functional relationship: Similar to the exchanger effectiveness ε, the thermal effectiveness P is a function of NTU$_t$, R, and flow arrangement:

$$P = \phi\,(\mathrm{NTU_t}, R, \text{flow arrangement}) \tag{2.23}$$

where P, NTU$_t$, and R are defined consistently based on the tubeside fluid variables. In this method, the total heat transfer rate from the hot fluid to the cold fluid is expressed by

$$q = PC_t(T_1 - t_1) \tag{2.24}$$

Thermal effectiveness, P: For a shell and tube heat exchanger, the temperature effectiveness of the tubeside fluid, P, is referred to as the "thermal effectiveness." It is defined as the ratio of the temperature rise (drop) of the tubeside fluid (regardless of whether it is hot or cold fluid) to the difference of inlet temperature of the two fluids. According to this definition, P is given by

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad (P \text{ is referred to tubeside}) \tag{2.25}$$

where
t_1 and t_2 refer to tubeside inlet and outlet temperatures, respectively
T_1 and T_2 refer to shellside inlet and outlet temperatures, respectively

Comparing Equations 2.25 and 2.22, it is found that the thermal effectiveness P and the exchanger effectiveness ε are related as

$$\begin{aligned} P &= \frac{C_{\min}}{C_t}\varepsilon = \varepsilon \qquad \text{for } C_t = C_{\min} \\ &= \varepsilon C^* \quad \text{for } C_t = C_{\max} \end{aligned} \tag{2.26}$$

Note that P is always less than or equal to ε. The thermal effectiveness of the shellside fluid can be determined from the tubeside values by the relationship given by

$$\begin{aligned} P_s &= P\frac{C_t}{C_s} = PR \\ \text{For } R^* &= 1, \quad P_s = P \text{ (tubeside)} \end{aligned} \tag{2.27}$$

(For TEMA shell types, the thermal effectiveness charts given in this chapter 2, depicts thermal effectiveness referred to tubeside only)

Heat capacity ratio, R: For a shell and tube exchanger, R is the ratio of the capacity rate of the tube fluid to the shell fluid. This definition gives rise to the following relation in terms of temperature drop (rise) of the shell fluid to the temperature rise (drop) of the tube fluid:

$$R = \frac{C_t}{C_s} = \frac{T_1 - T_2}{t_2 - t_1} \tag{2.28}$$

where the right-hand-side expressions come from an energy balance and indicate the temperature drop/rise ratios. The value of R ranges from zero to infinity, zero being for pure vapor condensation and infinity being for pure liquid evaporation. Comparing Equations 2.28 and 2.16, R and C^* are related by

$$\begin{aligned} R &= \frac{C_t}{C_s} = C^* \quad \text{for } C_t = C_{\min} \\ &= \frac{1}{C^*} \quad \text{for } C_t = C_{\max} \end{aligned} \tag{2.29}$$

Thus R is always greater than or equal to C^*.

Number of transfer units, NTU_t: For a shell and tube exchanger, the number of transfer units NTU_t is defined as a ratio of the overall conductance to the tubeside fluid heat capacity rate:

$$\text{NTU}_t = \frac{UA}{C_t} \tag{2.30}$$

Thus, NTU_t is related to NTU based on C_{min} by

$$\mathrm{NTU_t} = \mathrm{NTU}\frac{C_{min}}{C_t} = \mathrm{NTU} \qquad \text{for } C_t = C_{min}$$
$$= \mathrm{NTU}C^* \quad \text{for } C_t = C_{max} \tag{2.31}$$

Thus NTU_t is always less than or equal to NTU.

2.2.3.3 Log Mean Temperature Difference Correction Factor Method

The maximum driving force for heat transfer is always the log mean temperature difference (LMTD) when two fluid streams are in countercurrent flow. However, the overriding importance of other design factors causes most heat exchangers to be designed in flow patterns different from true countercurrent flow. The true MTD of such flow arrangements will differ from the logarithmic MTD by a certain factor dependent on the flow pattern and the terminal temperatures. This factor is usually designated as the log MTD correction factor, F. The factor F may be defined as the ratio of the true MTD to the logarithmic MTD. The heat transfer rate equation incorporating F is given by

$$q = UA\Delta t_m = UAF\Delta t_{lm} \tag{2.32}$$

where
Δt_m is the true MTD
Δt_{lm} is the LMTD

The expression for LMTD for a counterflow exchanger is given by

$$\mathrm{LMTD} = \Delta t_{lm} = \frac{\Delta t_1 - \Delta t_2}{\ln(\Delta t_1/\Delta t_2)} \tag{2.33a}$$

where $\Delta t_1 = t_{h,i} - t_{c,o} = T_1 - t_2$ and $\Delta t_2 = t_{h,o} - t_{c,i} = T_2 - t_1$ for all flow arrangements except for parallelflow; for parallelflow $\Delta t_1 = t_{h,i} - t_{c,i}(=T_1 - t_1)$ and $\Delta t_2 = t_{h,o} - t_{c,o}(=T_2 - t_2)$. Therefore, LMTD can be represented in terms of the terminal temperatures, that is, greater terminal temperature difference (GTTD or GTD) and smaller terminal temperature difference (STTD or STD) for both pure parallel- and counterflow arrangements. Accordingly, LMTD is given by

$$\mathrm{LMTD} = \Delta t_{lm} = \frac{\mathrm{GTTD} - \mathrm{STTD}}{\ln(\mathrm{GTTD/STTD})} \tag{2.33b}$$

The terminal temperature distribution to calculate LMTD is shown in Figure 2.2a.

2.2.3.3.1 LMTD Correction Factor, F

Charts to determine LMTD from the terminal temperature differences are shown in Figure 2.2b. (Note: While referring the nomogram of Figure 2.2b, use GTD in place of GTTD and STD in place of STTD.)

From its definition, F is expressed by

$$F = \frac{\Delta t_m}{\Delta t_{lm}} \tag{2.34}$$

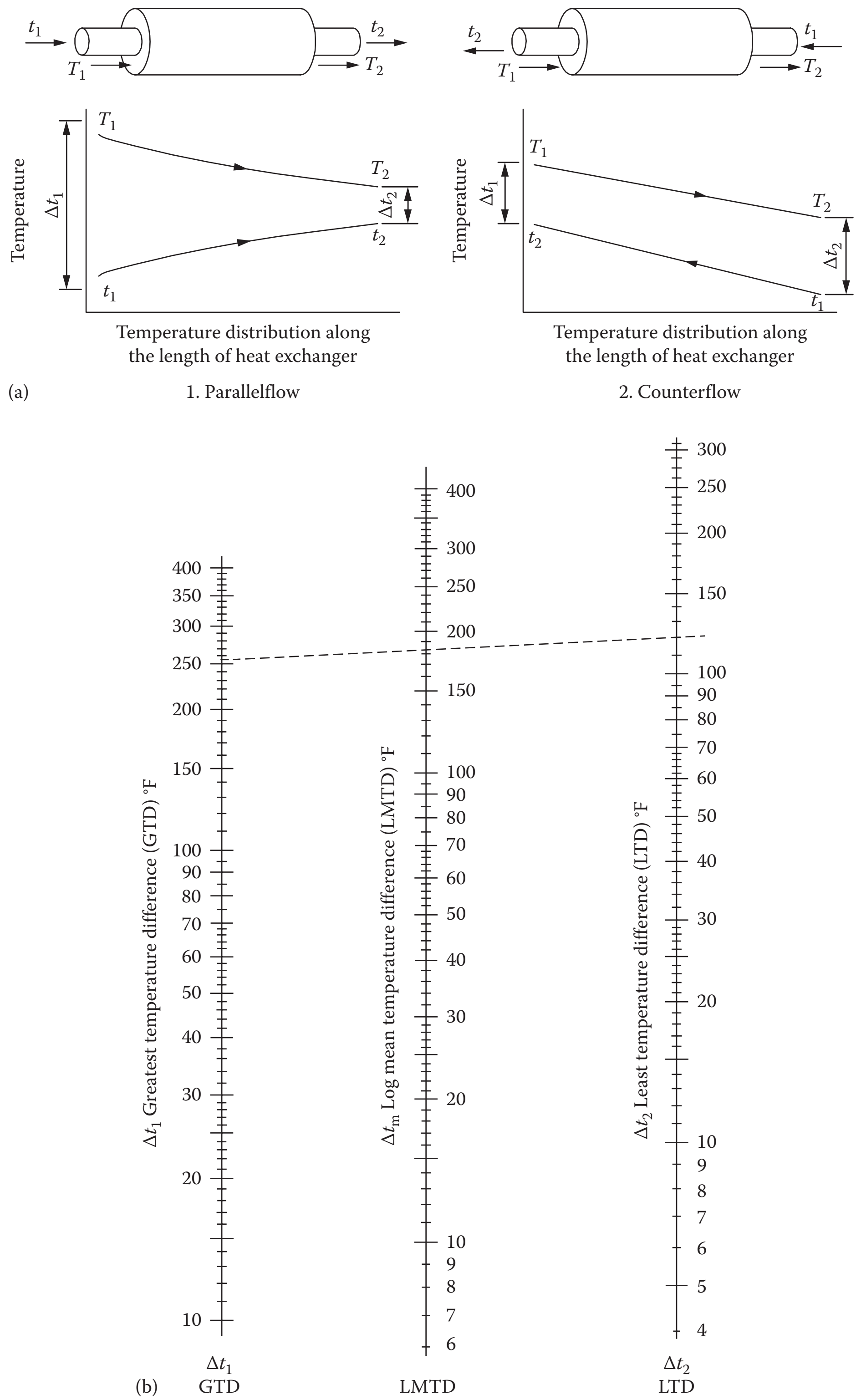


FIGURE 2.2 (a) Terminal temperature to calculate LMTD; (b) nomogram to find LMTD. (Courtesy of Paul-Muller Company, Springfield, MO.)

In situations where the heat release curves are nonlinear, the approach just described is not applicable and a "weighted" temperature difference must be determined.

It can be shown that, in general, F is dependent upon the thermal effectiveness P, the heat capacity rate ratio R, and the flow arrangement. Therefore, F is represented by

$$F = \phi\,(P, R, \mathrm{NTU_t}, \text{flow arrangement}) \tag{2.35}$$

and the expression for F in terms of P, R, and NTU is given by

$$F = \frac{1}{(R-1)\mathrm{NTU}} \ln\left[\frac{1-P}{1-PR}\right] \quad \text{for } R \neq 1$$

$$= \frac{P}{(1-P)\mathrm{NTU}} \quad \text{for } R = 1 \tag{2.36a}$$

$$F = \frac{1}{(1-C^*)\mathrm{NTU}} \ln\left[\frac{1-\varepsilon C^*}{1-\varepsilon}\right] \quad \text{for } C^* \neq 1$$

$$= \frac{\varepsilon}{(1-\varepsilon)\mathrm{NTU}} \quad \text{for } C^* = 1 \tag{2.36b}$$

The factor F is dimensionless.

