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HEAT TRANSFER IN SWIRLED FLOWS

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The present article is devoted to the state-of-the art in the field of heat transfer in internal swirled flows. A wide range of channels and boundary conditions are considered; the major publications on this subject matter are cited.

Introduction. A swirled gas or liquid flow is one of the most commonly encountered flow modes in the field of centrifugal mass forces. Such a flow is characterized by specific properties which distinguish it fundamentally from an axial flow: comparable axial, rotational and radial velocities, longitudinal and transverse gradients of static and total pressure, considerable gradients of transverse velocities, a high level of turbulent fluctuations, active and conservative action of centrifugal forces on a flow structure. As a consequence, the calculation methods of swirled flows differ radically from those used for calculation of axial flows. Swirled flow specific properties are applied widely in engineering: in liquid atomizers, gas burners, vortex power separators, heat exchangers, energotechnological cyclone separators, vortex chambers and hydrocyclones, pumps driers, power plants, etc.

The problems of the aerodynamics and heat and mass transfer of swirled flows are discussed in a number of monographs. A swirled flow in tubes is considered by V. K. Shchukin and A. A. Khalatov [1-3]. Processes in cyclone chambers and in vortex tubes are investigated by G. F. Knoppe [4], A. B. Reznyakov et al. [5], B. P. Ustimenko [6], S. S. Kutateladze, E. P. Volchkov [8], A. M. Shtym [9], E. N. Saburov [10], A. P. Merkulov [11], et al. Free jet swirled flows are studied by R. B. Akhmedov [12] and Yu. I. Khavkin [13], and flows near rotating axisymmetric bodies by L. A. Dorfman [14]. Such investigations are conducted abroad by A. Gupta. D. Lilli and N. Saired [15]. However many problems of the hydroaerodynamics and heat and mass transfer of swirled flows have not been adequately explored yet.

We now analyze the state-of-the art of the hydrodynamics and heat transfer of internal channels in axisymmetric channels and chambers.

Swirlers and Their Characteristics. Gas and liquid flows may be swirled by special facilities (swirlers) which impart a rotational velocity to the flow. In practice, swirlers make it possible to generate not only rotational but also different combinations of axial and rotational motions. Swirlers may be placed at a channel inlet (local swirling) or along its length (extended swirlers). In the latter, helical threading, spiral rolling and finning, twisted strips, and wire- and screw-type swirlers are most often used [1-3].

Local swirlers used in practice may be subdivided into three basic groups [3]. The first group includes swirlers in which a rotational motion is generated in a flow to be transformed then into translational-rotational (tangential, helical, tangential-vane swirlers). Swirlers which generate both rotational and translational motion in a flow (axial-vane, axial-tangential, screw-, strip-type swirlers, rotational units) fall into the second group. Local swirlers, which produce combined axial and rotational motions in a flow, belong to the third group [3]. In the last case swirled and axial flows may be generated both in the near-wall and central regions of a swirler.

To characterize the flow swirling intensity behind a swirler, several dimensionless parameters are used. The degree of swirling is most adequately characterized by the integral parameter $\Phi_*^g = M^g/K^gL$, where M^g, K^g are calculated by geometric parameters of the swirler, L is a linear dimension. Expressions for calculating the theoretical Φ_*^g values for the three groups of swirlers are given in [2, 3]. Also, correction functions are presented which determine the actual value of this parameter and take into account the nonuniformity of the velocity field in swirler channels.

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For swirlers with partial flow swirling the swirling intensity is at its maximum at definite geometric parameters of the swirlers. Optimal (with respect to the degree of swirling) characteristics of swirlers with isolated and nonisolated channels of axial and rotational motion are given in [3]. They make it possible to determine maximal swirling parameters and to predict their change in the case of deviation of channel and swirler dimensions from those prescribed in their manufacture and utilization.

Analysis of experimental data on the velocity and pressure distributions in swirled flows in tubes with different methods of initial swirling allows two regions with a qualitatively different flow structure to be singled out [3]. The first region ($0 < \bar{x} < x_*$) lies just behind the swirler; its length is from 0.5 to 3.0 of a tube diameter. The distribution of local flow parameters in this region is governed by the method and intensity of flow swirling, the geometric size of the swirler, and other conditions.

