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NEWER METHODS OF VIBRATION DAMPING IN WIDE ANGLE DIFFUSERS OF GAS TUBINES

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The paper deals with newer methods of dynamical load decreasing which impact on the outer walls of diffusers with large channel opening angle(15-20°) and divergence ration n=4.Two methods of impact on flow pattern in wide angle diffusers are under consideration:1) Placing of wedge-shaped ribs on the on the streamlined surfaces;2) Safeguard of streamlined surface from dynamical load at the expense of near-wall punched screen. Placing of mentioned above additional elements in the diffusor channel allows to decrease wall vibration in 2-3 times with insignificant decrease of pressure recovery factor.

Keywords: annular diffusor, wide-angle diffusor, punched screen, wedge-shape ribbing

Introduction

Consideration of steam and gas turbine efficiency increase methods shows that untapped reserves of efficiency increase at the expense of further blading improvement extremely limited.

Thereupon interest in the problem of turbine exhaust flow kinetic energy utilization increase. The amount of kinetic energy reaches rather high values.

So exhaust gas equivalent power for gas turbine V94.3A equals to 25MW at turbine total power 550MW. In the steam turbine these loses are lower and reaches 2-3% of turbine total power.

Natural way of this energy utilization is its' transformation to potential energy at the expense of flow velocity reduction by means of diffuser system application.

In turbines energy transformation is realized in geometrical diffusers where at the expense of velocity subsonic flow reduction in the divergent channels take place the pressure growth along flow direction.

Pressure reduction rate after last turbine stages is defined by transformation effectiveness of kinetic energy to potential pressure energy in the diffuser. Transformation effectiveness is characterized by coefficient of overall loss ζo or diffuser efficiency coefficient ηd or by coefficient of pressure recovery ξ which is equivalent to energy recovery coefficient. Physical interpretation of these parameters is given in [1].

In case of diffusor installation internal power capacity growth is defined by the following expression [1,2]:

$$\Delta N_i = G \cdot \Delta h_z \cdot \eta_{oi} = G \cdot \xi \cdot \frac{C_z^2}{2} \cdot \eta_{oi} \qquad (1.1)$$

If $N_i = G \cdot H_0 \cdot \eta_{oi}$ is turbine power capacity or power capacity of low pressure cylinder in case of steam turbine that efficiency increment specified by diffusor application is equal to

$$\Delta\eta_{oi} = \frac{\Delta N_i}{N_i} = \frac{\xi \cdot C_z^2}{H_0} \qquad (1.2)$$

For the gas turbine expression 1.2 may be transformed to

$$\Delta\eta_{oi} = M_z^2 \xi (k-1) \frac{\tau}{1-\tau}, \qquad (1.3)$$

where k – isoentropic factor, M – Mach number, $\tau = \frac{T_a}{T_0}$, T_0 – initial gas temperature, T_a - ambient temperature.

Diffusors geometry significantly impact on dynamical load magnitude which acts on their walls.

In consideration of the diffusers practical application in turbomachinery it is necessary to ensure high reliability characteristics of turboinstallations.

This requirement led to hard constraint on the divergence angle of the annual diffuser outer contour which values exceed 10° seldom. As a result even at divergence ration of the diffuser flow path n equal to 2 its' axial length exceed axial length of gas turbine and the energy recovery coefficient ξ is 45-50%.

In case of α1 increase up to 15° and divergence ratio n up to 4 energy recovery factor growth up to 70% at the same diffusor length. But its' vibration reaches dangerous magnitude what result from the appearance of non-stationary flow separation from the diffusor's walls. Mentioned flow regime is accompanied by sharp growth of the pressure pulsation magnitude which is defined the level of dynamical load sensible by diffuser walls. External evidence of the passage to separation flow regime is the heightened diffuser vibration.

For its' decreasing two new methods were applied. First method is based on the direct impact on the flow by means of wedge-shaped ribbing application of the streamlined surface. The second method is based on the diffusor walls protection from the direct contact with non-stationary flow at the expense of the punch screen installation.

Let's consider how mentioned methods of dynamically load which impact on the streamlined surfaces decrease can promote to increase vibrating reliability of the annular diffusers in case of different velocity fields in their inlet sections(have in mind the flow twist).