The value of F is unity for a true counterflow exchanger, and thus independent of P and R. For other arrangements, F is generally less than unity, and can be explicitly presented as a function of P, R, and $\mathrm{NTU_t}$ by Equation 2.36. The value of F close to unity does not mean a highly efficient heat exchanger, but it means a close approach to the counterflow behavior for the comparable operating conditions of flow rates and inlet fluid temperatures. Because of a large capital cost involved with a shell and tube exchanger, generally it is designed in the steep region of the P-$\mathrm{NTU_t}$ curve (ε-NTU relation for the compact heat exchanger) (ε or $P < 60\%$), and as a rule of thumb, the F value selected is 0.80 and higher. However, a better guideline for $F_{\min}$ is provided in the next section. For more details on heat exchanger thermal design methods, refer to Shah and Sekulic [5] and Ref. [6].

2.2.3.3.2 Approximate F *Value for Heat Exchanger Sizing Purpose*

This correction factor accounts for the two streams not in counterflow. At the estimation stage, we do not know the detailed flow and pass arrangement so we can assume the following for preliminary sizing:

- $F = 1.0$ for true counterflow, e.g., double-pipe heat exchanger in counterflow arrangement, *F* shell type of shell and tube heat exchanger
- $F = 0.7$ for crossflow heat exchanger
- $F = 0.7$ for TEMA *E* shell with single pass on both shellside and tubeside
- $F = 0.80$ for E_{1-2} shell and tube heat exchanger (refer Figure 2.28)
- $F = 0.95$ for G_{1-2} (refer Figure 2.34), H_{1-2} shell and tube heat exchanger (refer Figure 2.36)
- $F = 0.79$ for J_{1-2} shell and tube heat exchanger (refer Figure 2.38)
- $F = 0.9$ for multi-pass compact heat exchanger and multiple passes on both shellside and tubeside of TEMA *E* shell
- $F = 1.0$ if one stream is isothermal, $C^* = 0$, $R = 0$ or ∞ (typically boiling or condensation)

Applicability of ε-NTU and LMTD methods: Generally, the ε-NTU method is used for the design of compact heat exchangers. The LMTD method is used for the design of shell and tube heat exchangers. It should be emphasized that either method will yield the identical results within the convergence tolerances specified.

2.2.3.4 ψ-*P* Method

The ψ-*P* method was originally proposed by Smith [7] and modified by Mueller [8]. In this method, a new term ψ is introduced, which is expressed as the ratio of the true MTD to the inlet temperature difference of the two fluids:

$$\psi = \frac{\Delta t_m}{t_{h,i} - t_{c,i}} = \frac{\Delta t_m}{T_1 - t_1} \tag{2.37}$$

and ψ is related to ε and NTU and *P* and NTU_t as

$$\psi = \frac{\varepsilon}{NTU} = \frac{P}{NTU_t} \tag{2.38}$$

and the heat transfer rate is given by

$$q = UA\psi(t_{h,i} - t_{c,i}) \tag{2.39a}$$

$$= UA\psi(T_1 - t_1) \tag{2.39b}$$

Since ψ represents the nondimensional Δt_m, there is no need to compute Δt_{lm} in this method.

Functional relationship between the various thermal design methods: The general functional relationship for the ε-NTU, *P*-NTU_t, LMTD, and ψ-*P* methods is shown in Table 2.1, which has been adapted and modified from Ref. [1], and the relationship between the dimensionless groups of these methods is given in Table 2.2.

Thermal design methods for the design of shell and tube heat exchangers: Any of the four methods (ε-NTU, *P*-NTU_t, LMTD, and ψ-*P*) can be used for shell and tube exchangers.

TABLE 2.1
General Functional Relationship between Dimensionless Groups of the ε-NTU, *P*-NTU_t, and LMTD

Heat Transfer Parameters	ε-NTU Method	*P*-NTU_t Method	LMTD Method
Heat capacity rate ratio	$C^* = \frac{C_{min}}{C_{max}} = \frac{(mc_p)_{min}}{(mc_p)_{max}}$	$R = \frac{C_t}{C_s} = \frac{T_1 - T_2}{t_2 - t_1}$	$LMTD = \frac{\Delta t_1 - \Delta t_2}{\ln\left[\frac{\Delta t_1}{\Delta t_2}\right]}$
NTU	$NTU = \frac{UA}{C_{min}} = \frac{1}{C_{min}}\int_A U dA$	$NTU_t = \frac{UA}{C_t}$	$LMTD = \Delta t_{lm}$ $F = \phi(P, R, NTU_t$, flow arrangement)
Thermal effectiveness	$\varepsilon = \phi(NTU, C^*$, flow arrangement) $\varepsilon = \frac{C_h(t_{h,i} - t_{h,o})}{C_{min}(t_{h,i} - t_{c,i})} = \frac{C_c(t_{c,o} - t_{c,i})}{C_{min}(t_{h,i} - t_{c,i})}$	$P = \phi(NTU_t, R$, flow arrangement) $P = \frac{t_2 - t_1}{T_1 - t_1}$ $P_s = P\frac{C_t}{C_s} = PR$	$F = \frac{\Delta t_m}{\Delta t_{lm}}$ $F = \frac{1}{(R-1)NTU}\ln\left[\frac{1-P}{1-PR}\right]$ for $R \neq 1$ $= \frac{P}{(1-P)NTU}$ for $R = 1$ $q = UA\Delta t_m = UAF\Delta t_{lm}$
Heat transfer	$q = C_h(t_{h,i} - t_{h,o}) = C_c(t_{c,o} - t_{c,i})$	$q = PC_t(T_1 - t_1)$	

TABLE 2.2
Relationship between Dimensionless Groups of the ε-NTU, P-NTU$_t$, and LMTD Methods

$$R = \frac{C_t}{C_s} = C^* \quad \text{for } C_t = C_{min}$$

$$= \frac{1}{C^*} \quad \text{for } C_t = C_{max}$$

$$\text{NTU}_t = \text{NTU}\frac{C_{min}}{C_t} = \text{NTU} \quad \text{for } C_t = C_{min}$$

$$= \text{NTU}C^* \quad \text{for } C_t = C_{max}$$

$$F = \frac{1}{(R-1)\text{NTU}} \ln\left[\frac{1-P}{1-PR}\right] \quad \text{for } R \neq 1$$

$$= \frac{P}{(1-P)\text{NTU}} \quad \text{for } R = 1$$

$$\psi = \frac{\varepsilon}{\text{NTU}} = \frac{P}{\text{NTU}_t}$$

2.2.4 Some Fundamental Relationships to Characterize the Exchanger for "Subdesign" Condition

The partial derivatives of the temperature efficiency P with respect to NTU and R enable complete characterization of the exchanger performance around an operating point. Thus, the exchanger performance can be readily predicted for the "subdesign" conditions [9]. Singh [9] developed derivatives of P, F, and NTU. Derivatives for P and F are discussed next.

Dependence of thermal effectiveness: Thermal performance P and thermal effectiveness ε can be represented through R by [9]

$$\begin{aligned} \varepsilon &= P \quad R \le 1 \\ &= PR \quad R > 1 \end{aligned} \tag{2.40}$$

$$P = f(\text{NTU}, R) \tag{2.41}$$

$$\varepsilon = \phi(NTU, R^*) \tag{2.42}$$

Thus,

$$d\varepsilon = d\phi = \frac{\partial\phi}{\partial\text{NTU}} d\text{NTU} + \frac{\partial\phi}{\partial R} dR \tag{2.43}$$

$$d\varepsilon = dP = \frac{\partial P}{\partial\text{NTU}} d\text{NTU} + \frac{\partial P}{\partial R} dR \quad \text{for } R \le 1 \tag{2.44}$$

$$d\varepsilon = PdR + RdP \tag{2.45}$$

$$= PdR + R\left(\frac{\partial P}{\partial\text{NTU}} d\text{NTU} + \frac{\partial P}{\partial R} dR\right) \tag{2.46}$$

or

$$d\varepsilon = \left(P + R\frac{\partial P}{\partial R}\right)dR + R\frac{\partial P}{\partial \mathrm{NTU}}d\mathrm{NTU} \quad \text{for } R \geq 1 \tag{2.47}$$

Dependence of LMTD correction factor, F: The derivatives of F with respect to ε, P, and R are given by [9]

$$\frac{\partial F}{\partial \mathrm{NTU}} = -\frac{1}{\mathrm{NTU}^2(R-1)}\ln\left[\frac{1-P}{1-PR}\right] = \frac{-F}{\mathrm{NTU}} \tag{2.48}$$

$$\frac{\partial F}{\partial P} = \frac{1}{\mathrm{NTU}(1-P)(1-PR)} \tag{2.49}$$

$$\frac{\partial F}{\partial R} = \frac{-F}{(R-1)} + \frac{1}{\mathrm{NTU}(R-1)(1-PR)} \tag{2.50}$$

and

$$dF = \frac{\partial F}{\partial P}dP + \frac{\partial F}{\partial \mathrm{NTU}}d\mathrm{NTU} + \frac{\partial F}{\partial R}dR \tag{2.51}$$

2.3 THERMAL EFFECTIVENESS CHARTS

Broadly speaking, there are two types of heat exchanger problems: rating and sizing. To solve either type of problem from first principles is laborious and time-consuming. However, sizing and rating of heat exchangers are solved with the use of performance charts easily. The graphical charts were introduced many years ago and have gained wide acceptance throughout the industry. Five types of heat exchanger design charts are found in the literature, and the salient features of these charts are discussed by Turton et al. [10]. These charts are shown schematically in Figure 2.3a–f. The dimensionless variables used in these charts (ε, P, R, C^*, F, NTU, $\mathrm{NTU_t}$) have been defined in Section 2.2.

