Investigation into aerodynamic and heat transfer of annular channel with inner and outer surface of the shape truncated cone and swirling fluid flow

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Abstract. We have carried out Investigation into aerodynamic and convective heat transfer of the annular channel. Inner or outer surface of annular channel has shape of blunt-nosed cone tapering to outlet end. Truncated cone connects to a cyclone swirling flow generator. Asymmetric and unsteady flow from the swirling generator in the shape of periodic process gives rise to the formation of secondary flows of the type Taylor-Görtler vortices. These vortices occupy the whole space of the annular channel, with the axes, which coincide with the motion direction of the major stream. Contraction of cross-sectional area of channel (in both cases 52%) causes a marked increase in total velocity of flow, primarily due to its axial component and promotes a more intensive vortex generation. Vortex structures have a significant influence on both average heat transfer and surface distribution. At cross-sections of the annular channel we observe similarity of curves describing distribution of total velocity about wall and heat flux density on the surface. The coordinates of maximum and minimum values of velocity and heat flux coincide. At the average cross-section channel of maximum value of heat transfer is greater than minimum of about by a factor of 2.7 times for outer heat transfer surface and about by a factor of 1.7 times for inner heat transfer surface. Taper channel has a much higher influence on heat transfer of the inner surface than the outer surface and manifests itself at lower values of dimensionless axial coordinate. For the investigated taper cone geometry of the annular channel the heat transfer coefficient of inner surface increases at the outlet section and exceeds value in comparison with straight-line section by 91 ... 98%. Heat transfer of the outer cylinder in the same section increases only by 5 ... 11%. The increase in average heat transfer over the surfaces is 36% and 4% respectively.

1. Introduction
The use of swirling flows in technology is related to solving the intensification problem of work processes in power-producing units, motor systems and heat exchangers [1]. Swirling motion is a simple and effective way of heat exchange intensification in annular channels of recuperative devices, it represents thermal protection of energy intensive sections and equalize uneven temperature distributions. Local swirling of airflow based on tangential infeed in annular channel by means of cyclone generator, as the completed authors’ researches show [2], establishes substantial intensification of heat transfer, especially on its external surface. However, even if the flow is injected in a swirling generator from two diametrically opposed directions, measurable azimuthal unevenness is observed on a quite extended section of entrance length of the adjacent channel. Heat transfer on
both surfaces is decreasing towards the outlet end due to intensity reduction of flow’s swirling motion, which has a negative impact on operational reliability of recuperative devices. From that point of view, heat transfer control with its intensification in thermally stressed zones of recuperative devices’ construction, based on detailed study of aerodynamics and heat transfer, is an urgent task.

2. Research method
Numerical study of aerodynamics and heat transfer is conducted for both: a straight annular channel and an annular channel tapering to outlet end (Figure 1, \( z – \) axial coordinate, count off from swirling generator along the axis of annular channel). The annular channel has length \( l \) of 840 mm, inner diameter of the annular channel \( d_1 \) equals to 152 mm, outer diameter \( d_2 = 184 \) mm. Cylinders 1 and 5, which change channel 4 geometry, are made in the form of truncated cones. In the first variant (fig. 1a), channel’s tapering is accomplished through reduction in diameter of outer cylinder 1 from 184 to 168 mm; in the second variant (fig.1b), it is accomplished through inner diameter 5 increase from 152 to 169.9 mm. In both cases cross-sectional area decreased by 52%. Swirling generator 2 is a hollow, smoothbore cylinder with inner diameter \( D \) of 259 mm and length \( L \) of 126.5 mm, in which air is delivered tangentially to inner surface through channel 3, which internal dimensions are 35·70 mm (the larger size is along the cylinder generatix). The relative area of an inlet duct \( f_{in} = A_{in}/\pi D^2 \) equals to \( 5 \cdot 10^{-2} \).

![Figure 1. Tapering annular channel with cyclone swirling flow generator.](image)

Numerical simulation is conducted in three-dimensional shape through the instrumentality of ANSYS Fluent 15.0. The flow is described through nonstationary Navier-Stokes equations of continuity and energy that are Reynolds-averaged. Reynolds equations closure is performed using both: a two-parameter turbulence model SST (Shear Stress Transport) \( k-\omega \) with curve correction and standard wall functions, and a unsteady Reynolds stress transfer model RSM (Reynolds Stress Models). The calculation is conducted in structured hexahedral mesh with 9 million meshes, while its minimum (at the surfaces \( y^+ = 1 \)) and maximum dimensions are \( 2 \cdot 10^{-6} m \) и \( 3 \cdot 10^{-3} m \) respectively. Results verification of numerical simulation of aerodynamics and heat transfer is carried out based on empirical data, obtained from the model of straight annular channel with similar geometrical dimensions and unilateral heating of its walls [2].