2. Annular diffusors with longitudinal wedge-shape ribbing vibrating condition research

Carried out research connected with searching of ribs optimal shape allows to bring out that zone of active geometrical impact on the moving in the channels working medium must be limited to boundary layer. In case of turbulent flow the boundary layer thickness along streamlined surfaces growth almost by the linear law.

Then in case of ribbing surfaces application ribs height must increase downstream by the linear law and have a wedge shape with the root height in the diffusor outlet section commensurable with boundary layer thickness δ.

In case of non-gradient flow the thickness of boundary layer δ is defined by the following expression: $\delta_{max} = \frac{0{,}37L}{Re^{0{,}2}}$, (1.4), where L – the length of the streamlined surface, and

Reynolds number is estimated by the velocity c in the channel inlet section and length L ($Re = \frac{cL}{\nu}$).

In the diffuser maximal thickness of boundary layer in the outlet section appears by 30-40% more. However friction forces in the area of boundary layer outer border appears small and for the estimation of rib height in the diffusor outlet section it make sense to use reduced expression 1.4. So the root height of near-wall wedge-shaped ribs h may be estimated by the following expression: $h_{max} = \frac{(0{,}3 \div 0{,}4)L}{Re^{0{,}2}}$, (1.5)

The angle β at vertex of wedge-shape rib is equal to: $tg\beta = \frac{0{,}3 \div 0{,}4}{Re^{0{,}2}}$, (1.6)

Mentioned reasons and expressions were laid in the base of ribbing of considering annular diffusor with diverging angle α_1=15° and divergence ratio n=4 (ribs angle pith θ=5°).

Research results of such diffusor are presented on the figure 1 as a function of full losses $\xi_п$ from twisting angle of the flow φ in the diffusor inlet section. Presented characteristic shows that near-wall wedge-shaped ribbing application doesn't provoke significant growth of additional energy losses.

Important practical interest of this research is a fact of high efficiency retention of ribbed diffusor not only in case of axial working substance entrance (φ=0) but in case of significant flow twisting in the inlet section too(φ≤20°).

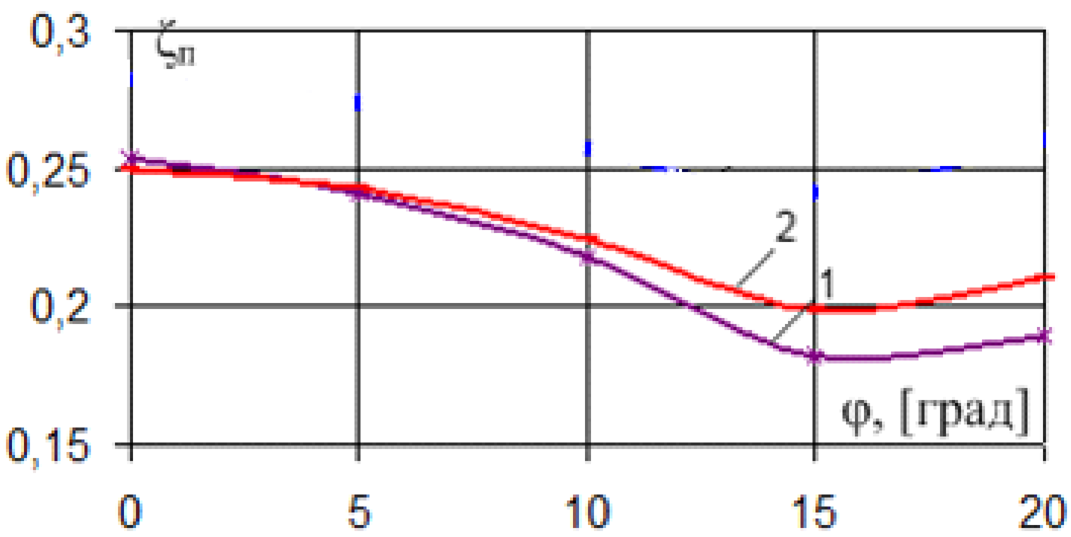


Fig.1 $\zeta = f(\varphi)$: 1- α_1 = 15° n = 4, 2-α_1=15° n=4

Results which were received in case of near-wall wedge-shaped ribbing application are indicated of successful solving of the first part of the problem – ensuring of full losses low coefficient.