Figure 2.3a is the most widely used of these charts and was introduced by Bowman et al. [11] in 1940. In this chart, the LMTD correction factor, F, is presented as a function of the effectiveness, P, and the heat capacity rate ratio, R. Using this chart, the design problem where terminal temperatures and flow rates are usually specified but overall U and/or A are unknown can be solved; however, the rating problem can be solved by a trial-and-error solution. Since F compares the true MTD of a given flow arrangement with that of the counterflow arrangement, these charts provide a well-suited means of finding out the best of several possible flow arrangements. The one with the higher F will require the lower NTU, that is, the lower area if U remains constant, operating with the same R and P. Underwood [12] first derived the expression for true MTD for E_{1-2}, E_{1-4}, and E_{2-4} shell and tube exchangers in 1934. Bowman et al. [11] published a summary of correction factors for exchangers of different flow arrangements. Ten Broeck [13] further constructed charts using dimensionless groups, $UA/(mc_\mathrm{p})_\mathrm{t}$, $P = (t_2 - t_1)/(T_1 - t_1)$, and $R = (T_1 - T_2)/(t_2 - t_1)$ for direct calculation of terminal temperatures with known surface area of a heat exchanger. At present, F charts are available for all TEMA shells.

Figure 2.3b and c are due to Kays and London [14] and TEMA [15], respectively. Figure 2.3c is plotted on a semilog paper, since the most useful NTU and $\mathrm{NTU_t}$ design range for compact heat exchangers and shell and tube exchangers, respectively, is 0.2–3.0. A careful look at the linear graphical presentation of the ε-NTU results of Figure 2.3b indicates that the NTU scale in this range is short and hence one cannot obtain the accurate values of ε or NTU from graphs.

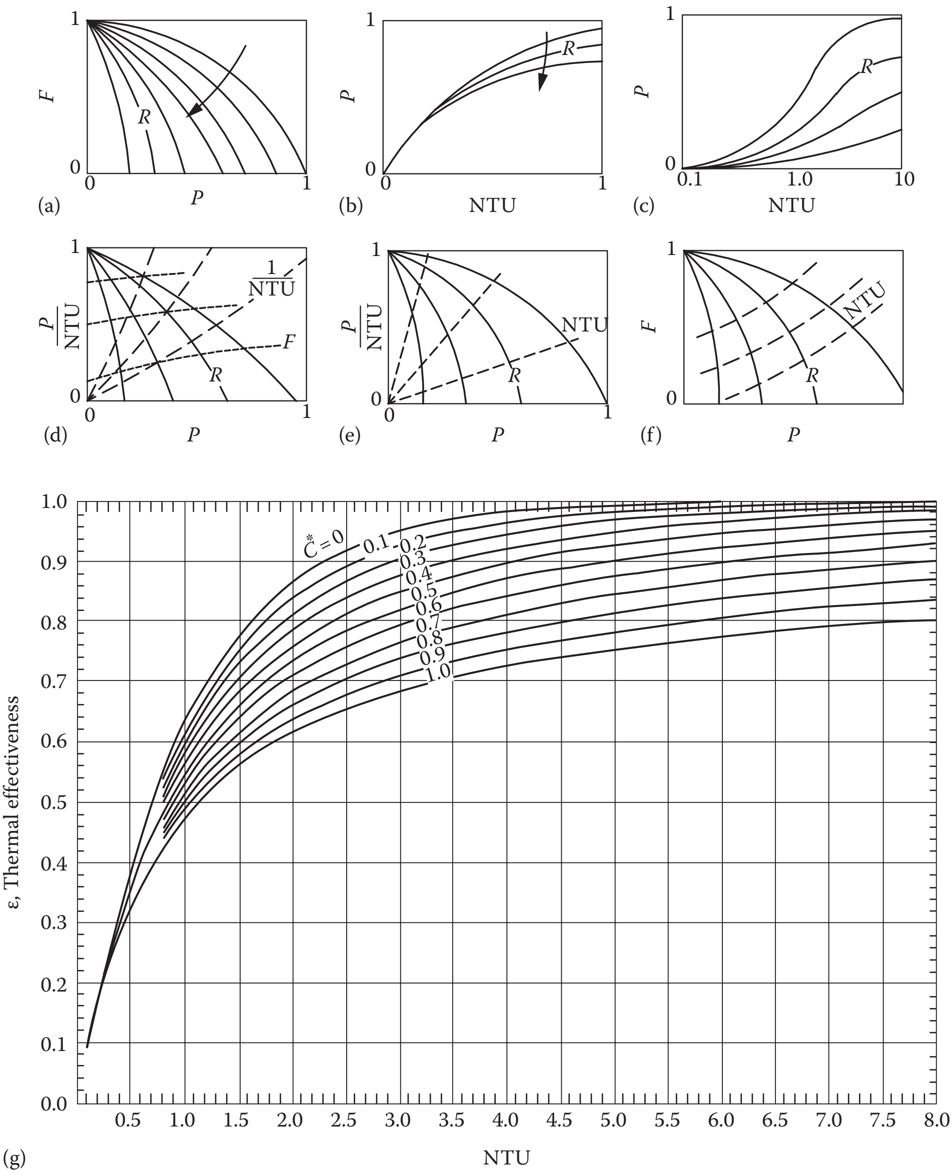


FIGURE 2.3 Thermal effectiveness charts. (a) Bowman chart; (b) Kays and London chart; (c) TEMA chart; (d) *F–P–R*-chart; (e) ψ chart; (f) *F-P-R*-NTU chart (From Turton, R. et al., *Trans. ASME J. Heat Transfer*, 106, 893, 1984); (g) ε-NTU chart for unmixed–unmixed crossflow, as per Eq. T4 of Table 2.4.

For better appreciation, this is illustrated through the thermal relation chart (ε-NTU) for crossflow heat exchanger in Figure 2.3. An alternative is to stretch the NTU scale in the range 0.2–3.0 by using a logarithmic scale. Thus, the P-$\mathrm{NTU_t}$ results are generally presented on a semilog paper, as shown for example in Figure 2.3c, in order to obtain more accurate graphical values of P or $\mathrm{NTU_t}$. Using these charts, both the sizing and rating problem can be solved. However, the LMTD correction factor F is not shown in these charts. Hence, it is to be calculated additionally.

Muller [8] proposed the charts of Figure 2.3d with its triple family of curves. This chart can be used to solve both the sizing and rating problems and in addition gives the F values. However, Figure 2.3d is somewhat cramped and difficult to read accurately and introduces yet another parameter,

P/NTU$_t$. The Muller charts have been redrawn recently by Taborek and included in HEDH [16]. The present form of this chart is shown in Figure 2.3e. The main difference between Figure 2.3d and e is that the F parameter curves have been omitted in the latter, and thus the problem of having to separately calculate the F values has been retained.

In a system with four variables, F, P, R, and NTU or NTU$_t$, any chart displays just one family of curves, such as Figure 2.3a–c, and does not give all the interrelationships directly. On the other hand, a chart with three families of curves, as in Figure 2.3d, has one set that is redundant. To show all the interrelationships between these four variables requires a chart with two families of curves. This is satisfied by Figure 2.3e.

In the graphical presentation, ψ is plotted against P and R as a parameter as shown in Figure 2.3e. The lines of constant R originate at $\psi = 1$ and terminate at $\psi = 0$ so that the asymptotic values of P for NTU tend to infinity. Thus the curves of constant R are similar to those for the F-P charts. In order to tie in with the P-NTU$_t$ and LMTD methods, the lines of constant NTU$_t$ and constant F are also superimposed on this chart. Figure 2.3e also has one limitation: It does not show directly the four parameters of interest.

Constraints due to the charts in Figure 2.3a–e are overcome by a chart, shown in Figure 2.3f, proposed by Turton et al. [10]. The chart in Figure 2.3f extends the easy-to-read Bowman charts of Figure 2.3a to include a second family of curves representing the variable NTU. Both the sizing and rating problems can be solved using this form of chart, and F values can be found directly for both types of problems. Thus to find exchanger surface area, use P and R to evaluate F and NTU. To find terminal temperature, use NTU and R to evaluate P and F. Most of the charts included in this book are of the type of Figure 2.3f.

2.4 SYMMETRY PROPERTY AND FLOW REVERSIBILITY AND RELATION BETWEEN THE THERMAL EFFECTIVENESS OF OVERALL PARALLEL AND COUNTERFLOW HEAT EXCHANGER GEOMETRIES

2.4.1 Symmetry Property

The symmetry property relates the thermal behavior of a heat exchange process to that of the reverse process, in which the directions of flow of both fluids are reversed [17]. Figure 2.4 shows four different flow arrangements for the TEMA E_{1-2} shell and tube heat exchanger that are equivalent if complete transverse mixing of the shell fluid is satisfied.

2.4.2 Flow Reversibility

Flow reversibility establishes a relation between the thermal effectiveness of two heat exchanger configurations that differ from each other in the inversion of either one of the two fluids [18].

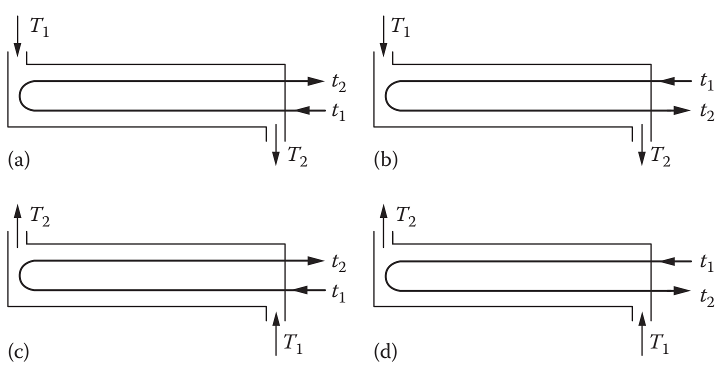


FIGURE 2.4 Flow reversibility principle. (a) Basic $E_{1\text{-}2}$ case, (b) basic case with tube fluid reversed, (c) basic case with shell fluid reversed, and (d) basic case with both shell and tube fluids reversed. (Symmetry operations performed on the TEMA E_{1-2} shell.) (From Pignotti, A., *Trans. ASME*, *J. Heat Transfer*, 106, 361, 1984.)

