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PREFACE

Heat exchangers nowadays are widely used in aviation engineering for cooling aviation engine systems, vehicle members, aircraft instrument compartments, and cabins [30].

Such heat exchangers must be small in size and mass, have minimum hydraulic losses, and be highly reliable in operation. These demands also hold true for heat exchangers employed in power engineering, chemical, and other branches of industry, since these account for the essential part of power plants and basic equipment by mass and volume. Thus, in chemical and petroleum refinery industries, heat exchangers account for a major proportion of the mass and cost of the basic plant.

Also, the share of heat exchanger mass and volume is appreciable in heat engines. Heat transfer augmentation in channels due to flow swirling [56, 57] is among one of the more promising ways to design compact heat exchangers. In [7], intricate heating surfaces shaped as helical ovals and three-petal tubes were proposed to increase the efficiency and reliability of power-stressed installations. A crossflow helical tube heat exchanger was considered in [6] where it was shown to be possible to improve heat transfer at small twisting pitches of tube blades for moderate pressure losses.

Enhancement of convective heat transfer and flow mixing is, at present, one of the urgent problems. Its solution is of great scientific and practical importance. Decreasing the size and mass of heat exchangers, the amount of metal spent for their production, and their cost improves crossflow mixing, reduces cross-sectional temperature nonuniformity, and diminishes heat exchanger surface heating.

Heat transfer in circular tube bundles can be mainly enhanced by increasing a heat carrier velocity, which is far from being beneficial because of drastic energy consumption. The development of artificial roughness, the use of corrugated channels, and the mounting of the wire, diaphragms, washers, and rods in the channels hinder heat transfer increase as opposed to hydraulic resistance and energy consumption for heat carrier pumping.

In helical oval tube heat exchangers [2, 13, 24, 58, 59], convective heat transfer is enhanced by flow swirling in channels intricately shaped by a close-packed bundle of such tubes. In this case, success is attained not only in improving heat transfer due to flow swirling both inside the helical tubes and in the intertube space, but also in a substantial increase of heating surfaces per unit volume of the apparatus. Higher thermal and hydraulic characteristics can be obtained in helical tube heat exchangers, as compared to circular tube apparatus, since helical heat carrier swirling in complex-geometry channels originates transverse velocity components, additional turbulization, and secondary circulation of the flow. These mechanisms cause an intensive exchange of liquid portions between the wall layer and the flow core, thus improving heat and mass transfer.

The spiral cross flow in the intertube space of a helical tube heat exchanger allows a temperature field to be levelled in the intertube space and at its outlet; and to increase apparatus efficiency and its operational reliability. Moreover, the spiral flow swirling substantially reduces wall temperature nonuniformity over the perimeter of helical tubes.

This monograph proposes the physically grounded helical tube flow models and their mathematical description. Some methods of approximate closure of the equation systems for the flow in the core and in the wall layer are considered. The possibility of using the methods of calculating a boundary layer at the entrance length of a bundle is shown. The similarity and dimension theories were adopted to propose a new similarity number to take into account specific features of the flow in a helical tube bundle and to correlate the experimental data on heat transfer, interchannel heat carrier mixing, hydraulic resistance, and flow structure. The same similarity number was obtained using the semiempirical turbulence theory.

The proposed flow model based on the effective wall layer thickness allows the laws for heat transfer and friction in the circular tubes to be used to generalize experimental data on heat transfer and hydraulic resistance in helical tube bundles. Such experimental data processing offers ample scope for modelling and diminishes the number of experiments to establish criterial relations. The flow model for a homogenized medium replacing a real tube bundle is experimentally substantiated. The methods are proposed to solve the systems of flow equations for axisymmetric and asymmetric heat supply nonuniformity. The criterial relations are specified to calculate the effective turbulent viscosity coefficient and thermal conductivity that enter these equations. Experimental methods to study heat transfer and flow dynamics are developed.