The ribbing under consideration successful solve second(main) part of the problem – problem of outside diffusor wall vibration decrease. This fact is confirmed by the vibration tests. The results of test mentioned above are presented on the figure 2 as function of relative vibration velocity(fig. 2a) and relative vibrodisplacement(fig. 2b) from the flow twist angle φ in the inlet section of the diffusor.

All vibration velocity C and vibrodisplacemetn δ are related to base values C_0 and δ_0. The base values are the one for the diffusor with outer contour divergence angle $\alpha_1 = 7°$ and divergence ration of the channel n=2.

Near-wall ribbing of streamlined surface allowed to decrease vibrating velocity magnitude in 2,8 times and to decrease the value of vibrodisplacement in 2 times(curves 4 on the figure 2 a and b) in comparison with non-ribbing diffusor(curve 2) in case of axial flow inlet(φ=0°). At flow twist presence vibration decreases on all diffusors under consideration. When φ equals to 15° vibration velocity decreases in 3 times and vibrodisplacement in 1.7 times.

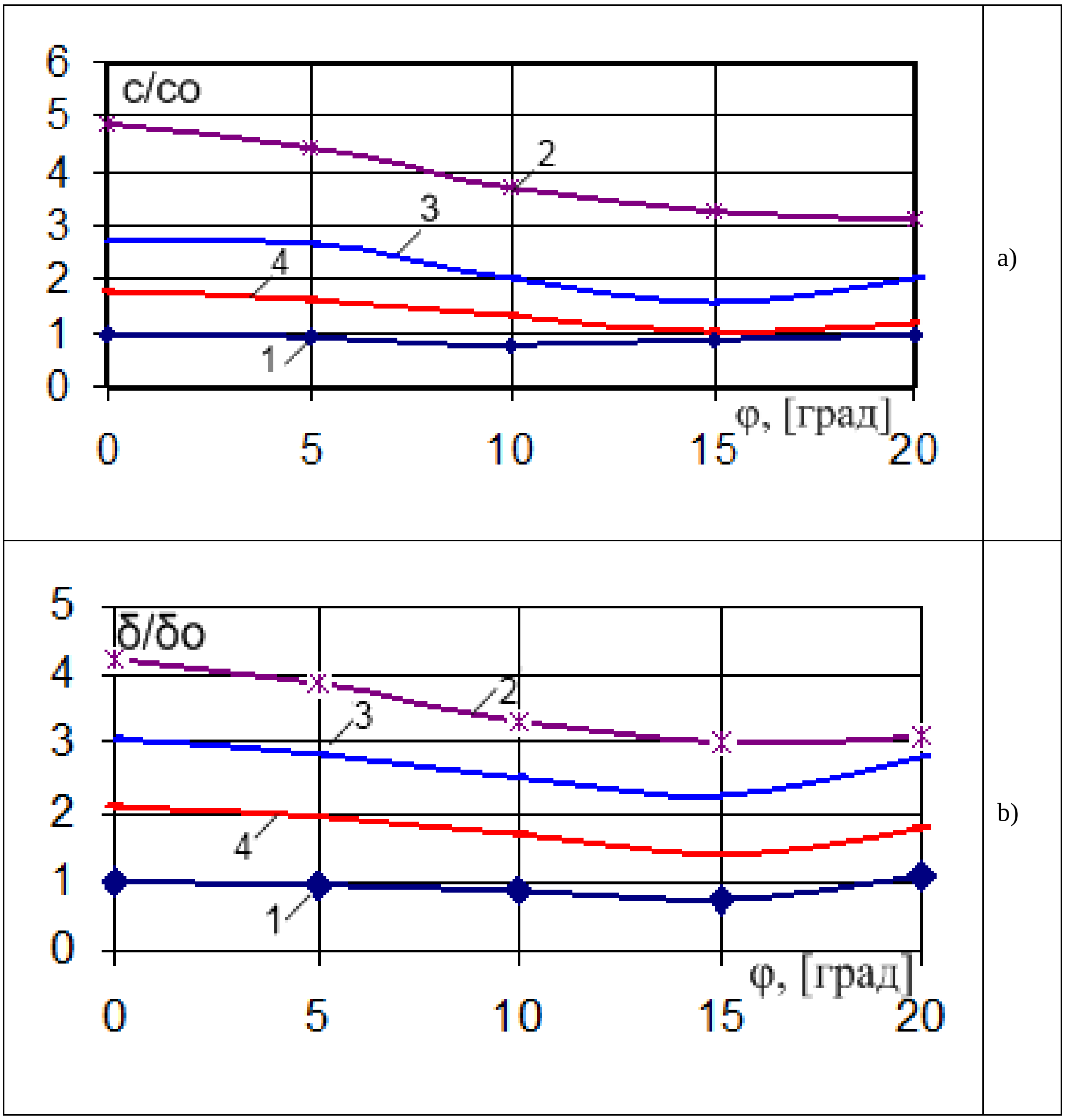


Fig. 2. Changes of vibration characteristics as a function of flow twist at the diffusor inlet, a – vibrating velocity, b - vibrodisplacement: 1 - α_1=7° n=2, 2 -α_1=15° n=4, 3 - α_1=15° n=4 trapezium-shape ribs, 4- α_1=15° n=4 wedge-shape ribs.

3. Aerodynamic and vibration characteristics of wide angle diffuser with punched screen ($\alpha_1 = 7°$ n=4)

Among the methods ensuring cardinal decrease of dynamical loads which act on the diffuser walls the method of pulsation damping near the walls at the expense of near-wall punched screens application provoke big interest.