A specific feature of the second region ($\bar{x}_* \le \bar{x} \le \bar{x}_{in}$) lies in the fact that the character of the velocity distribution in this region does not depend on the methods of initial flow swirling; it is determined only by its intensity. Here the flow is of a self-similar nature (see below); the characteristic parameters of the velocity and pressure profiles are determined by the flow swirling and Reynolds number values. For the length $\bar{x} > \bar{x}_{in}$, which is determined by the equation [2]

$$\bar{x}_{in} = \bar{x}_{in}^0[1 + (0{,}57 - 8\cdot 10^{-6}\,\mathrm{Re}_d)(\Phi_*^0 - 0{,}1)^{0{,}57 - 7\cdot 10^{-7}\mathrm{Re}_d}],$$

the specific features stipulated by the swirling no longer exist in a flow, and it is transformed into the axial one with the velocity profile of a stabilized flow. The quantity $\bar{x}_{in}$ in this case may be considered as the length of the hydrodynamic initial section of the swirled tube flow ($\bar{x}_{in}^0$ is the same for an axial flow).

The most important characteristics of a swirled flow within the region $\bar{x} < \bar{x}_*$ include the radius of the zone of backward flows behind the swirler, the angle of flow swirling on the channel wall, aximuthal and radial nonuniformities of the velocity field. Equations for determination of these characteristics are given in [3], where it is shown that at small distances from the axial-vane swirler the azimuthal velocity variation is of a wave nature with the number of maxima and minima being equal to the number of channels in the swirler. The highest nonuniformity of the velocity field is produced by swirlers with small angles of flow swirling.

Stability of swirled flows. In a swirled flow the influence of centrifugal forces on the flow structure is analogous in its character to the temperature stratification effect in a gravitation field. In this case, depending on the radial velocity distribution the kinetic energy of pulsational motion of the liquid particles may convert into the potential energy of dynamic stratification (conservative effect) and vice versa (active effect). By dynamic stratification in such a case is understood the liquid "stratification" in the field of centrifugal forces into regions with the active and conservative effect.

In [2, 3] the stability conditions for an ideal translational-rotational swirled flow are formulated. For the power law of flow swirling $\mathrm{tg}\,\varphi = \mathrm{tg}\,\varphi_{in}(R/r)^n$ the stability condition is formulated as follows:

$$[1 + 2n + (n + 1)\,\mathrm{tg}^2\,\varphi]/(1 + 2n + \mathrm{tg}^2\,\varphi) < 0.$$

From this equation one may determine n, φ_{in}, and the dimensions of the regions where a flow undergoes the active and conservative action of centrifugal forces. In [3] it is shown that at $n \ge 0$ for all φ_{in} values the mass forces are active, while for $n < -1$, depending on the $|n|$ and φ_{in} values, active, conservative, or mixed flow modes (conservative near a wall, active in the near-axial region) may be observed. By varying φ_{in}, n one may control the flow structure behind the swirler and change the heat transfer mechanism. It should be noted that at $\bar{x} > \bar{x}_*$ the flow completes its rearrangement: from the structure obeying the law preset by the swirler to that governed by the universal law $r\,\mathrm{tg}\,\varphi/R = \mathrm{const}$ [2].

Heat Transfer near a Swirler. The active and conservative character of centrifugal forces causes a complex change of the heat transfer coefficient near the swirler ($\bar{x} < \bar{x}_*$). Equations for calculation of local heat transfer for turbulent (with a conservative action) and transient flows are given in [2, 3]. Equations for heat transfer calculation in the region of the swirler itself are presented in [1-3].

Hydrodynamics of a Swirled Flow. To characterize the flow swirling intensity, different parameters are used [3]. In the same work direct experiments revealed that over the main section of the tube ($\bar{x} > \bar{x}_*$) the local $\mathrm{tg}\,\varphi_w$ and integral Φ_* swirling parameters represent additional similarity conditions of swirled flows in tubes. Generalization of experimental data for different methods of initial swirling points to an unambiguous relationship between these parameters, which is determined by the equation $\operatorname{tg} \varphi_w = 1.18\Phi_*^{0.7}$. The same relationship is fulfilled for porous cylindrical and conic convergent channels. This is explained by the self-similarity law $r \operatorname{tg} \varphi/R = \text{const}$ realized on the main section for all types of swirlers. As a corollary of this law, the important equation $\tau_{r\varphi}/\tau_{rx} = \operatorname{tg} \varphi$ follows which may be used in analytical and experimental studies.