3. Research findings
The unilateral gas injection in a swirling generator determines measurable azimuthal unevenness of total velocity and all of it components, which results in unsymmetrical flow efflux into the annular channel. Swirling generator’s aerodynamics and conditions of the flow efflux do not change if outer and inner cylinders of annular channel are made in the form of tapering blunt-nosed cone (for the studied geometrical characteristics).
Distributions of nondimensional total velocity $\overline{V}$, its tangential $\overline{w}_\phi$ and axial $\overline{w}_z$ components from nondimensional radius $\eta = (r-r_1)/(r_2-r_1)$ in cross-section of the annular channel are presented in figure 2 ( $\overline{V} = V/V_{in}$, $\overline{w}_\phi = w_\phi/V_{in}$, $\overline{w}_z = w_z/V_{in}$, $V_{in}$ – average flow velocity in the inlet vent, in the calculations is equal to 37 m/s ; $r_1$ and $r_2$ – current values of inner and outer radius). Velocity fields are presented at axial coordinate value $z = 800$ mm and angle $\varphi = 0^\circ$ ($\varphi$ - central angle, counted in cross-section towards the flow direction from the junction point of inlet vent and the inner surface, figure 1).

In the tapered channel, in contrast to the straight one, reduction of cross-section area towards the outlet end leads to increase of total flow velocity due to growth of its axial component. At mid-radius of channel’s cross-section ($\eta = 0.5$), total velocity is increasing by 20 and 31% while axial velocity by 200 and 209%. In this setting, the lower values are corresponding to the variant $b$ (fig.1). Decrease in flow’s swirling along the length of channel occurs due to both: reduction in tangential velocity and rise in axial component. The intensive drop of flow’s swirling with cylinder’s widening is presumably connected with larger degree of channel’s relative reduction. For instance, in the variant $a$ relation $d_1/d_2$, at the channel’s outlet equals to 0.905 while in the variant $b$ – 0.921.

Contraction of cross-section area of channel in both variants causes increase of device’s resistance. Table 1 represents values of total resistance coefficients of annular channel with swirling generator $\xi_E$ and its components: swirling generator $\xi_{\varphi}$, annular channel $\xi_{ch}$ and its outlet $\xi_{out}$. The values are calculated according to the formula $\xi = 2\Delta p/pV_{in}^2$ ($\Delta p$ - total pressure drop at the estimated area).

| Variant          | $\xi_E$  | $\xi_{\varphi}$ | $\xi_{ch}$ | $\xi_{out}$ |
|------------------|----------|-----------------|------------|-------------|
| Straight channel | 3.30 (2.94) | 1.20 (1.24)    | 1.74 (1.25) | 0.36 (0.45) |
| Variant $a$ in figure 1 | 3.90      | 1.18            | 2.03       | 0.69        |
| Variant $b$ in figure 1 | 3.77      | 1.19            | 1.95       | 0.63        |

Resistance values for the straight annular channel, calculated according to empirical velocity and total stress (measured at the angle $\varphi = 0^\circ$) distributions, are shown in parentheses. Discrepancy between the values, obtained through numerical calculations and experimental data, can be presumably explained by the fact that in the latter case azimuthal unevenness of total stress and velocity in the cross-sections are not taken into account. Two variants of channel’s tapering have no significant impact on resistance...
of swirling generator. The increase of device’s total resistance by 18% in the variant \(a\) and by 14% in the variant \(b\) occurs due to growth of its components \(\xi_{ch}\) and \(\xi_{out}\). Large-scale unsteady transverse vortexes are developing from the leading edge of the outer cylinder of annular channel as a batch process, which can be characterized by Strouhal numbers around 0.022. Isotach distributions of total velocity as well as axial velocity fields of flow on entering annular channel at \(\varphi = 90^\circ\) at the time moment differing by a half period are presented in figure 3.

![Figure 3](image-url)

**Figure 3.** Distribution of total and axial velocity at the flow’s entry in the annular channel at multiple time points.

Bending of vortex axes and their flow orientation leads to the structures formation in the mould of Taylor-Görtler vortices, which gradually occupy the entire annular space, with the axes, which coincide with the motion direction of the major flow. The amount of formed vortexes is increasing towards the outlet section. Isotach distribution of radial velocity in cross-section of the straight annular channel at axial coordinate value \(z = 800\) mm are presented in figure 4. There are observed approximately 14 well-shaped vortexes in this cross-section, while new vortexes keep emerging.