Although the inversion of both fluids often does not alter the configuration, the inversion of only one of them usually leads from one configuration to an entirely different one, as is the case in going from a pure parallelflow to a pure counterflow arrangement or vice versa. Using this relation, if the expression for the effectiveness, P, of a configuration as a function of the heat capacity rate ratio, R (or C^*), and the number of heat transfer units NTU is known, the corresponding expression for the "inverse" configuration is immediately obtained from the simple relation [18]:

$$P_{\mathrm{i}}(R,\mathrm{NTU}) = \frac{P(-R,\mathrm{NTU})}{1+RP(-R,\mathrm{NTU})} \tag{2.52}$$

where P denotes the effectiveness of a given arrangement, and P_{i}, that of the same one with fluid direction reversed. The relation is valid under the assumptions of temperature independence of the heat transfer coefficient and heat capacity rates, when one of the fluids proceeds through the exchanger in a single, mixed stream. In some cases with special symmetry, the inversion of both fluids does not alter the geometry, and therefore this property is trivially satisfied. Pignotti [18] illustrates the property of flow reversibility with several examples from the available literature. An example to clarify the meaning of Equation 2.52 is given next. Consider the well-known expression for the effectiveness of a parallelflow configuration:

$$P(R,\mathrm{NTU}) = \frac{\left[1-\exp\left[-\mathrm{NTU}(1+R)\right]\right]}{(1+R)} \tag{2.53}$$

Let us derive from it the expression for the effectiveness of a pure counterflow configuration, which we denote $P_{\mathrm{c}}(R, \mathrm{NTU})$. Equation 2.52 is applicable, because the counterflow geometry is obtained from parallelflow by inverting the direction of flow of one of the fluids, and the condition that at least one of the fluids should be mixed throughout the exchanger is satisfied. After replacing R by $-R$ in Equation 2.53 and performing the elementary algebraic operations indicated in Equation 2.52, we obtain the expression for the effectiveness of the counterflow configuration:

$$P_{\mathrm{c}}(R,\mathrm{NTU}) = \frac{\left\{1-\exp\left[-\mathrm{NTU}(1-R)\right]\right\}}{\left\{1-R\exp\left[-\mathrm{NTU}(1-R)\right]\right\}} \tag{2.54}$$

Observe also that the inversion of one fluid leads from a parallelflow connection to a counterflow one, and likewise, from the latter to the former; therefore, Equation 2.52 can be used to go from parallelflow to counterflow and vice versa.

The transformation property of Equation 2.52 can also be expressed in terms of the variables referred to the mixed fluid. For example, if the thermal relation on the shellside or tubeside is known in terms of P_x, R_x, and NTU_x, the thermal relation for the other side P_y, R_y, and NTU_y may be obtained from the relation

$$P_y = R_x P_x, \quad R_y = \frac{1}{R_x}, \quad \mathrm{NTU}_y = R_x \mathrm{NTU}_x \tag{2.55}$$

For example, let the tubeside values of an $H_{1\text{–}2}$ exchanger be $P = 0.752$, $R = 0.7$, and NTU = 2.5. Then the shellside values will be $P = 0.7 \times 0.752$, $R = 1/0.7$, and NTU = 0.7×2.5. For $R = 1.0$, both the tubeside and shellside values are the same.

When the thermal effectiveness is the same for the original case and the inverted case, it is referred to as stream symmetric. Typical examples for stream symmetric are parallelflow, counterflow, and crossflow unmixed–unmixed and mixed–mixed cases.

2.5 TEMPERATURE APPROACH, TEMPERATURE MEET, AND TEMPERATURE CROSS

The meanings of temperature approach, temperature meet, and temperature cross are as follows. Temperature approach is the difference of the hotside and coldside fluid temperature at any point of a given exchanger. In a counterflow exchanger or a multipass exchanger, (1) if the cold fluid outlet temperature $t_{c.o}$ is less than the hot fluid outlet temperature $t_{h.o}$, then this condition is referred to as temperature approach; (2) if $t_{c.o} = t_{h,o}$, this condition is referred to as temperature meet; and (3) if $t_{c.o}$ is greater than $t_{h.o}$, the difference $(t_{c.o} - t_{h.o})$ is referred to as the temperature cross or temperature pinch. In this case, the temperature approach $(t_{h.o} - t_{c.o})$ is negative and loses its meaning. Temperature cross indicates a negative driving force for heat transfer between the fluids. It requires either a large area for heat transfer or the fluid velocity to increase overall heat transfer coefficient. The underlying meanings of these three cases are brought out in Table 2.3 and the same are shown in Figure 2.5a.

The temperature cross is undesirable, particularly for shell and tube exchangers, because the tube surface area is not utilized effectively and hence there is wastage of capital cost. If outlet

TABLE 2.3
Temperature Approach, Temperature Meet, and Temperature Cross

Temperature Approach	Temperature Meet	Temperature Cross
$t_{h,i} \rightarrow t_{h,o}$	$t_{h,i} \rightarrow t_{h,o}$	$t_{h,i} \rightarrow t_{h,o}$
$t_{c,o} \leftarrow t_{c,i}$	$t_{c,o} \leftarrow t_{c,i}$	$t_{c,o} \leftarrow t_{c,i}$
$t_{c,o} < t_{h,o}$	$t_{c,o} = t_{h,o}$	$t_{c,o} > t_{h,o}$

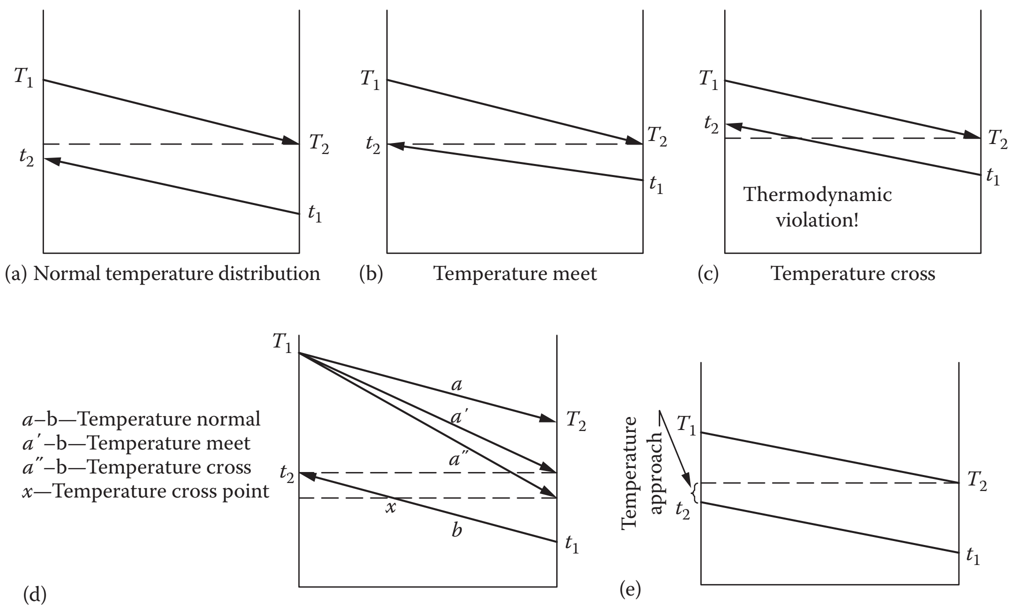


FIGURE 2.5 Principle of temperature approach, temperature meet, and temperature cross. (a) Normal temperature distribution; (b) temperature meet; (c) temperature cross; (d) temperature approach, meet, and cross superimposed; (e) temperature distribution in an E_{1-2} exchanger without temperature cross;

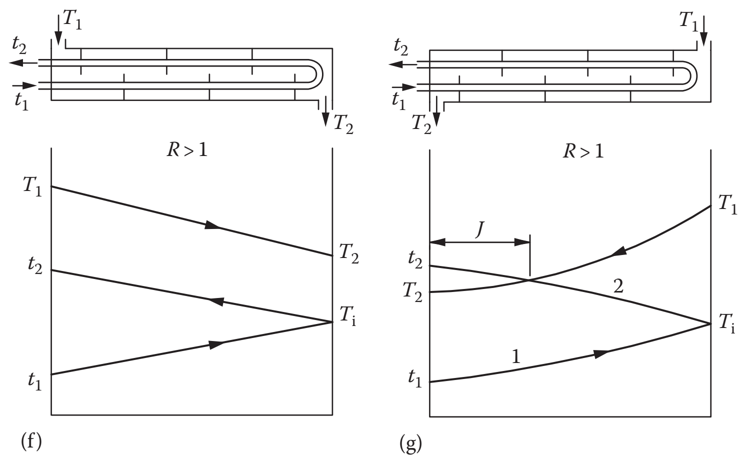


FIGURE 2.5 (continued) Principle of temperature approach, temperature meet, and temperature cross. (f) with temperature cross.

temperatures form a temperature cross in a multiple tube pass heat exchanger, a lower than desirable LMTD correction factor will occur. A simple way to avoid this is to use more exchanger shells in series. Other engineers suggest that a small temperature cross may be acceptable and may provide a less expensive design than the more complex alternatives. For a E_{1-2} heat exchanger, the temperature cross occurs around a relatively narrow range of F value about 0.78–0.82. Lower values of F may be taken as an indication that temperature cross will occur.

The concept of the temperature cross or meet at the exchanger outlet can be utilized to determine the number of shells in series required to meet the heat duty without having a temperature cross in any individual shell. Temperature cross is undesirable for shell and tube heat exchanger because the tube surface area is not utilized cost-effectively. An optimum design would mean that the temperature cross or meet point lies just at the end of the second tube pass. This phenomenon is explained in detail by Shah [2] and is briefly dealt with here with reference to an E_{1-2} exchanger.

For $E_{1\text{-}2}$ two possible shell fluid directions with respect to the tube fluid direction are shown in Figure 2.5b. The temperature distributions of Figure 2.5b reveal that there is a temperature cross. In region X, the second tube pass transfers heat to the shell fluid. This is contrary to the design objective, in which ideally the heat transfer should have taken place only in one direction (from the shell fluid to the tube fluid, as shown in Figure 2.5a) throughout the two passes. The reason for this temperature cross is as follows: Although an addition of surface area (a high value of NTU_t, or a low value of LMTD correction factor F) is effective in raising the temperature of the tube fluid and rises in the second pass up to point X, beyond this point the temperature of the shell fluid is lower than that of the tube fluid, since we have considered the shell fluid mixed at a cross section and it is cooled rapidly by the first pass. Thus, the addition of the surface area in the second tube pass left of point X is useless from the thermal design point of view. A “good” design avoids the temperature cross in a shell and tube exchanger. Theoretically, the optimum design would have the temperature cross point just at the end of the second tube pass, which will satisfy the following condition:

$$t_{t,o} = t_{s,o} \quad \text{or} \quad t_{t,o} - t_{s,o} = 0 \tag{2.56}$$

This condition leads to the following formula:

$$P = \frac{1}{1+R} \tag{2.57}$$

Thus for a given R, Equation 2.57 provides the limiting (maximum) value of P. Corresponding to P and R, the limiting (maximum) value of NTU_t beyond which there will be a temperature cross can be determined from its thermal relation formula. Therefore, from P, R, and NTU, F can be calculated. This F value is known as the F_{min} value beyond which there will be a temperature cross. This is illustrated for an E_{1-2} exchanger here. For a known value of R, determine the limiting value of P from Equation 2.57 and NTU from the following equation:

$$\mathrm{NTU}_{E_{1-2}} = \frac{1}{(1+R^2)^{0.5}} \ln\left[\frac{2-P\left[R+1-(1+R^2)^{0.5}\right]}{2-P\left[R+1+(1+R^2)^{0.5}\right]}\right] \tag{2.58}$$

For known values of P, R, and NTU, determine F from Equation 2.36.