The discovered specific features of the flow made it possible to substantiate thermal and hydraulic characteristics depending on basic similarity numbers. The criterial relations for heat transfer and hydraulic resistance in longitudinal and crossflows past and inside helical tube bundles are employed to estimate the efficiency of heat exchangers with such tubes. The heat exchanger efficiency estimated by the elaborated methods has shown that, at the assigned heat power and the same hydraulic losses, the helical tube bundles, when used instead of the straight circular ones, offer about a 20 to 30% decrease of the heat exchanger mass and volume.

Preface, Chapters 1, 3, and 4 were written by B. V. Dzyubenko, Chapters 2 and 5 were written by B. V. Dzyubenko, G. A. Dreitser, and Yu. I. Danilov, Chapters 6 and 7 were written by L. A. Ashmantas, and Chapter 8 was written by G. A. Dreitser.

NOMENCLATURE

a	thermal diffusivity, coefficient for the flow structure
b	"half" width of a jet, $b = 2r_{\text{mean}}$
c_p	heat capacity
d	maximum size of a tube profile
d_{eq}	equivalent diameter
D_t	effective heat diffusion coefficient
E	wire voltage drop
e'	voltage fluctuation
F_{f}	area of the flow cross section of a tube bundle
G	air mass flowrate
G_i	axial heat carrier flowrate in a cell
G_{ij}	heat carrier flow in the transverse direction from a cell i to a cell j per unit length of a channel
I	enthalpy of unit mass
j_0	enthalpy head in a boundary layer
k	dimensionless effective diffusion coefficient
$\bar{k}$	mean value of the coefficient k
L_{E}, L_{L}	spatial integral turbulence scales for the flow according to Euler and Lagrange
l_{c}	control section length
l	mixing path
m	bundle porosity with respect to heat carrier

q	energy flow in a boundary layer; heat flux density
q_v	volumetric density of heat release
p	pressure
r	radial coordinate
r_k	bundle radius
r_{mean}	"half" jet radius
s	tube twisting pitch
T	temperature
t	time
t_{mean}	mean temperature head in a boundary layer
u, v, w	averaged velocity components in the orthogonal coordinate system
u', v', w'	pulsational velocity components
u_t, u_r	tangential and radial velocity components in cylindrical coordinates
V	averaged velocity vector modulus
v_1	mean quadratic pulsational velocity
v_c	velocity vector component along the x_c axis (equations are written in a tensor form)
$w^2 = \overline{(v_i' v_i')}/3$	pulsational squared velocity to a turbulence isotropy approximation
x_k (where $k = 1, 2, 3$)	Cartesian coordinates
$\bar{y}^2$	mean statistical squared displacement
α	heat transfer coefficient, dimensionless friction coefficient
α_m	dimensionless heat transfer coefficient
Γ	flow swirling degree
Δp	pressure drop
δ	wall layer thickness
δ^*	displacement thickness of a boundary layer
ϵ	effective tubulence intensity
λ	thermal conductivity
μ	dynamic viscosity coefficient, coefficient for interchannel flow mixing (Chapter 4)
η	second viscosity coefficient
ν	kinematic viscosity coefficient
ν_{eff}	effective turbulent viscosity coefficient
λ_{eff}	effective thermal conductivity
ξ	hydraulic resistance coefficient
ρ	density
τ	shear stress
τ_{xw}	axial component of wall shear stress
τ_{zw}	tangential component of wall shear stress
τ_{z0}	tangential component of shear stress along the channel axis
$\tau_{\Sigma w}$	total wall shear stress

φ	angular coordinate, angle between the anemometer wire and the velocity vector direction
φ'	fluctuation of an angle φ
ψ	temperature factor (Chapters 6 and 7), bundle porosity with respect to heat carrier (Chapter 8)
Fr_M	number for the specific features of the flow in a helical tube bundle
Nu	Nusselt number
Pe	Peclet number
Pr	Prandtl number
Re	Reynolds number

SUBSCRIPTS

in	inside a tube
hot	hot side of a heat exchanger
max	maximum, modified
c	control, casing
out	outside a tube
o	on the bundle axis
f	flow
w	wall
mean	mean mass
t	turbulent
tube	tube
cold	cold side of a heat exchanger
c	centrifugal
1, 2	experimental section inlet and outlet
I, II	heat exchanger inlet and outlet, respectively
d	determined with respect to d_{eq}
m	at a mean wall layer temperature
δ	determined with respect to a wall layer thickness