The hydrodynamics of a swirled flow in tubes on complete and partial swirling at the inlet, with injection via a porous wall, and in annular channels is studied in detail in [1-3, 7, 14]. Based on the unified approach to generalization of experimental data and using the similarity parameter Φ_*, correlations are obtained in [3] for the axial and rotational velocity profiles, total and static pressure, the radius of the backward flow zone, the radii of zero static and total pressures, form-parameters, etc. Also, dependences are given which characterize the universal velocity profile in the wall flow region. To calculate the swirling intensity change along a tube as well as the change of integral parameters (moment and momentum), universal relations are proposed in [2, 3, 7] which are distinguished by a high accuracy.

The hydrodynamics of partial flow swirling for different conditions of its formation is thoroughly explored in [2, 3, 7]. These investigations revealed that a swirled jet in a channel is characterized by a higher ejection property as compared to an axial jet and the regularities of forming a flow in the near-axis region are determined by particular specific features of partial swirling generation.

The hydrodynamics of a swirled flow in porous tubes with an outlet diaphragm in annular channels is considered in [2, 3]. In particular, in annular channels different critical Reynolds numbers and angles of flow swirling on the inner and outer surfaces are noted. Specific features of the hydrodynamics in tubes with swirling of coaxial flows at the inlet in opposite directions are discussed in [3]. The hydrodynamics of a flow in vortex and cyclone chambers of different configuration is studied in [4-10]. It is found, in particular, that in a cyclone chamber with a control flow and swirling on the end conditions are realized under which particles of size 0.3 to 0.5 μm are retained for a long time on stable trajectories and are not carried away from the chamber. The specific hydrodynamic features in channels with screwlike oval tubes are discussed in [16].

Turbulent Characteristics. The field of turbulent characteristics in flow swirling is essentially three-dimensional and differs considerably from axial flows. Detailed results for a flow in tubes and annular channels, with injection, providing a diaphragm at the outlet, are considered in [2, 3, 7]. Turbulent characteristics in cyclone and vortex chambers are discussed in [5, 6].

As a rule, over the main section near the tube surface there occurs turbulence suppression (the conservative mode), and in the axial region - its development (the active mode). As a consequence of the alternating action of centrifugal forces the turbulent friction stress changes its sign over the channel cross section. The rotational components of turbulent viscosity change in the same fashion.

In cylindrical small-length channels ($l/d = 0.5$-2.5) for definite combinations of the degree of flow swirling and the channel length with flow efflux into a medium of the same nature (a gas into a gas) large-scale periodic pressure and velocity pulsations develop behind the channel. The minimal channel length at which these pulsations arise is $l_m = 0.91 \operatorname{tg} \varphi^{0.45}$ for $\operatorname{tg} \varphi > 0.65$. The fundamental frequency of pulsations f and the amplitude of velocity pulsations at a channel section are determined by the following equations:

$$fd/\overline{W} = 0{,}45 \operatorname{tg} \varphi \,(1{,}12 - 0{,}04l/d) \exp (0{,}01 \operatorname{tg} \varphi^{1,5}),$$

$$(\overline{W'^2})^{0,5} = (0{,}2 + 0{,}1 \operatorname{tg} \varphi) \operatorname{tg} \varphi,$$

where W is the average axial flow velocity.

Mathematical Models. To calculate internal swirled flows, both semiempirical and integral methods are used. The model of a constant turbulent viscosity gives no satisfactory results [3]. Better agreement with experimental data is attained when the model of a turbulent viscosity that is variable over the radius with an account of the streamline curvature effect on transfer characteristics as well as two-parameter turbulence models are used [2, 3]. Sufficiently accurate results are produced both by integral methods [2, 3] and by the asymptotic theory of boundary layer turbulence extended to such a type of flows [2].