2.5.1 Temperature Cross for Other TEMA Shells

Temperature cross for other TEMA shells such as G_{1-2}, H_{1-2}, and J_{1-2} can be evaluated from Equation 2.57 [19]. The F_{min} curves for G_{1-2}, H_{1-2}, and J_{1-2} cases are given in the next section.

2.6 THERMAL RELATION FORMULAS FOR VARIOUS FLOW ARRANGEMENTS AND PASS ARRANGEMENTS

The heat exchanger effectiveness is defined as the ratio of the overall temperature drop of the weaker stream to the maximum possible temperature difference between the fluid inlet temperatures. The following assumptions are commonly made in deriving thermal effectiveness:

1. The overall heat transfer coefficient is constant throughout the exchanger.
2. Each pass has the same heat transfer area; that is, unsymmetrical pass arrangements are not considered.
3. There is no phase change.
4. The specific heat of each fluid is constant and independent of temperature.
5. The flow rates of both streams are steady.
6. The flow of both fluids is evenly distributed over both the local and the total transfer area.
7. Heat losses from the system are negligible.

In this section, thermal relation formulas for (1) various flow arrangements—parallelflow, counterflow, and crossflow—(2) various types of heat exchangers—compact and shell and tube—and (3) multipass arrangements or multiple units of both compact and shell and tube heat exchangers are presented. Most of the formulas are tabulated and the thermal effectiveness charts are given. Mostly counterflow arrangements are considered. For shell and tube exchangers, formulas are given for both parallelflow and counterflow, but thermal effectiveness charts are given only for counterflow arrangements referred to tubeside (similar to TEMA Standards [15]). For stream symmetric cases, thermal effectiveness relations referred to the shellside can be derived from the "flow reversibility" principle. From counterflow thermal effectiveness relations, thermal effectiveness relations for parallelflow arrangements can be easily derived (for stream symmetric cases only) from the "flow reversibility" principle. Customarily, the ε-NTU method is employed for compact heat exchangers. In this method, the capacity ratio C^* is always ≤ 1. Hence, thermal effectiveness charts are given in terms of ε–C^*–NTU, and wherever possible, the thermal effectiveness charts are also given in terms of P–R–F–NTU, instead of ε–C^*–NTU.

2.6.1 Parallelflow

For a given set of values of C^* or R, and NTU, (1) the thermal effectiveness is much lower for parallelflow than for counterflow arrangement, except in the limiting case $C^* = R = 0$, where it is the same for both cases and approaches unity as NTU increases to infinity, and (2) at a given value of NTU, the effectiveness increases with decreasing capacity ratio, C^* or R. The formula for thermal effectiveness is given by Equation T1 in Table 2.4, and the thermal effectiveness chart is given in Figure 2.6.

2.6.2 Counterflow

Among the various flow arrangements, counterflow has the highest thermal effectiveness. For counterflow exchangers, at a given value of NTU, the effectiveness increases with decreasing capacity ratio, C^* or R. The formula for thermal effectiveness is given by Equation T2, and the thermal effectiveness chart is given in Figure 2.7.

2.6.3 Crossflow Arrangement

2.6.3.1 Unmixed–Unmixed Crossflow

This is an industrially important arrangement representing the case of a large number of unmixed channels in both sides. The original solution was due to Nusselt [20] and was later reformulated into a more manageable equation by Mason [21]. Mason's formula is given by Equation T3, and this equation can be used for P–NTU–R relation. Baclic [22] presents Nusselt's equation in terms of a modified Bessel function of the first kind as given in Equation T4; Eckert [23] provides a simplified formula without involving Bessel function as given by Equation T5, and this equation predicts ε within ±1% of ε from Equation T4 for $1 < \text{NTU} < 7$; Equations T4 and T5 can be used for formulas involving $C^* \leq 1$ only. The thermal effectiveness chart as per Equation T3 is given in Figure 2.8 and as per Equation T4 is given in Figure 2.9.

2.6.3.2 Unmixed–Mixed Crossflow

In this arrangement, one fluid is mixed and the other is unmixed. A typical example is a bare tube compact heat exchanger in which the fluid outside the tube is mixed, whereas the tubeside fluid is unmixed. There are two possible cases: (1) weaker fluid (C_{min}) is mixed and (2) stronger fluid (C_{max}) is mixed. Formulas for thermal effectiveness for the weaker fluid mixed are given by Equation T6 and for the stronger fluid mixed by Equation T7. The thermal effectiveness charts are given in Figure 2.10 for the weaker fluid mixed and Figure 2.11 for the stronger fluid mixed. For $R = 1$ or $C^* = 1$, the thermal effectiveness is the same for both cases.

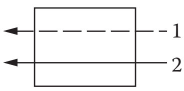
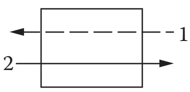
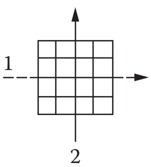
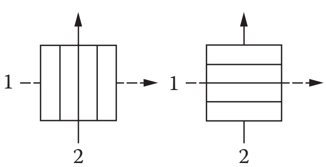
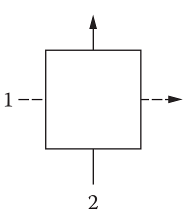
2.6.3.3 Mixed–Mixed Crossflow

This case has no industrial application and is shown here only as an extreme of the crossflow arrangement. The solution is identical to the TEMA *J* shell with infinite tubeside passes. The formula for thermal effectiveness is given by Equation T8.

2.6.3.4 Single or Multiple Rows in Crossflow

Many process heat exchangers provide a crossflow arrangement between the hot (or cold) process fluid that flows through the tubes and the external coolant (or hot air such as supercharged engine intake air), usually air. Because this flow arrangement is not strictly countercurrent, the MTD must be corrected by applying a correction factor, F. The factor F depends on the terminal temperatures, the number of tube rows per pass, and the number of passes. The basic unmixed–unmixed case shown in Figure 2.12 assumes a large number of flow channels in both streams. For a single tubeside pass with one or more tube rows, the thermal effectiveness formula is different from that of the basic unmixed–unmixed case. Thermal relations for single-pass tube rows arrangements are discussed next.

TABLE 2.4
Thermal Effectiveness Relations for Basic Cases

Flow Arrangement	Equation No./ Reference	General Formula	Value for R = 1 and Special Cases
 Parallelflow; stream symmetric	T1	$P = \frac{1-e^{-\mathrm{NTU}(1+R)}}{1+R}$	$P = \frac{1}{2}[1-e^{(-2\mathrm{NTU})}]$ for $R=1$ $= 1-e^{-\mathrm{NTO}}$ for $R=0$ $P_{\max} = 50\%$ for $R=1$
 Counterflow; stream symmetric	T2	$P = \frac{1-e^{-\mathrm{NTU}(1-R)}}{1-Re^{-\mathrm{NTU}(1-R)}}$	$P = \frac{\mathrm{NTU}}{1+\mathrm{NTU}}$ for $R=1$ $= 1-e^{-\mathrm{NTU}}$ for $R=0$
	T3 [21]	$P = \frac{1}{R\mathrm{NTU}}\sum_{k=0}^{\infty}\left\{\left[1-e^{-\mathrm{NTU}}\sum_{m=0}^{k}\frac{\mathrm{NTU}^m}{m!}\right]\left[1-e^{-R\mathrm{NTU}}\sum_{m=0}^{k}\frac{(\mathrm{RNTU})^m}{m!}\right]\right\}$ For $R=1$, this equation holds.	
	T4 [22]	$\varepsilon = 1-e^{[-(1+C^*)\mathrm{NTU}]}\left[I_0(2\mathrm{NTU}\sqrt{C^*})+\sqrt{C^*}I_1(2\mathrm{NTU}\sqrt{C^*}) - \frac{1-C^*}{C^*}\sum_{\mathrm{n}=2}^{\infty}C^{*\mathrm{n}/2}\,I_{\mathrm{n}}(2\,\mathrm{NTU}\sqrt{C^*})\right]$	
Crossflow; both the fluids unmixed; stream symmetric		For $C^* = 1$. $\varepsilon = 1-e^{-2\mathrm{NTU}}\left[I_0(2\mathrm{NTU})+I_1(2\mathrm{NTU})\right]$	
	T5 [23]	$\varepsilon = 1-\exp\left\{\frac{\mathrm{NTU}^{0.22}}{C^*}[\exp(-C^*\mathrm{NTU}^{0.78})-1]\right\}$	
Crossflow; one fluid mixed and the other fluid unmixed (1) weaker ($C_{\min}$) fluid mixed; (2) stronger ($C_{\max}$) fluid mixed	T6	Weaker ($C_{\min}$) fluid mixed $P_1 = [1 - \exp(-K/R)]$ $K = 1 - \exp(-R\mathrm{NTU})$	$P_1 = 1-e^{-(1-e^{-\mathrm{NTU}})}$
	T7	Stronger ($C_{\max}$) fluid mixed $P_1 = \frac{[1-\exp(-KR)]}{R}$ $K = 1 - \exp(-\mathrm{NTU})$	$P_1 = 1-e^{-(1-e^{-\mathrm{NTU}})}$
 Crossflow; mixed–mixed flow; stream symmetric same as $J_{1-\infty}$	T8	$P = \frac{1}{\left(\frac{1}{K_1}+\frac{R}{K_2}-\frac{1}{\mathrm{NTU}}\right)}$ $K_1 = 1 - e^{(-\mathrm{NTU})}$ $K_2 = 1 - e^{(-R\mathrm{NTU})}$	$P = \frac{1}{\frac{2}{K_1}-\frac{1}{\mathrm{NTU}}}$

Note: P_2 can be found using Equation 2.55;
I_0, I_1 and I_n are modified Bessel functions of the first kind.

