HEAT TRANSFER in TUBE BANKS in CROSSFLOW

A. Žukauskas
R. Ulinskas

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A. Žukauskas

Academy of Sciences of the Lithuanian SSR, Vilnius

R. Ulinskas

Institute of Physical and Technical Problems of Energetics Kaunas, Lithuanian SSR

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HEAT TRANSFER IN TUBE BANKS IN CROSSFLOW

Experimental and Applied Heat Transfer Guide Books

A. Žukauskas, *Editor*

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J. Stasiulevičius and *A. Skrinska,* Heat Transfer of Finned Tube Bundles in Crossflow

A. Žukauskas and *R. Ulinskas,* Heat Transfer in Tube Banks in Crossflow

IN PREPARATION

A. Žukauskas, V. Katinas, and *R. Ulinskas,* Fluid Dynamics and Flow-Induced Vibrations of Tube Banks

PREFACE

Heat transfer, resistance, and characteristics of a flow across banks of smooth, rough, and finned tubes in a wide range of variation of typical parameters have long been investigated at the Institute of Physical and Technical Problems of Energetics of the Academy of Sciences of the Lithuanian SSR.

These questions were discussed in three monographs in the series *"Thermophysics"*: A. Žukauskas, V. Makarevičius, and A. Šlančiauskas, *"Heat transfer in banks of tubes in crossflow of liquid,"* J. Stasiulevičius and A. Skrinska, *"Heat transfer in banks of finned tubes in crossflow,"* and A. Žukauskas, R. Ulinskas, and V. Katinas, *"Hydrodynamics and vibrations of banks of tubes in flow."*

The authors of the present book accumulated additional material on the study of local and average heat transfer for in-line and staggered banks of tubes and the determination of their optimal pitch, roughness parameters, finning, and effectiveness. A method is suggested for generalizing experimental data to optimize the design of tube banks and to give the desired heat transfer and resistance characteristics. The material is enhanced with new data and is presented to the reader in a convenient form—as graphs and generalized equations normally required for practical computations.

In addition to the material in monographs and papers published during the last few years, the authors of the present book made extensive use of data from later investigations carried out by them with Č. Sipavičius and other colleagues.

The authors express their sincere gratitude to J. Žiugžda, L. Burkoi, and to all those who helped with the preparation of the manuscript.

NOMENCLATURE

$a = s_1/d$	relative transverse pitch
a	thermal diffusivity, m^2/sec
$b = s_2/d$	relative longitudinal pitch
$b' = s_2'/d$	relative diagonal pitch of staggered bank
$c_f = 2\tau_w/\rho\bar{u}_1^2$	coefficient of friction resistance
c_p	heat capacity, J/(kg · K)
d	tube diameter
F	heat transfer surface, m^2
h	height of fin, m
k	height of roughness element, m; overall coefficient of heat transfer, W/(m^2 · K)
$k^+ = ku_*/\nu_f$	dimensionless roughness height
l, L	length, m
p	pressure, Pa
Δp	pressure drop, Pa
Q	heat rate (heat flowrate), W
q	heat flux (density of heat flowrate), W/m^2
s	fin pitch, m
s_1, s_2	transverse and longitudinal pitches between tubes of a bank, m
s_2'	diagonal pitch of staggered bank of tubes, m
t, T	temperature, °C, K
U_0	incoming velocity, m/sec
U	maximum velocity, m/sec
$\bar{u}$	average velocity, m/sec
V	volume enclosed by heat transfer area, m^3
u, v, w	velocity components, m/sec
$\overline{u', v', w'}$	fluctuation components of the velocity
$u_* = \sqrt{\tau_w/\rho}$	dynamic velocity, m/sec
$u^+ = u/u_*$	dimensionless velocity
x, y, z	Cartesian coordinates, m
α	local coefficient of heat transfer, W/(m^2 · K)
βh	fin parameter, dimensionless fin height
δ	thickness of hydrodynamic boundary layer, m
δ^*	displacement thickness, m
δ^{**}	momentum thickness, m