Tube Heat Transfer in Full Flow Swirling at the Inlet. Heat transfer in tubes with flow swirling at the inlet is studied both by analytical and experimental methods. Analytical methods may be classified into two basic groups. The first group uses differential equations of motion and energy with different simplifying assumptions, while the second group comprises semiempirical hypotheses which take into consideration the basic specific features of a swirled flow (increase of the velocity near a surface, the curvature of streamlines, etc.). Relative swirling functions with different boundary conditions are discussed in [2, 3, 7]; the influence of a nonisothermal condition is studied in [7]. The most general case is analyzed in [2], where along with the curvature effect on the mixing length the spatial nature of turbulence is taken into account.

Experimentally, different methods of initial swirling with a relative tube varying from 4 to 150 are investigated. In [2], based on a unified approach, a great deal of experimental data is generalized, and the relative swirling function is represented by the equation $\Psi_\varphi = 1+m\Phi_*^p$, where for different swirling methods m varies from 0.57 to 0.66, and p - from 0.63 to 0.89. Partial flow swirling at the inlet, provision of a diaphragm in the tube outlet section, and injection of different gases through a porous wall are a matter of concern in [2, 3]. In [3] experimental data on the mean heat transfer in tubes with helical threading of different geometry, internal helical finning, and a wire-type swirler on the internal surface of the tube are reported. Also, the relative functions of swirling are given for a wide range of boundary conditions and it is shown that the generalized swirling function may be represented by the equation $\Psi_\varphi = 1+ 0.58(\Phi_*-0.1)^{0.8}$. The heat transfer law for tangential-slot-type swirlers at the inlet in a wide temperature range is discussed in [3].

Heat Transfer in Annular Channels. In annular channels the centrifugal forces on the inner surface have a stabilizing character, while on the outer surface these forces are active. The inner surface of an annular channel has two regions, namely, where $Nu \sim Re^{0.5}$ and $Nu \sim Re^{0.8}$. The former region is observed at a small swirling intensity ($\Phi_* < 0.5$); the heat transfer intensity here is higher than in the case of an axial stabilized flow. In the latter region ($\Phi_* > 0.5$), the heat transfer intensity corresponds, in fact, to an axial stabilized flow. The boundary of the Reynolds number between these regions depends on the initial swirling intensity and the ratio of channel diameters. On the outer surface of an annular channel $Nu \sim Re^{0.8}$. The similarity equations for the inner and outer surfaces of a smooth annular channel are given in [3]. Heat transfer in annual channels with inner helical tubes as well as in a twisted bundle of screw-type tubes is analyzed in [15]. The influence of regular roughness of the inner surface of an annular channel on heat transfer is considered in [3].

Heat Transfer in Cyclone and Vortex Chambers. A comprehensive analysis of heat transfer in cyclone chambers with one-sided and many-sided air supply is carried out in [5-7, 9, 10]. The similarity equations are given here for different chamber dimensions, number of inlets, and degree of diaphragming. With increase of the number of inlets (with their area being preserved), the ratio of inlet to cross-sectional area of the channel, and the degree of outlet section diaphragming the heat transfer is enhanced. The generalized similarity equation for the side surface of cyclone chambers of different design, including those with an internal cylinder, is derived in [10].

The regularities of heat transfer from an internal cylinder to a swirled flow in a cyclone chamber are discussed in [3, 10]. The similarity equations for a flow with "pressed" and "free" maxima are given in [10], and for thin bodies placed at the center of a vortex tube in [11]. Results of these investigations point to a high level of heat transfer due to unsteady-state and separation effects as well as to a high level of turbulence.

Depending on a degree of swirling the developing and developed flow modes may be realized on the face (side) surface of cyclone and vortex chambers. The former is characterized by the zeroth velocity in the flow core, while in the latter case the entire flow passes through the boundary layer. The boundary between these modes is determined by the degree of flow swirling, a gas flow rate, and the chamber geometry. A study of the hydrodynamics and heat transfer in a face three-dimensional boundary layer is the subject matter in [3, 7], where similarity equations are given for both flow modes. In particular, in [3] the similarity equation takes into account the separate influence of the radial gradient of the static pressure and the spatial structure of the boundary layer. Also, it is shown that the heat transfer intensity in the developed regime is, on the average, 40% higher than in the developing regime, but for both regimes $Nu \sim Re^{0.5}$.