![Figure 4](image-url)

**Figure 4.** Distribution of radial velocity in cross-section of the straight annular channel at \(z = 800\) mm.

![Figure 5](image-url)

**Figure 5.** Distribution of radial velocity in cross-section of the tapered annular channel in the variant \(a\) at \(z = 800\) mm.
Formation of secondary vortexes occurs earlier in the tapered channels, comparing to the straight one, so at the cross-section $z = 800$ mm there are observed from 24 to 26 secondary vortexes with substantially reduced intensity. In that case, interchange of large vortexes with the smaller ones is encountered (figure 5).

Total velocity distribution in cross-section of the tapered channel in the variant $a$ at equal distances of 2 mm from the inner surface (curve 1) and outer surface (curve 2) are given in figure 6. Nature of total velocity change can reveal close agreement between angular coordinates of maximum values on curve 1 and minimum values on curve 2 and conversely. Interchange of maximum and minimum velocity values near the surface is explained by the fact that secondary vortex-type flows transport particles, which have higher velocity, from the flow to the wall, while particles with lower velocity values are carried from the wall towards the flow.

Intensification of heat transfer on the surfaces of tapered annular channels occurs as a result of flow’s velocity increase, formation of secondary vortex-type flows and their influence on turbulence.

Analysis shows that correlation between the stated factors and heat transfer of inner and outer surfaces can differ widely. Figures 7 and 8 (for the variant $b$ in cross-section of channel $z = 800$ mm) demonstrate distributions of total velocity and effective viscosity $\mu_{ef}$ at a distance of 2 mm from channel’s surfaces, along with heat flux density $q$ on them.

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Let us highlight the variation peculiarities of the stated characteristics. According to the investigated variants, channel’s tapering leads to increase of average total velocity value at the outer surface by...
24...27% and at the inner surface by 57...60%. At the same time, total velocity fluctuations are considerably larger at the outer surface (±11%), rather than at the inner surface (±5%). Effective viscosity near the outer surface is about 24% higher than its value near the inner surface, and changes insignificantly in the azimuthal direction (±8...10%). Average value of effective viscosity near the inner surface in the variant a is 39% and in the variant b 100% higher than in the straight channel. In contrast, \( \mu_{ef} \) is decreasing by 6...10% near the outer surface. Near the inner surface \( \mu_{ef} \) has maximum deviation against the average value (±36%), and vibration frequency is substantially higher than its value near the outer surface.

The maximum heat flux density in the channel’s cross-section exceeds the minimum value by around 2.7 times for outer and by around 1.7 times for inner surface. The curves, which describe heat flux density variations along the perimeter of outer surface, are in reasonable agreement with total velocity distributions near the outer surface and with effective viscosity for outer surface. Presumably, heat transfer intensification on the outer surface of tapered annular channel is determined by increase of total velocity components – axial and radial, which is caused by secondary vortexes, as well as by increase of velocity and flow turbulence on the inner surface.

The change along the length of averaged cross-section coefficients of heat transfer for the straight and tapered annular channel is presented in figure 9. Reduction of cross-section area greatly influences heat transfer of inner surface and is evident at substantially smaller axial coordinate \( z \) values. For the investigated tapered channels heat transfer coefficient of inner cylinder begin its growth at \( z > 170 \) mm and nearby the outlet section is by 91...98% higher than in straight channel. Heat transfer of outer cylinder in the same cross-section increases only by 5...11%. Increase in average heat transfer coefficient on the whole inner surface is approximately 36% and on the outer surface – 4%.

![Figure 9. The \( \alpha \) change along the length of inner (1, 2, 3) and outer (4, 5, 6) surfaces of straight (1, 4) and tapered annular channels in variants a (2, 5) and b (3, 6). Dots – experimentation [2].](image)

4. Conclusions

Formation of secondary vortexes in the mould of Taylor-Görtler vortices occurs earlier, if cylinders of annular channels with swirling flow are made in the form of tapering blunt-nosed cones, comparing to the straight channel.

Intensification of heat transfer on the outer surface of annular channels occurs due to increase of total velocity components – axial and radial, which is caused by secondary vortexes, as well as by increase of axial velocity and flow turbulence on the inner surface.

Heat transfer of outer cylinder increases nearby the outlet section and in the studied tapered channels is by 91...98% higher comparing to the straight channel, heat transfer of the outer cylinder in the same cross-section increases only by 5...11%.

References

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