CONTENTS

CHAPTER

ONE

SPECIFIC FEATURES OF THE PROCESSES OF HEAT TRANSFER AND FLOW IN COMPLEX-GEOMETRY CHANNELS

1.1 SPECIFIC FEATURES OF THE DESIGN OF A HELICAL TUBE HEAT EXCHANGER

The specific features of heat transfer and fluid dynamics in complex-geometry channels formed by helical tube bundles are defined by the structural characteristics of these bundles. Figure 1.1 is a schematic of a shell-tube heat exchanger with oval-shaped helical tubes, with their straight round ends fastened into the tube plates*. The tubes in this heat exchanger are located relative to each other such that they have contact over a maximum amount of the oval. A spiral swirling of the fluid occurs by its circulation in the tubes and in the intertube spaces. The flow in the intertube space is of a more complex nature, which may be conventionally considered as a system of alternating interconnected spiral and through channels. The turbulence in such a system is generated by the fixed wall and is due to the friction between liquid layers with different velocities. The flow is swirled in opposite directions in the spiral channels of adjacent tubes, which results in discontinuities of the tangential velocity components. The longitudinal velocity components in the flow core also undergo tangential discontinuities due to different flow conditions in the through channels and behind the locations of contacting adjacent tubes. The flow in a helical tube bundle is also affected by secondary circulations due to the centrifugal forces that exist with fluids flowing in spiral channels. Thus, the turbulence in a helical tube bundle enhances heat transfer and interchannel flow mixing.

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*B. V. Dzyubenko and Yu. V. Vilemas. A shell-tube heat exchanger. Author's Certificate No. 761820 (USSR). Bulletin of Inventions, 1980, No. 33, p. 194.

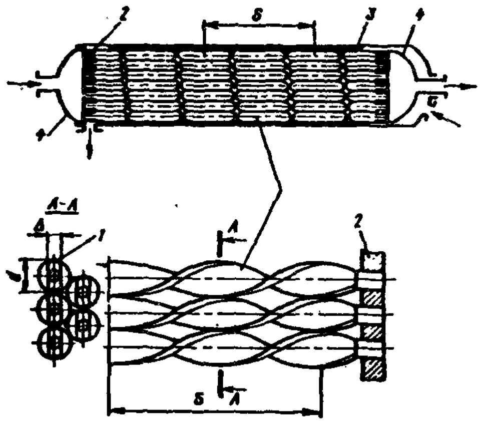

Figure 1.1 Swirled flow heat exchanger: *1*) helical tubes; *2*) tube plates; 3) tube shell; *4*) bottom.

It is a characteristic of this type of flow that flow swirling is produced along the entire tube bundle and therefore the flow structure is stabilized at some distance from the inlet. This results in a stabilization, on the average, of the coefficients of heat transfer, hydraulic resistance, and mixing, although the heat transfer coefficient may vary along the perimeter of a complex channel, such as a helical tube bundle. Within some limits, the heat transfer coefficient may also periodically change along the tube bundle because of the periodic spacing of the tubes in contact.

The flow in a helical tube bundle is spatial, i.e., together with a longitudinal velocity vector component there exist transverse velocity components that greatly enhance interchannel flow mixing in the bundle. The high turbulence level of the flow, convective transfer in a cell, and ordered transfer in the cross section of the bundle due to spiral flow swirling by means of twisted tubes are the mechanisms responsible for the specific features of cross flow mixing in a bundle, as compared to the transport phenomena in a straight circular tube.

The most important geoemtrical parameter of a helical tube bundle is the blade twisting pitch s based on the maximum oval-shaped tube size d. This parameter, to a considerable degree, specifies the intensity of the centrifugal force field in the bundle as well as the characteristic properties of heat transfer and mixing of the heat transfer fluid. An optimum relative twisting pitch s/d is determined only when maximum improvement of these processes is provided at acceptable values of the hydraulic resistance coefficients for a bundle. In this case, it is possible to design rather compact helical tube heat exchangers. Solving the urgent problems of convective heat transfer and mixing enhancement due to circular tubes being replaced by oval-shaped ones enables one not only to substantially decrease the overall dimensions and mass of heat exchangers, their metal consumption, and cost, but also to reduce temperature nonuniformities over the heat exchanger cross section and to depress the heating of a heat exchanger surface. Such heat exchangers can be successfully used in aviation engineering as well as in different branches of industry.

