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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machines

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poświęcone są publikacjom naukowym z zakresu teorii i badań doświadczalnych w dziedzinie mechaniki i termodynamiki przepływów, ze szczególnym uwzględnieniem problematyki maszyn przepływowych

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Numerical approach to study the non-stationary flow phenomena in exhaust hood compartment – last stage of steam turbine

The paper describes 3D numerical models and some results of numerical simulation of self- excited oscillatory flow regimes through exhaust diffusers of large steam turbines operating jointly with last stage.

1. Introduction

It is known, that the flow through the last compartment is essentially non-stationary due to stator-rotor interaction, rotor diffuser axial asymetrical interaction and, for certain regimes, due to self-exited diffuser flow oscillations (Maier, 1989).

It is convenient to describe the flow nonstationarity by the amplitude-frequency composition of integral and local parameters obtained as an expansion into Fourier series of the harmonics with frequencies being multiple of the rotation frequency. So, the non-stationarity in the stator channel is determined by the main harmonic with nz_2 main frequency due to stator-rotor interaction and by revolution main harmonic with n-main frequency due to stage axial asymmetry of the exhaust hood interaction. The non-stationarity in the rotor channel is determined by main harmonic with nz_1 main frequency, by revolution main harmonic with n-main frequency and by diffuser self-exited oscillation spectrum frequencies. This last one can effect also on the non-stationarity in the stator.

Diffuser flow may contain all these types of non-stationarity. Namely, flow in the diffuser is nonstationary due to the stator-rotor interaction effect, but has self-exited oscillations of flow pattern whose spectrum is not multiple to rotor revolution frequency and depends on operational regime and geometry of compartment.

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The paper represents the model of the exhaust compartment, which includes the mechanism of stator-rotor aerodynamical interaction, the mechanism of stage-exhaust hood interaction and describes low-frequency flow instability in the exhaust diffuser.

2. Formulation of the problem

The inviscid and non-heat conductive gas flow through the domain of last compartment of axial turbine is considered. The last compartment domain consists of the last stage and the exhaust hood including the axial-radial diffuser and the space under casing (exhaust chamber) (Solodov, 1994).

The flow is described by non-stationary 3D Euler equations used in the conserved integral form. Integration of Euler equations is carried out by means of a numerical method with use of explicit finite-difference Godunov's scheme of 2^{nd} order accuracy both in time and space. The difference scheme uses the Riemann's problem for calculation of fluxes through the surface of finite difference division of computational region. The 2^{nd} order accuracy for spatial directions is ensured by use of the linear distribution of gasdynamical parameters in cell on the base of minimum principle (Kolgan's extension for exhaust hood domain, TVD and ENO extensions for the diffuser domain). The 2^{nd} order accuracy for the time variable is reached due to the use of extrapolation procedure along the time coordinate.

The computational region of a stage consists of one channel of stator and one chanel of rotor and the computational region of exhaust hood consists of volume of axial-radial diffuser and exhaust chamber.

The coupling procedure beetween this different scale volumes firstly proposed by author and described in details in (Solodov, 1994; Solodov et al., 1995) consists in quasi-static approach, which is based on experimental data about sufficiently smooth character of circular distributions of gasdynamical parameters in the stage-diffuser axial spacing. This fact permits to approximate them by the piecewise-constant functions. For every interval of constancy the flow in the stage fragment is calculated with use of generalized space-time periodical boundary conditions. The general solution for fixed time point in the stator and rotor domain is given by the superposition of solutions for all stage fragments on the interval of piece-wise constancy. Values of the parameters in the inter-stage spacing are the inlet boundary conditions for the flow calculation through the exhaust hood. The general interaction problem is solved by the global iterations in a common time scale with all gasdynamical conservative variables transmission from stage to exhaust hood and back for each time iteration. The receipt of a periodical flow in stage and stable one in the exhaust hood is based on the time-marching procedure. Proposed approach permits to realize the multiscale interaction between stage and the exhaust hood.

Neglecting circular gradients of parameters in the inter-stage spacing leads to a full stationary model of rotor and exhaust hood interaction, which is rigorous in the case of stage-axisymmetrical diffuser interaction and may be used to characterize the radial parameter distributions in the inter-stage spacing.

The general model permits to reveal and separate high frequency (HF) non-stationary phenomena from stator-rotor interaction and low frequency nonstationarity due to axial asymmetry of diffuser or exhaust hood. Low frequency (LF) non-stationarity because of a moving shock wave has a frequency spectrum with *a priori* unknown range and values of main harmonic frequencies which depend on inlet swirl, regime, geometry of outer shell etc. These frequencies are not multiple to the rotational one, so all the advantages of quasi-static stage-hood joint model vanish and self-exited oscillations cannot be correctly described by this model.

The phenomenon can be evaluated by use of the simple stage-diffuser model (in view of the axially symmetrical one). This model contains HF nonstationary flow phenomena and LF self-exited non-stationarity for every diffuser section.