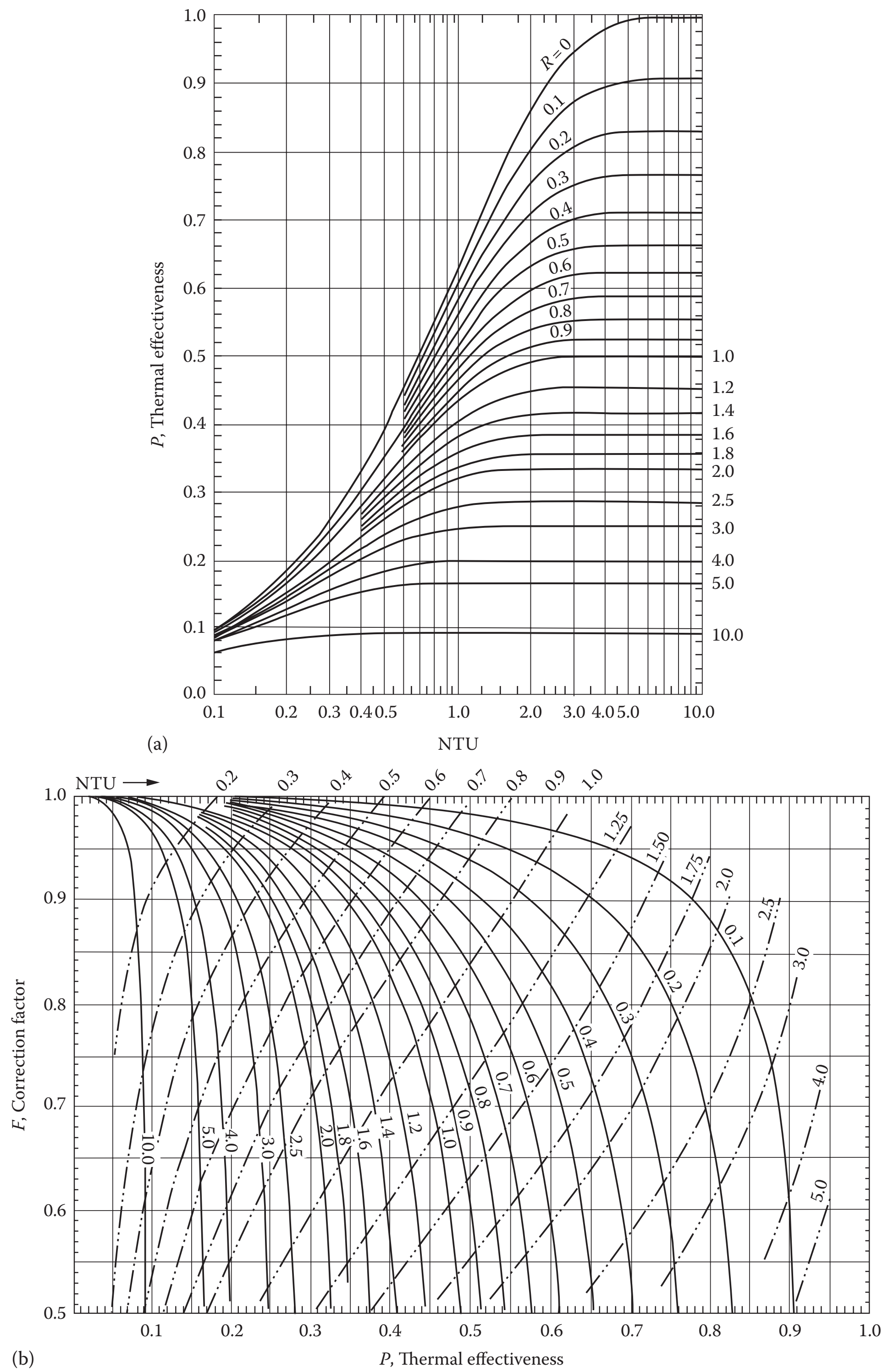


FIGURE 2.6 (a) Thermal effectiveness chart—parallelflow; stream symmetric, R–P–NTU chart (as per Equation T1, Table 2.4); (b) parallelflow; stream symmetric, F–R–P–NTU chart; F as a function of P for constant R (solid lines) and constant NTU (dashed lines) (Equation T1, Table 2.4).

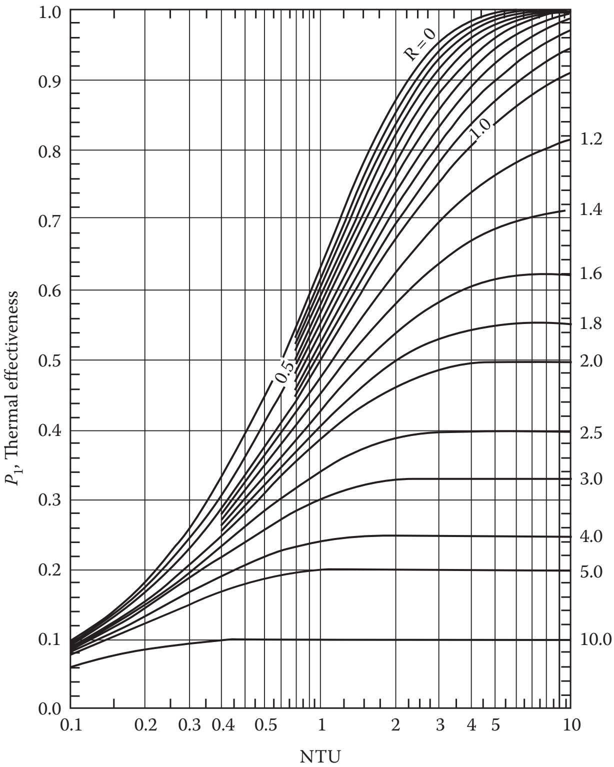


FIGURE 2.7 Thermal effectiveness chart—counterflow; stream symmetric, *R*–*P*–NTU chart (as per Equation T2, Table 2.4).

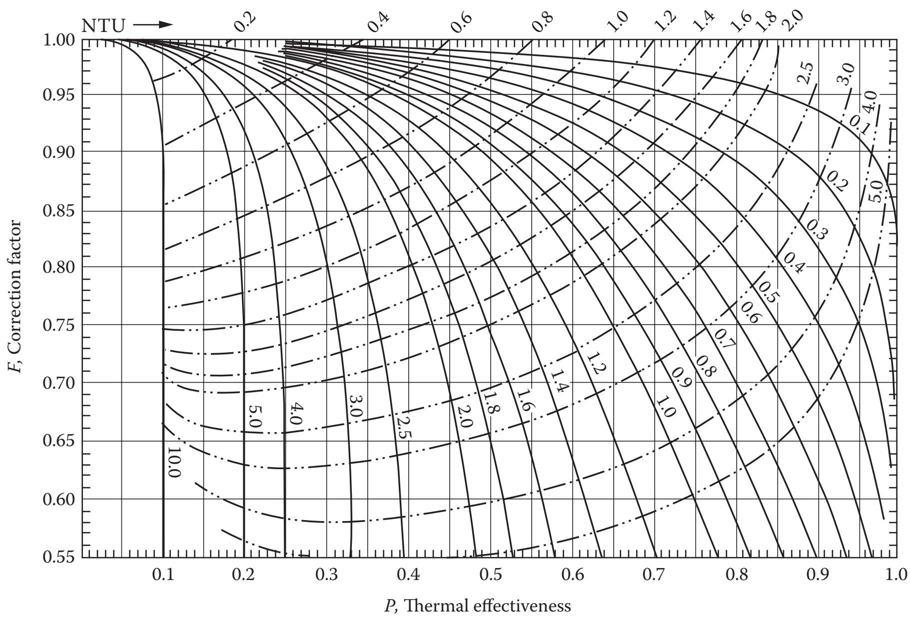


FIGURE 2.8 Thermal effectiveness chart—crossflow; both the fluids unmixed; stream symmetric; *F*–*R*–*P*–NTU chart; *F* as a function of *P* for constant *R* (solid lines) and constant NTU (dashed lines) (as per Equation T3, Table 2.4).

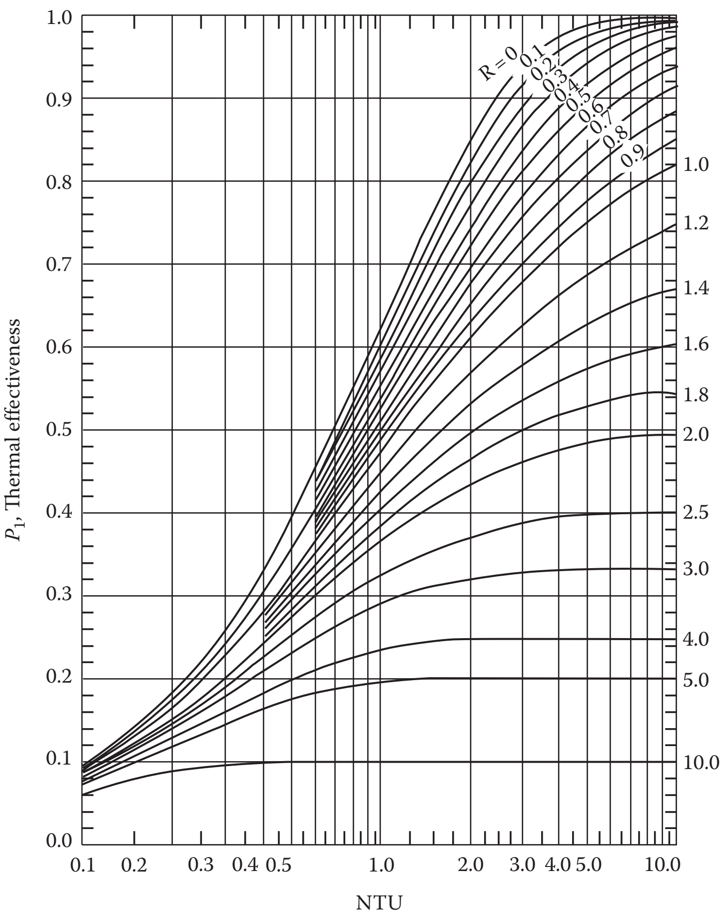


FIGURE 2.9 Thermal effectiveness chart—crossflow; both the fluids unmixed; stream symmetric; *R–P–*NTU chart (as per Equation T4, Table 2.4).

2.6.3.4.1 *Single Tubeside Pass, N Rows per Pass, Both Fluids Unmixed*

A common header at one end of the tubes distributes the tubeside fluid into a single pass having N rows in parallel. A similar header at the other end collects tubeside fluid. For given terminal temperatures, F increases with the number of rows per pass and the number of passes being increased and is more sensitive to the latter. Taborek [24], Pignotti and Cordero [25], and Pignotti [26] present values of F for a variety of crossflow configurations, applicable to air-cooled heat exchangers.