Channels of Power Plants. Heat transfer for two-phase (gas-solid particles) flow swirling in a Laval nozzle is investigated in [3]. The presence of solid particles in a swirled flow causes a 6-8-fold increase of heat transfer. Equations for calculation of heat transfer in one- and two-phase flows are given in [3]. Swirled gas curtains and equations for their calculation are considered in [7], heat transfer in diffusor channels is investigated in [3], on boiling and condensation of a liquid, in a gas-liquid rotational flow in [3]. In particular, it is shown that flow swirling causes a 1.8-2.5-fold increase of the critical heat flux, while deterioration of heat transfer proceeds in the field of a vapor content close to unity. Mass transfer in a flow of swirled liquid films is analyzed in detail in [2, 3]. Mass transfer characteristics of coaxial gas flows swirled in different directions are presented in [3].

Intensification Factors and Thermohydraulic Effectiveness. The analysis made in [3] shows that heat transfer intensification in the case of a swirled flow in tubes is due to an increase of the flow velocity near the surface, the spatial structure of the flow, and the simultaneous influence of the curvature of streamlines and turbulence ε_*. The generalized equation for ε_* is as follows: $\varepsilon_* = 1 + 0.44(\Phi_* - 0.2)^{0.8}$, whence it follows that the model of streamline "smoothing" for heat transfer calculation in tubes may be used only at $\Phi_* \le 0.2$.

Swirling as a means for heat transfer enhancement in tubes is analyzed in [1-3]. In these works the conditions of energetically advantangeous use of fully and partially swirled flows, channels of different length, and inner finning and helical rolling of different kinds are determined. The effectiveness of swirled film flows of a liquid is discussed in [2, 3].

Practical Implementation. Swirled flows are shown to find application in vortex-type refrigerating machines, ejectors and pumps, drying chambers, gas burners, film apparatuses, rectifiers, classifiers, nuclear power and aviation facilities, heat exchangers, heaters, process apparatuses, etc. [3-5, 9-14]. Also, in these works the basic equations for calculation of different vortex apparatuses and specific features of new designs are presented.

The Goals of Further Investigations. The investigation of the heat transfer and hydrodynamics of swirled flows is far from being exhausted. In connection with this it seems reasonable to formulate the most important problems to be investigated:

1) heat transfer and hydrodynamics in two-phase (gas-solid particles, gas-liquid) flows;

2) thermal and hydromechanical unsteadiness in channels with different boundary conditions;

3) specific features of the transition from laminar flow to turbulent, viscous stability related with the active and conservative influence of centrifugal forces;

4) heat transfer and hydrodynamics in convergent and divergent channels, compressible flows;

5) considerable nonisothermality and the influence of the heat flux direction on heat transfer, friction, and the structure of the near-wall flow;

6) the turbulent structure in channels and chambers, including under nonisothermal conditions;

7) development of semiempirical calculation methods taking into account the basic specific features of turbulent transfer in the field of centrifugal forces and, especially, of the three-dimensional nature of turbulence;

8) interdependence of the structures of different groups of swirled flows and elaboration of sufficiently general calculation methods for separate groups of swirled flows.

Problems concerned with swirled flows in vortex- and cyclone-type chambers are discussed in [6, 8, 10].

NOTATION

$d = 2R$, channel diameter; K, flow momentum; M, moment of flow motion; Nu, Nusselt number; r, current radius; R, surface radius; Re, Reynolds number with respect to a characteristic dimension; Re_T^{**}, Reynolds number with respect to an energy loss thickness; $\bar{x} = x/2R$, relative length; φ, angle of flow swirling on radius r; φ_{in}, angle of initial flow swirling on radius R; $\tau_{r\varphi}$, τ_{rx}, tensor components of turbulent tangential stresses; $\Psi_\varphi = (St/St_0)_{Re_T^{**}}$, relative function of swirling.

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