3. Results and discussion

Non-stationary phenomena in the compartment have been studied for isolated diffusers in axisymmetrical formulation, in full 3D formulation and for compartments "stage-diffuser", "stage-exhaust hood". Some preliminary results for isolated diffusers had been published in [2-5].

Typical meshes used in the numerical simulations contain near 50×25 cells in the meridional diffuser section. The methodic study for 2^{nd} order scheme has shown that predicted results weakly depend on mesh dimensions for 40×20 and more fines. The dispersion scheme errors in calculations were conserved under frozen parameter derivatives in the limits of 5 time steps of explicit scheme. The TVD and ENO approximations of parameters in cells weakly effect integral non-stationary amplitude – frequency flow characteristics of the rate, aerodynamic force etc., and amplitudes of their main harmonics.

4. Axisymmetrical flow in a circular diffuser

The simplest model diffuser to be investigated numerically, whose contours are formed by concentric circle quarters with radii r and $R(R/r = 3.0)$ has the area ratio $n = F_{outlet}/F_{inlet}$ within the limits $1.0 < n < 2.0$.

Flow through the model diffuser under uniform inlet and outlet conditions without any inlet swirl and for the pressure ratios $\varepsilon = P_{out}/P_{tot} < 0.9$ has a supersonic region in the outer shell locked by the shock wave. In the regimes $0.5 < \varepsilon < 0.8$ the oscillatory phenomena occur in the flow, which are characterized by a low-frequency inlet and outlet rate oscillations, accompanied by periodical flow separation behind the shock and a formation of the recirculation zone near the outer diffuser shell. Flow separation on the outer diffuser shell arises because of strong shock interaction and artificial viscosity inherent in finite-difference Euler gas model. Evolution of oscillatory process for diffuser family depending on area and pressure ratios n, ε is given in Fig. 1-3.

Decrease of the pressure ratio causes the gradual decrease of oscillatory period

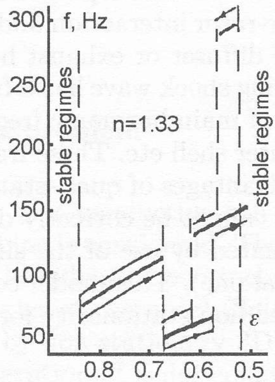


Fig. 1. Frequency hysteresis for the model diffuser.

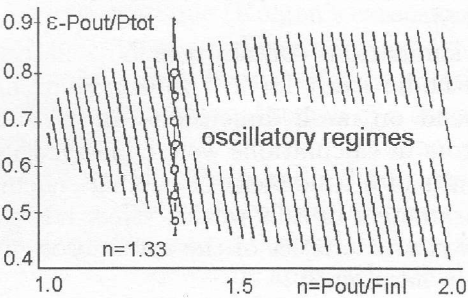


Fig. 2. Stability range for the model diffuser.

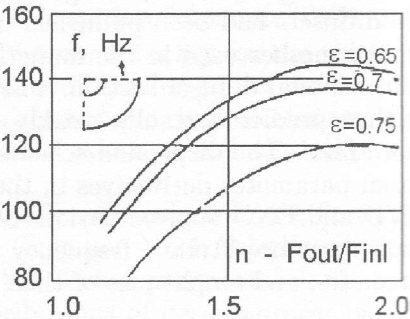


Fig. 3. Main harmonic frequency for diffuser fam

with simultaneous rise of rate amplitude in the outlet section, of shock inte and maximal dimensions of the recirculation zone. In the range $0.58 < \varepsilon <$ one can observe the appearance of the difference between the neighbouring pe dical intervals of oscillations. Harmonic analysis of the rate function for inlet an outlet sections in the limits of double period shows the rise of odd harmo For $\varepsilon = 0.6$ the first harmonic reaches its maximum and the subsequent decre of the pressure ratio causes the disappearance of cycle bifurcation and flow osc lations at all. The study shows that numerical solution of diffuser flow prob for transonic regimes depends on initial data. For example, for the diffuser w $n = 1.33$ the range of pressure ratio $0.5 < \varepsilon < 0.9$ generates flows with diffe oscillation frequencies depending on initial regime (Fig.1). The decrease of pre sure ratio from 1.0 to 0.5 generates the first solution branch with cycle bifurc within the limits of $0.58 < \varepsilon < 0.68$ and flow stabilization for $\varepsilon > 0.5$. Rise o pressure ratio starting from steady regime $\varepsilon < 0.5$ gives another solution branch with double frequencies, which passes into first branch with some hysteresis.