δ_T	thickness of thermal boundary layer, m
ϑ	temperature, read from the wall, °C
$\vartheta + q_w/\rho c_p u_*$	dimensionless temperature
$\Pi = F/V$	compactness of heat exchanger, 1/m
λ	thermal conductivity, W/(m · K)
μ	coefficient of dynamic viscosity, Pa · sec
ν	coefficient of kinematic viscosity, m^2/sec
ρ	density, kg/m^3
τ	shear stress, Pa
φ	angle, degree
$\mathrm{Eu} = \Delta p/\rho \bar{u}^2$	Euler number
$\mathrm{Nu} = \alpha d/\lambda$	Nusselt number
$\mathrm{Pr} = \nu/a$	Prandtl number
$\mathrm{St} = \alpha/\rho c_p u_1$	Stanton number

SUBSCRIPTS

f, 0	free stream (main flow)
w	conditions on the wall
x	local conditions
$(\bar{\ })$	averaging
$(\ )'$	fluctuation component

Other notations are given in the text.

CONTENTS

CHAPTER
ONE

INTRODUCTION

1.1. GENERAL

Consumption of fuel energy sources is rapidly increasing; therefore, the efficient and rational use of these sources is considered an economical objective of extreme importance. In power engineering and in industry, the greater part of heat energy is transferred using various heat-exchange equipment. Generally, heat exchangers operate on the principles, "gas-gas," "vapor-liquid," "liquid-gas," or "liquid-liquid." Tubular heat exchangers, and other devices with heating surfaces made of tubes, are widely used. Therefore, there is a need to augment heat transfer processes and to increase the thermal efficiency of tubular heat exchangers. In the present book, the problems of heat transfer for tubes with external flow are discussed.

Intensity of heat transfer depends mainly on the type of thermal carrier. For example, for similar conditions and equal flow velocities, the heat transfer coefficient in a stream of water is one to two orders of magnitude higher than in an air stream, even though air is less aggressive chemically than water.

Flow velocity and stream conditions, which depend mainly on the type and design of the heat exchanger, have great influence on the intensity of heat transfer. In practice, the most widely used heat exchangers are shell-and-tube exchangers with smooth, straight tubes, U-shaped tubes, or helical tubes (Fig. 1.1.). These are typically used at high pressures and temperatures. The ratio of specific metal content to heat output of shell-and-tube heat exchangers is relatively high.

The problem of intensification (augmentation) of heat transfer in gas-gas

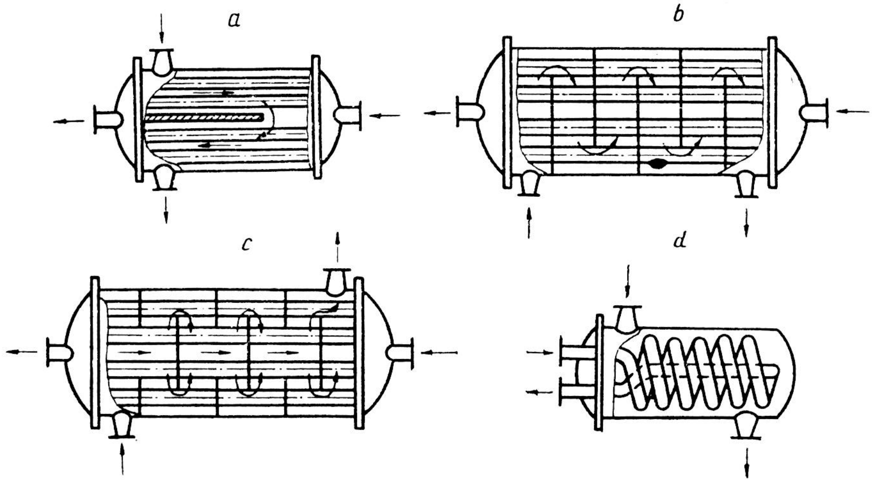

Figure 1.1. Types of shell-and-tube heat exchangers. *a*) with longitudinal and transverse flow around tubes; *b*) with segmented baffles; *c*) with annular baffles; *d*) helical.