Schedwill's formula for the thermal effectiveness of N rows is given by [27]

$$P = \frac{1}{R}\left\{1 - \left[\frac{Ne^{\mathrm{NKR}}}{1 + \sum_{i=1}^{N-1} \sum_{j=0}^{i} \binom{i}{j} K^j e^{-(i-j)\mathrm{NTU}/N} \sum_{k=0}^{j} \frac{(\mathrm{NKR})^k}{k!}}\right]\right\}^{-1} \tag{2.59}$$

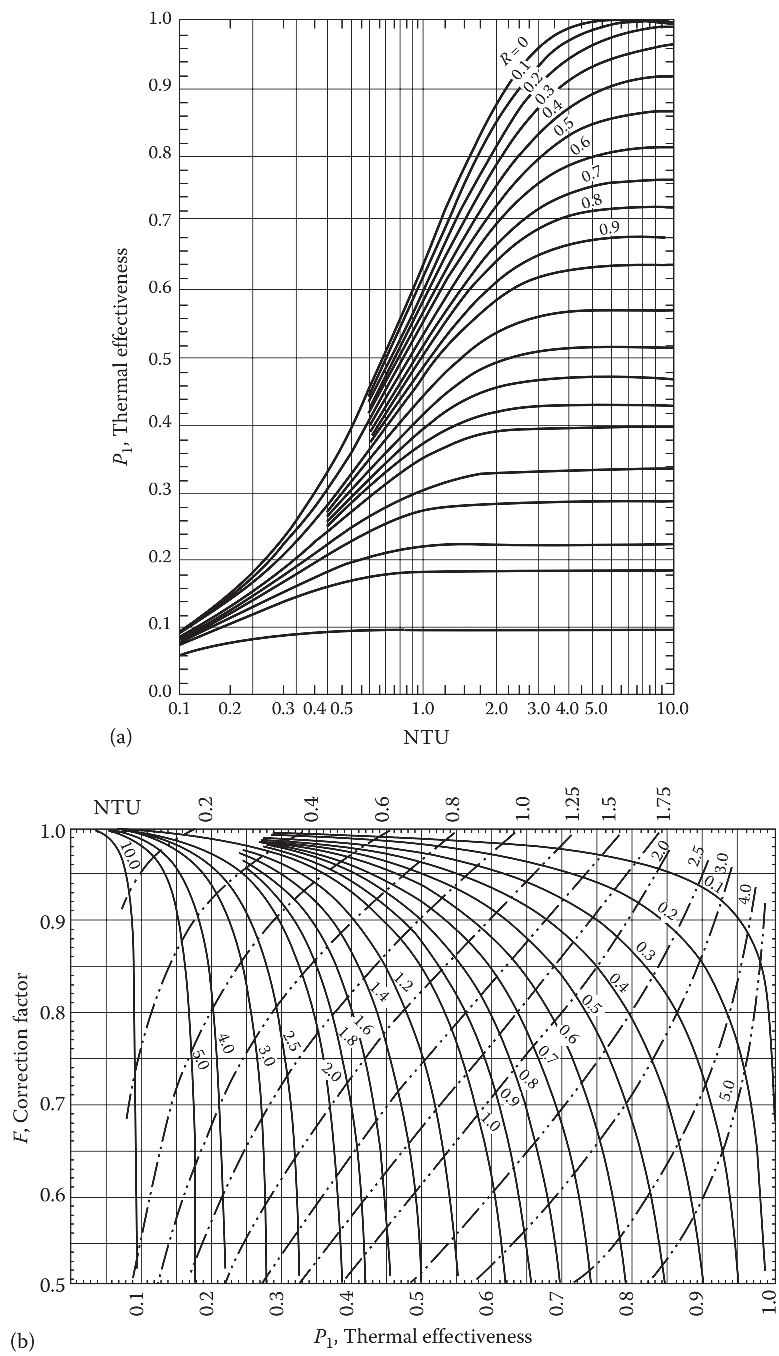


FIGURE 2.10 (a) Thermal effectiveness chart—crossflow: unmixed–mixed—the weaker (C_{min}) fluid mixed, R–P–NTU chart (as per Equation T6, Table 2.4); (b) F–R–P–NTU chart; F as a function of P for constant R (solid lines) and constant NTU (dashed lines) (as per Equation T6, Table 2.4).

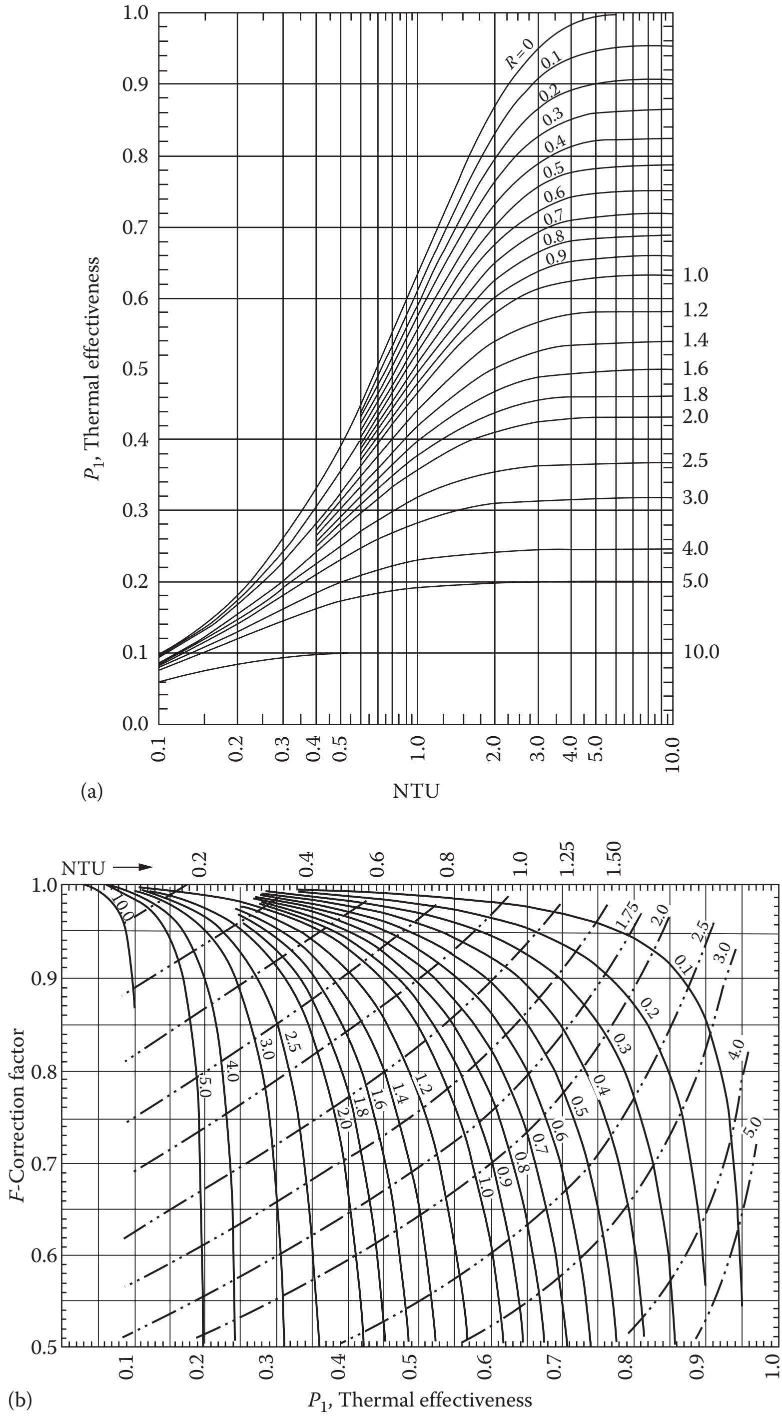


FIGURE 2.11 (a) Thermal effectiveness chart—crossflow: unmixed–mixed—the stronger (C_{max}) fluid mixed, ε-C^*-NTU chart (as per Equation T7, Table 2.4); (b) Thermal effectiveness chart: F–R–P–NTU chart; F as a function of P for constant R (solid lines) and constant NTU (dashed lines) (as per Equation T7, Table 2.4).

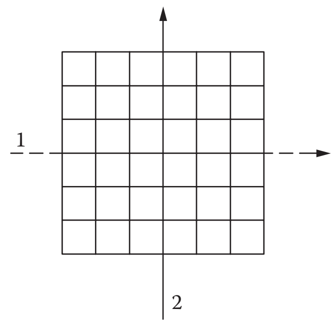


FIGURE 2.12 Unmixed–unmixed crossflow arrangement.

where

$$\binom{i}{j} = \frac{i!}{(i-j)!\,j!} \tag{2.60}$$

that is, the number of combinations of i and j taken j at a time, and

$$K = 1 - \exp\left(-\frac{\text{NTU}}{N}\right) \tag{2.61}$$

By substituting $N = 1, 2, 3, \ldots$ in Equations 2.59 and 2.61, equations for thermal relations are obtained for the specific arrangements by Nicole [28], and this is given in Table 2.5 (Equations T9 through T12) for one row, two rows, three rows, and four rows. For a larger number of tube rows (for all practical purposes, when N exceeds 5), the solution approaches that of unmixed–unmixed crossflow arrangement. Values of F for $N = 1, 2, 3, 4$ are shown in Figures 2.13 through 2.16 and are always less than the basic case of unmixed–unmixed crossflow (Figure 2.8).