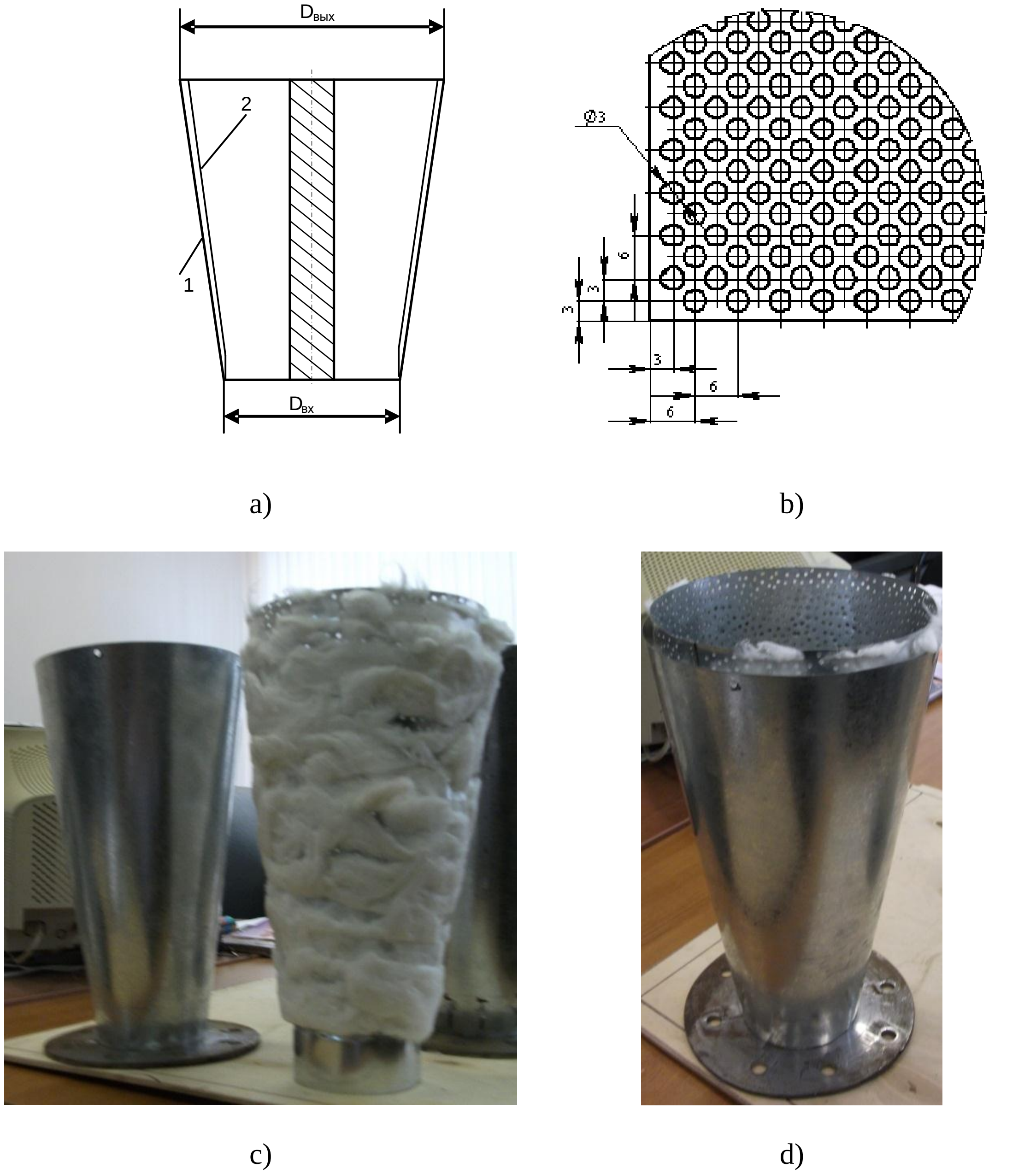


Fig.3 Annula diffusor with punched screen:
a –annula diffusor with installed internal conical punched screen(1 – outer surface, 2 – puched conical screen); b – scheme of screen punching; c- components of the diffusor with punced screen and wadding staffing; d – bodily diffusor model with punched screen(without internal hub)

As applied to tubes such method of walls protection from dynamical loads conditioned by high magnitude of pressure pulsation in the flow was considered in 2001[patent № 23000444 by Zarynakin A.E., Istomin S.A., Chernoshtan V.I.].

Components of such wide angle diffuser and bodily diffusor are shown on figure 3(n=4, α_1=15°).

Direct application of the punched screens in annular diffusers appears impossible because of secondary flow beginning in the clearance between the wall and screen. This fact almost destroys diffusor effect. For secondary flow avoidance the gap was fill by silicate cotton (figure 3).