heat exchangers is of prime importance, since the heat transfer coefficient for both carriers is low, and the size of the exchanger is large. Consequently, metal consumption is considerable. The heat transfer coefficient for air is always much lower than for liquids. Therefore, measures should be taken to increase heat removal from the air side. In order to achieve this, finned surfaces, allowing an increase of the heat transfer area by 20 times or more, are widely used. In order to intensify heat transfer by finned surfaces, the fins are cut and deformed. This increases the heat exchange at the roots where the velocity of the thermal carrier is particularly low.

Plate and spiral heat exchangers are also used extensively in practice, since they are easy to manufacture, compact, and have low metal content. The main disadvantage of plate heat exchangers is their inability to operate at high pressures. The majority of problems connected with heating, cooling, and medium condensation are solved by using developed heat transfer surfaces. It is possible to balance the heat transfer coefficient, decrease the weight, size, and cost of heat exchangers.

In spiral heat exchangers, full counterflow is provided, there is no sharp change in flow direction, and fouling and hydraulic resistance are lower than in shell-and-tube heat exchangers. They are widely used for heating or cooling highly viscous fluids by employing the liquid-liquid scheme.

Modern energy equipment in which thermal energy of burned fuel is converted to electrical energy, and heat-exchange devices found in industrial equipment do not always operate at optimum conditions. In this regard, the investigations directed towards determining the optimum conditions and parameters for the operation of heat exchangers are of paramount importance.

In order to increase thermal efficiency, preserve scarce metals, and decrease the mass and size of heat-exchange equipment, it is necessary to consider the possibility of the rational increase in compactness. The question of heat transfer augmentation should be solved with the help of the most rational means, depending on flow conditions and the type of thermal carrier.

The investigations aimed at improving heat-exchanger energy factors are of great practical interest. These are usually conducted over a wide range of varying characteristics which determine the heat transfer and permit the derivation of generalized relations for smooth, rough, and finned surfaces. The results of these investigations make it possible to compare various heat transfer surfaces and choose the optimum versions and conditions of operation of heat exchangers.

In designing modern heat exchangers, it is imperative to know not only the heat transfer in inner rows of the banks, but also the heat transfer in the first rows, where it is usually lower. When the banks contain few rows, heat transfer is determined from one row to another.

A significant portion of the heat transfer surface in a shell-and-tube heat exchanger is subjected to flow at a certain angle. Therefore, when designing such exchangers the variation of heat transfer as a function of the flow angle of attack relative to the tube must be taken into account.

During usage, technically smooth tubes become rough due to technological processes, fouling, and erosion of tube surfaces. Therefore, it is important to know the dependence of heat transfer on the degree of surface roughness.

Consideration of these and a number of other factors is important to the design of efficient heat exchangers.

Considerable attention has been paid recently to the problems of heat transfer computations in heat exchangers. This is due to the development of nuclear power generation and of the chemical industry, further improvement of various industrial equipment, and the development of energy systems for the transportation sector.

At the Institute of Physical and Technical Problems of Energetics of the Academy of Sciences of the Lithuanian SSR (IPTPE AS LitSSR), a study has long been conducted on the mechanism of heat transfer and resistance in heat exchangers and in their main components, namely, tube banks in crossflow, which are widely used in numerous branches of the industry [1–6].

In this book, the results of experimental and theoretical investigations of heat transfer in banks of smooth, rough, and finned tubes are presented. These results have been obtained at IPTPE AS LitSSR and other scientific centers both in the USSR and abroad for the last 25 years. Considerable attention has been given to the characteristics of local heat transfer along the tube perimeter, including separation and recirculation regions of liquid flow obtained by flow visualization and photography. The problems of resistance in flows around tube banks have been discussed in detail in later works by the authors [3, 5].