2.6.3.4.2 Multipass Tube Rows Cross-Counterflow Arrangements, Both Fluids Unmixed, and Multiple Tube Rows in Multipass Tube Rows, Cross-Counterflow Arrangements

This would apply to a manifold-type air cooler in which the tubes in one row are connected to the next by U-bends. The solutions are based on Ref. [28]. Solutions for the 2 rows-2 pass and 3 rows-3 pass cases are based on Stevens et al. [29]. The general formula for thermal effectiveness referred to the air side (fin side) is given by [28]

$$P_1 = \frac{1}{R}\left(1 - \frac{1}{\zeta}\right) \tag{2.62}$$

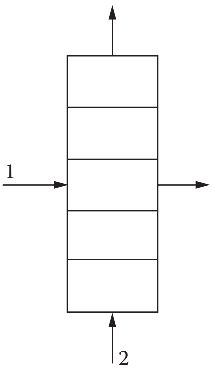
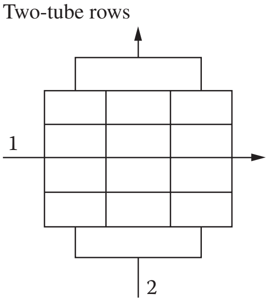
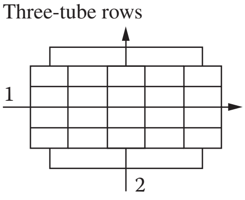
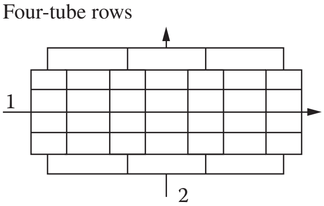
The expressions for ζ for various cases are as follows:

1. Two-tube rows, two passes, as shown in Figure 2.17a [29]

$$\zeta = \frac{K}{2} + \left(1 - \frac{K}{2}\right)e^{2KR} \tag{2.63a}$$

$$K = 1 - \exp\left(-\frac{\text{NTU}}{2}\right) \tag{2.63b}$$

TABLE 2.5
Thermal Effectiveness Relations for Tube Rows with Single Pass Arrangement

Flow Arrangement	Equation No./Reference	General Formula, Ref. [28]. Note: These Formulas Are Valid for $R = 1$
One-tube row 	T9 [28]	$P_1 = \frac{1}{R}(1 - e^{-KR})$ $K = 1 - \exp(-\mathrm{NTU})$
Two-tube rows 	T10 [28]	$P_1 = \frac{1}{R}\left[1 - e^{-2KR}(1 + RK^2)\right]$ $K = 1 - \exp\left(-\frac{\mathrm{NTU}}{2}\right)$
Three-tube rows 	T11 [28]	$P_1 = \frac{1}{R}\left\{1 - \left[\frac{e^{3KR}}{1 + RK^2(3 - K) + (3/2)R^2K^4}\right]^{-1}\right\}$ $K = 1 - \exp\left(-\frac{\mathrm{NTU}}{3}\right)$
Four-tube rows 	T12 [28]	$P_1 = \frac{1}{R}\left\{1 - \left[\frac{e^{4KR}}{[1 + RK^2(6 - 4K + K^2) + 4R^2K^4(2 - K) + (8/3)R^3K^6}\right]^{-1}\right\}$ $K = 1 - \exp(-\mathrm{NTU}/4)$

Note: To find P_2, use Equation 2.55.

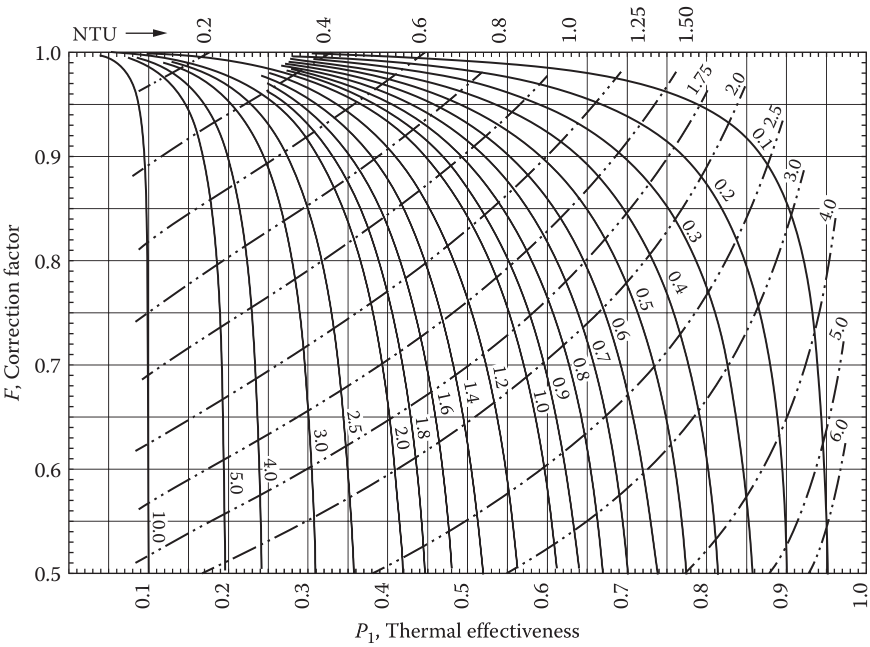


FIGURE 2.13 Thermal effectiveness chart—one-tube row; F as a function of P for constant R (solid lines) and constant NTU (dashed lines) (as per Equation T9, Table 2.5).

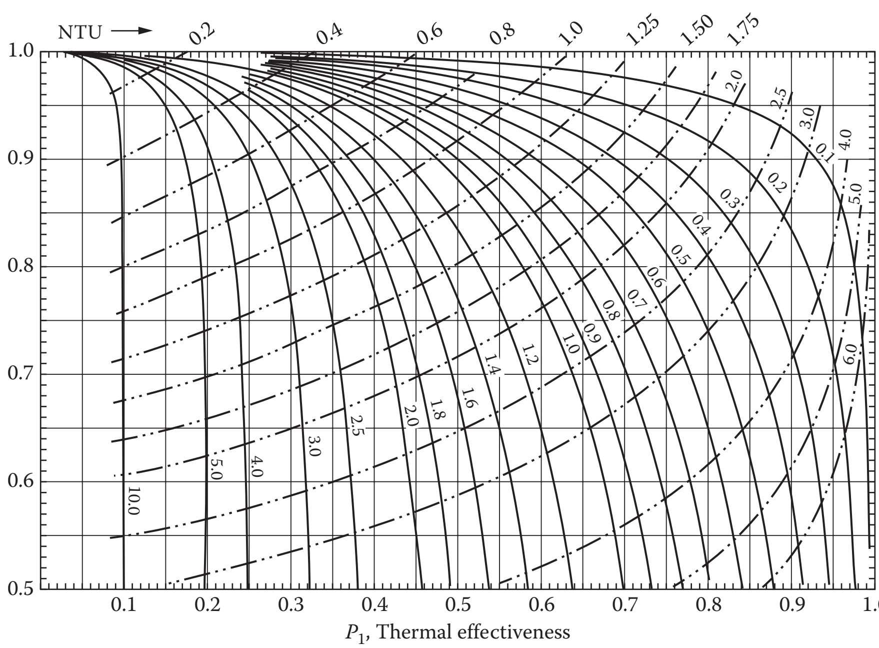


FIGURE 2.14 Thermal effectiveness chart—two-tube rows; F as a function of P for constant R (solid lines) and constant NTU (dashed lines) (as per Equation T10, Table 2.5).

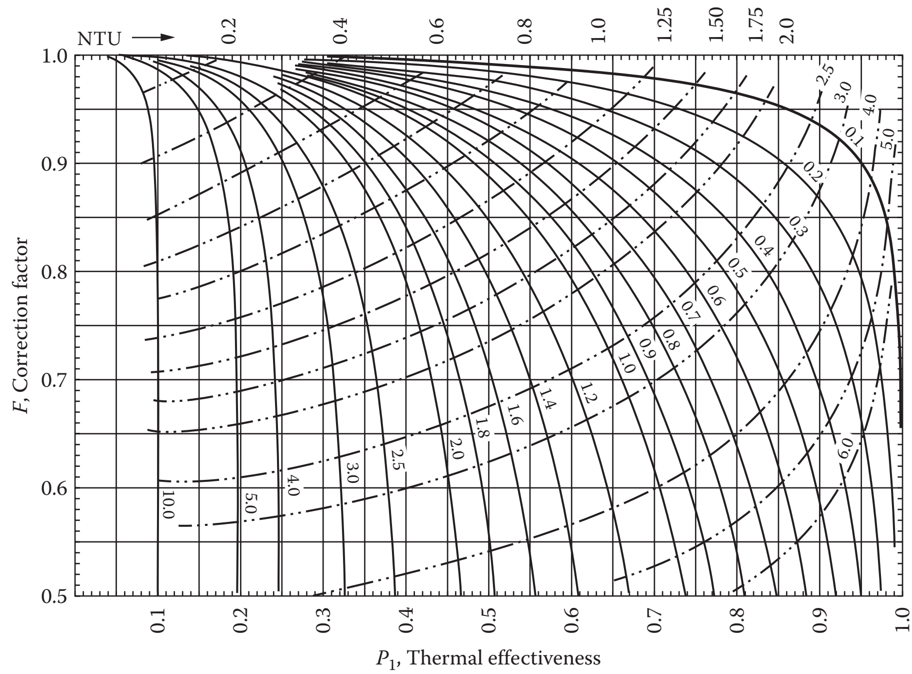


FIGURE 2.15 Thermal effectiveness chart—three-tube rows; F as a function of P for constant R (solid lines) and constant NTU (dashed lines) (as per Equation T11, Table 2.5).

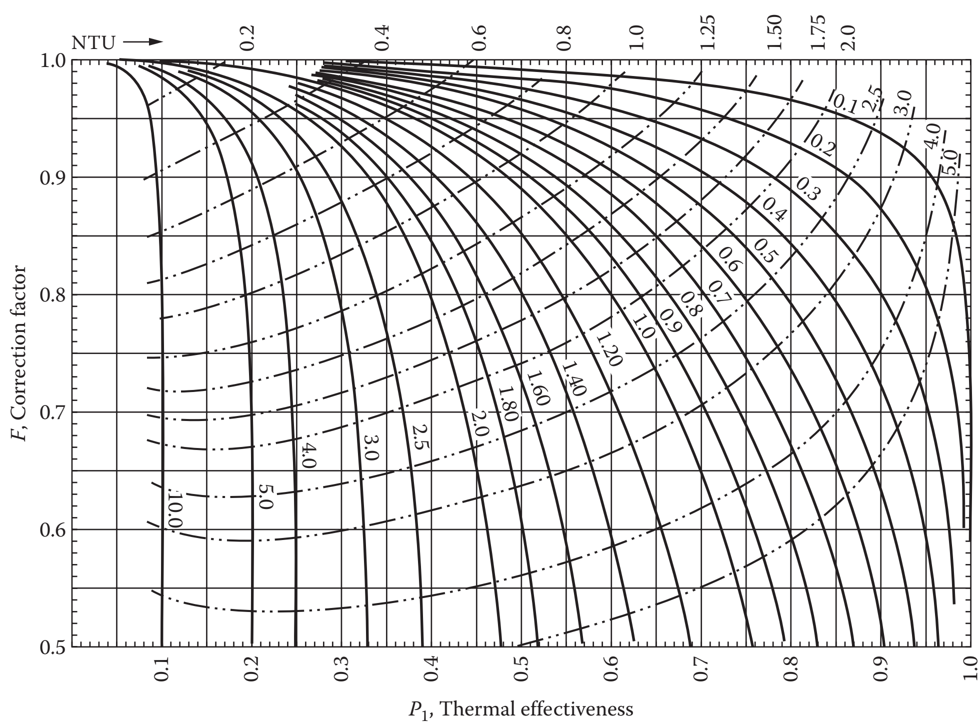


FIGURE 2.16 Thermal effectiveness chart—four-tube rows; F as a function of P for constant R (solid lines) and constant NTU (dashed lines) (as per Equation T12, Table 2.5).