Such diffusor test at working substance axial flow shows that accepted method of secondary flows prevention in the near-wall area allowed to safe high diffusor effect . Full losses coefficient $\zeta_п$ equals to 0.296 and by 4.5% exceed this value in the same diffusor without near-wall punched screen.

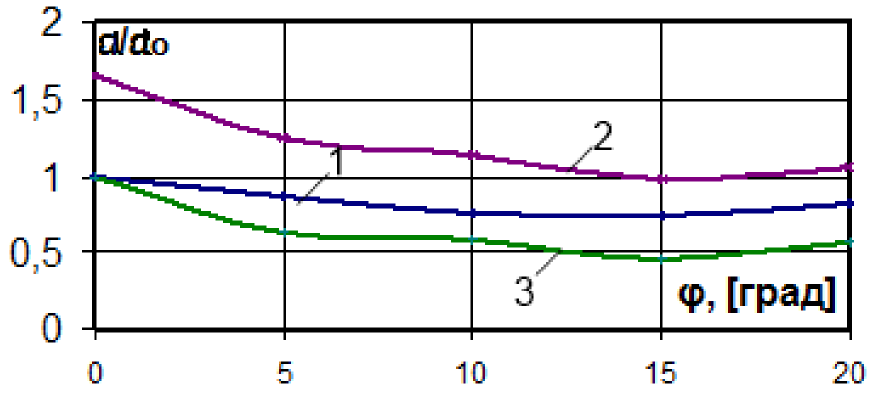
All vibration characteristics of the diffusor under consideration are considerably decreased in comparison with smooth-wall diffusor. So vibration acceleration in the diffusor outlet section reached the level of the diffusor with divergence angle 7° and divergence ration n=2 and occurs in 1.7 times lower than in the same diffusor (α_1=15° n=4) without screen and vibration velocity decreased more than in 2 times.