In order to increase the thermal efficiency of heat-exchange apparatus, the mechanism of heat transfer is investigated and the possibility of heat transfer augmentation is examined. Proper space is allocated to the analysis of the characteristics of developed heat-exchange surfaces which are implemented in the form of finned banks of tubes.

The material presented in the book is based on research carried out in flows of air, water, transformer and aviation oils over the Reynolds number interval from 1 to 2×10^6 and Prandtl numbers from 0.7 to 10^4.

1.2. THERMAL DESIGN PRINCIPLES OF RECUPERATORS

Modern heat exchangers which operate within a wide range of velocities, physical properties of liquids, temperatures, and pressures must have the following qualities:

- transmit the predicted amount of heat from one thermal carrier to another and be stable in operation within the stipulated range of temperatures; the heat flow must be carried out with the maximum possible heat transfer coefficients and at the given velocities of the thermal carriers;
- incur minimum hydraulic losses during the transfer of the given amount of heat;
- be reliable and convenient in operation;
- be compact, light, strong, etc.

Satisfying all these requirements is extremely difficult. For example, augmentation of heat transfer by increasing flow velocity leads to an increase in hydraulic resistance. Therefore, the need arises to optimize a multitude of interrelated factors.

In practice, two types of heat exchangers are widely used: non-accumulators and accumulators of heat. In the first case, the heat flow is transferred through a wall from one thermal carrier to another. Such heat exchangers are sometimes called recuperators. In heat exchangers that accumulate heat, the same heat transfer surface comes into contact with the thermal carriers alternately. The temperature of the wall and thermal carrier changes in time, causing the process of heat transmission to be non-stationary. Such heat exchangers are called regenerators. In the present book, heat transfer in tubular recuperators is the subject of discussion.

The main problem accompanying thermal calculations is the determination of heat exchange surfaces. If these are known, then the problem is to determine the amount of heat transmitted, final temperatures of the thermal carriers, and operating conditions of the heat exchanger.

Heat exchanger surface is determined by the simultaneous solution of the heat balance equation

$$dQ = c_p G dt \tag{1.1}$$

and the equation for overall heat transfer

$$dQ = k \, \Delta t dF \tag{1.2}$$

where

$$dQ = c_{p1} G_1 dt_1 = c_{p2} G_2 dt_2 = k \, \Delta t dF \tag{1.3}$$

Subscript 1 denotes direction towards the hot thermal carrier, and subscript 2 denotes direction towards the cold thermal carrier; $\Delta t = t_1 - t_2$ is the difference of mean temperatures of the hot and cold thermal carriers.

From the balance equation (1.3) in its integrated form, the heat transfer surface of the element at a constant flow rate of the thermal carriers is determined by

$$F = G_1 \int_{t_1''}^{t_1'} \frac{c_{p_1} dt_1}{k \Delta t} \tag{1.4}$$

or

$$F = G_2 \int_{t_2'}^{t_2''} \frac{c_{p_2} dt_2}{k \Delta t} \tag{1.5}$$

where c_{p1}, c_{p2} is the specific heat capacity of the hot and cold thermal carriers; t_1', t_1'', t_2', and t_2'', are the temperatures of the hot and cold thermal carriers at the inlet and outlet of the heat exchanger. If the values of c_{p1}, c_{p2}, dt_1, dt_2, k and Δt change insignificantly along the length of heat transfer surface, then the heat balance equation can be expressed as follows:

$$Q = c_{p_1} G_1 (t_1' - t_1'') = c_{p_2} G_2 (t_2'' - t_2') = k (t_1 - t_2) F \tag{1.6}$$

The heat transfer surface is determined from equation (1.6):

$$F = c_{p_1} G_1 (t_1' - t_1'') / k (t_1 - t_2) \tag{1.7a}$$

or

$$F = c_{p_2} G_2 (t_2'' - t_2') / k (t_1 - t_2) \tag{1.7b}$$

The character of temperature change is peculiar to recuperators with parallel and counterflow directions of the thermal carriers (Fig. 1.2).