The analyzed details of helical tube bundles and the specific features of the flow in these bundles may also give rise to a particular behavior of heat transfer and hydrodynamic processes as a function of the Reynolds number. In fact, from general considerations, the spiral swirling of the fluid in the transition range of Reynolds numbers must, to a great extent, improve the heat transfer and mixing at high Reynolds numbers. The mechanisms of these processes must be experimentally specified.

The swirling spiral flow initiates laminar flow instabilities at Reynolds numbers which are approximately, by an order and a half, less than those in a straight circular tube. However, unlike the flow in coils, the swirling flow in the intertube space of a helical tube heat exchanger also results in an earlier transition to turbulent flow (at $Re \approx 10^3$). In this case, the transition from laminar to turbulent flow in helical tube bundles is smooth.

In the present investigation, fully developed flow models were adopted to generalize the comprehensive data on the flow structure, heat transfer, hydraulic resistance, and interchannel fluid mixing. Also, experimentally based methods were proposed for the thermal and hydraulic calculations of helical tube heat exchangers with regard to cross mixing. In this case, preference was given to the longitudinal flow past close-packed helical tube bundles, although much attention was also paid to other structural schematics of helical tube heat exchangers. Figure 1.2 shows a schematic of a shell-tube apparatus* which differs from the previous one. In this device, a helical tube bundle is twisted relative to its longitudinal axis so that the relative twisting pitch decreases with respect to

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*B. V. Dzyubenko, Yu. V. Vilemas, R. R. Varshkyavičius and G. A. Dreytster. A shell-tube heat exchanger. Author's Certificate No. 937954 (USSR). Bulletin of Inventions, 1982, No. 23, p.189.

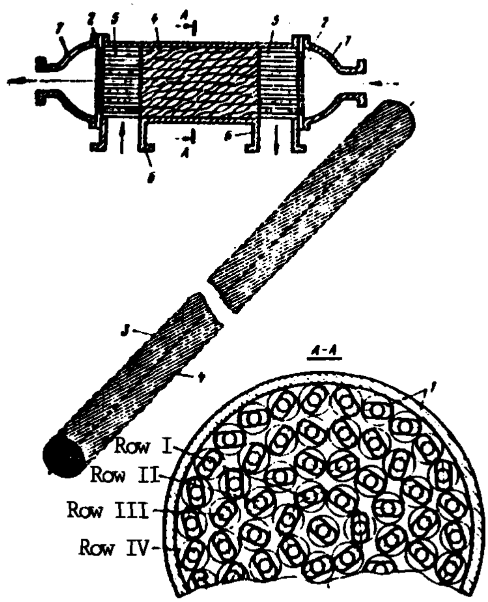

Figure 1.2 Twisted helical tube bundle heat exchanger: *1)* tube shell; *2)* tube plate; *3)* twisted tube bundle; *4)* helical tube; *5)* straight round tube ends; *6, 7)* connections.

the bundle radius, and the lengths of the straight tube ends are equal to the diameter of the inlet and outlet connections. The connecting channels are formed in the tube shell. The porosity of these channels is higher than that of the twisted part of the bundle. In this case, the heat transfer and mixing of the fluid are improved, and the inlet velocity and temperature fields are smoothed out by the lateral supply and removal of the heat transfer fluid. In addition, a relative tube deformation may be expected with varying temperatures. The helical tube twisting relative to the bundle axis initiates a radius redistribution of the fluid flow in the intertube space because of the difference in hydraulic resistance coefficients for the tube rows located on different radii. Such a heat exchanger design permits suppression of azimuthal nonuniformities of velocity and temperature that develop, in particular, in the lateral fluid flow.

Figure 1.3 shows a schematic of a crossflow helical tube heat exchanger. The design of this heat exchanger should differ substantially from a crossflow circular tube heat exchanger