Research results of the diffuser with near-wall screen in case of flow twist presence in the inlet section also indicate of high damping property of near-wall screen with filling gap between the wall and screen by mineral cotton.

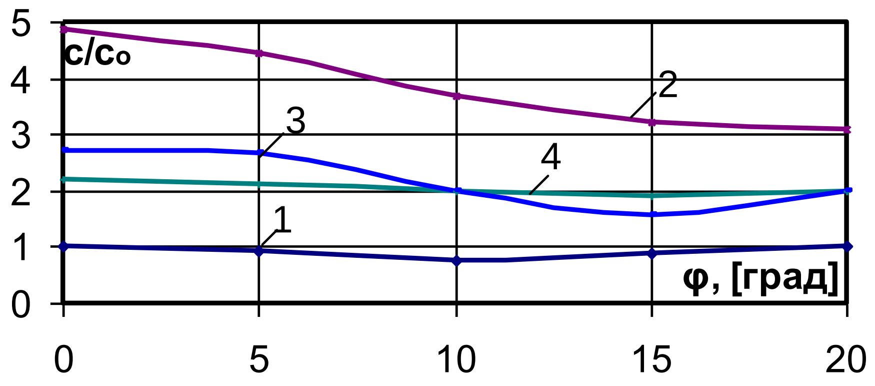
Presented on the figure 4a dependencies of relative vibration acceleration $\overline{a} = a / a_0$ (a – measured vibration acceleration, a_0 – vibration acceleration on the diffusor without flow separation with divergence angle $\alpha_1=7^0$ and divergence ratio n=2) from the twist angle in the diffuser inlet section show that this value decreases when the swirl angle φ rises for of diffusor under comparison.

For the base diffusor (α_1 = 7°, n = 2) in present test series (curve 1) maximal decrease of the relative vibration was obtained at φ = 15° and equals to $\Delta\tilde{a} = 25\%$. For investigated diffusor (α_1=15°, n=4) vibration acceleration occurs lower in comparison with base values in all cases of flow swirling and for φ=15° its' magnitude decreased in to times relevant to non-twisted (φ=0°) axial entrance of the working substance (curve 3). For the diffuser with smooth surface (α_1=15°, n=4) as expected vibration acceleration at all angles of flow swirling occurred considerably higher than base values (figure 4, curve 2).

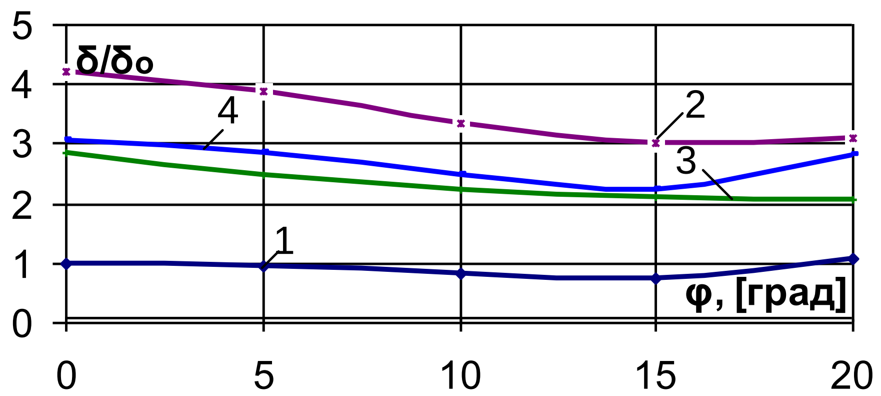
For this diffusor vibration acceleration decrease intensity which determines dynamical load level occurs maximal because of in case of working medium axial entrance the flow in the near-wall area separates from the walls. When swirling flow moving in the channel the centrifugal force not only eliminates flow separation from the streamlined surface of outer contour but it is strong pulsation damper.



a)



b)



c)

Fig 4. Vibration characteristics as function of flow swirling at the diffusor inlet: *a* - vibration acceleration, b- vibration velocity, c- vibrodisplacemetn
1– α_1=7° n=2, 2– α_1=15° n=4, 3– α_1=15° n=4 with screen, 4- α_1=15° n=4 with trapezium-shape ribs.

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