The water equivalent, sometimes called the water number, is of particular importance:

$$W_1 = c_{p1} G_1, \; W_2 = c_{p2} G_2 \tag{1.8}$$

Mass flow rate of a thermal carrier G is often replaced by volume flow rate V. In this case, the water equivalent referred to 1 m^3 of the thermal carrier:

$$W = c_p V \rho \tag{1.9}$$

Taking the water equivalent into account, equation (1.3) yields

$$dQ = -W_1 \, dt_1 = -W_2 \, dt_2 \tag{1.10}$$

or

$$Q = W_1 (t_1' - t_1'') = W_2 (t_2'' - t_2') \tag{1.11}$$

from which the flow rate of the thermal carrier or the amount of transmitted heat can be found. From equation (1.11) an important relation follows

$$\frac{t_1' - t_1''}{t_2'' - t_2'} = \frac{W_2}{W_1} \tag{1.12}$$

Relation (1.12) is valid for the whole heat transfer area and for an individual element dF:

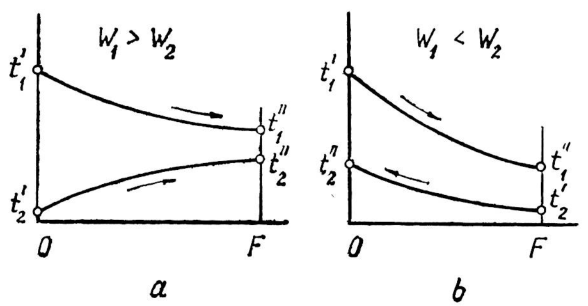

Figure 1.2. Temperature change in parallel-flow (*a*) and counterflow (*b*) recuperators.

$$\frac{dt_1}{dt_2} = \frac{W_2}{W_1} \tag{1.13}$$

Relations (1.12), (1.13) allow to determine the optimum values of dt_1 and dt_2 depending on the values of the water equivalent of the thermal carriers.

For significant temperature changes of thermal carriers along the length of a heat transfer surface, the specific heat capacity and the coefficient of heat transfer are functions of temperature:

$$c_{p1} = f(t_1) \tag{1.14a}$$

$$c_{p2} = f(t_2) \tag{1.14b}$$

Then equation (1.7) transforms to

$$F = G_1 \int_{t_1''}^{t_1'} \frac{c_{p_1}(t_1)\, dt_1}{k\,(t_1 - t_2)} \tag{1.15a}$$

or

$$F = G_2 \int_{t_2'}^{t_2''} \frac{c_{p_2}(t_2)\, dt_2}{k\,(t_1 - t_2)} \tag{1.15b}$$

the solution of equation (1.15) is possible if the values c_{p1} (t_1) c_{p2} (t_2), and $k(t_1 - t_2)$ are known. They are, however, very complex and are not convenient for practical applications. Therefore, for separate parts of the tube bank, it is expedient to use numerical integration of equations (1.15) with local values of the integrated mean time intervals dt_1, and dt_2 from which the thermophysical properties of the heat carriers are determined. It is assumed, in this case, that the physical properties of the thermal carriers are constant within the bounds of the single section of the tube bank.

The flow condition of the thermal carrier (parallel or counterflow) influences how dt_1 and dt_2 vary along the length of the heat transfer surface.

In practice, counterflow is far more common due to the larger heat flux it gives as compared with parallel flow. Fig. 1.2 shows that the final temperature of the cold thermal carrier t_2'' for parallel flow is always lower than the final temperature of the hot thermal carrier t_1''. For counterflow t_2'' can be higher than t_1''.

The mean temperature difference (head) is determined by various methods, and it depends on the conditions of the heat transfer process. If the temperature variation across the heat transfer surface is insignificant, then the mean temperature head (difference) depends on the character of the thermal-carrier move-