The range of the shock motion depends on the pressure ratio and has limits from $(0.2 - 0.7)L$ for $\varepsilon = 0.7$ and $n = 1.33$. If the pressure ratio decreases, the range of shock motion decreases too and for $\varepsilon < 0.55$ collapses into the point of shock fixing. In this case the separation region diminishes to fixed dimensions. Fig. 3 represents also the behavior of the main harmonic frequency of oscillations as a function of the area ratio n. For interval $1.5 < n < 2.0$ one can observe the maximum of frequency for the all studied regimes.

Diffuser area ratio is the main factor to determine the amplitude and frequency characteristics of oscillatory flow regimes. Fig. 2 represents the region of oscillatory regimes for the model diffuser. Boundaries of the region divide the plane into the part with oscillatory regimes and the domain of shock stability.

Diffusers of real exhaust systems have similar regions of oscillatory instability accompanied by the shock motion and flow rate oscillations. For example, the diffuser (Ŝt'astný,1989), under uniform inlet conditions and for Mach numbers $M > 0.7$ is characterized by flow oscillations.

Analysis of the flow through the isolated diffusers with $n = 1.325$ (Maier, 1989) under uniform inlet and outlet conditions shows the existence of oscillatory regimes for $0.6 < \varepsilon < 0.8$, up- and downstream shock displacement with frequencies 40-70Hz. All the phenomena, discovered for the model diffuser, occur in the case of the Maier's diffusers too.

Calculations show, that injection of high energetic supersonic jet shifts the range of shock movement upstream the channel and the flow rate oscillations do not spread till the inlet section (Fig. 4). The inlet tip clearance swirl within the limits $0 < \alpha < 30^o$ stabilizes the oscillatory regime and increases its frequency. Oscillatory process in the outlet channel part is conserved, accompanied by the shock motion and periodical rise of flow separation zone, reaches frequency near 130Hz, when the swirl increases up to 30^o. The increase of swirl angle from 30^o to 40^o damps the amplitude of oscillations approximately by 10 times and under $\alpha = 40^o$ the oscillations fade out. So, the results of qualitative modeling correspond to the experimental data (Maier,1989) (and confirm the structure of the flow) and nonstationary characteristics. Evolution of the shock and recirculation zone at the period corresponds to the experimental one. The flow calculation with inclusion of the tip leakage gives frequencies reduced by 1.3 times in comparison to the experiment. More detailed data are presented in (Solodov et al., 1994).

5. Spatial effect in oscillatory instability in diffusers

In order to get a deeper understanding of oscillatory phenomena in real exhaust hoods the flow simulations through the axial-radial diffuser under variable static pressure $P_{out}(\varphi)$ with constant circular gradient were conducted. The circumferential gradient of $P_{out}(\varphi)$ under condition of $P_{tot} = const$ leads to

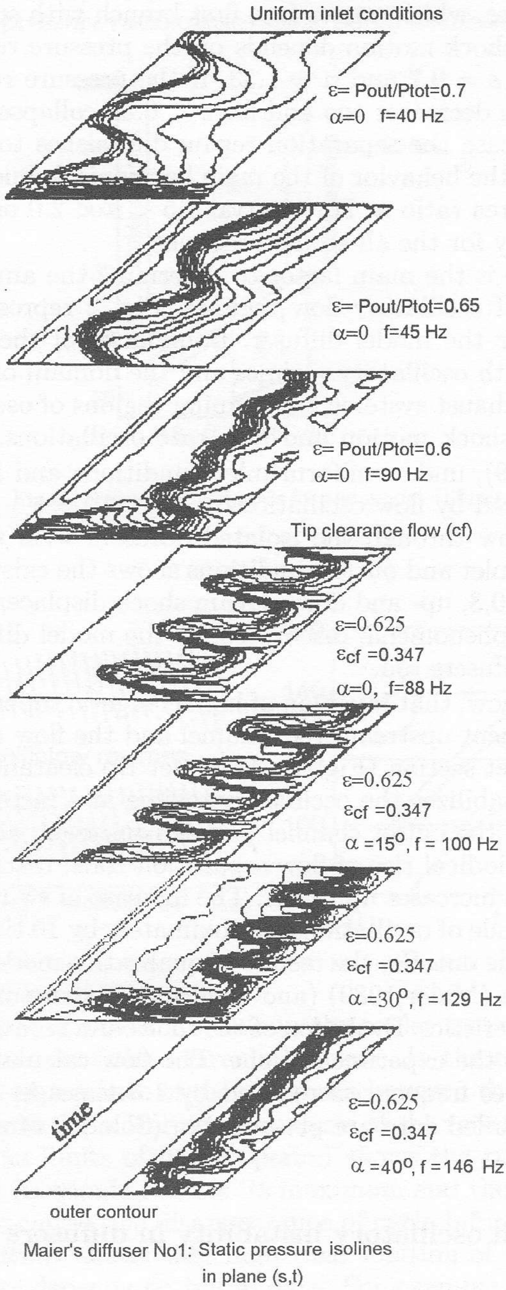


Fig. 4. Range of shock wave movement depending on regime.

development of circumferential velocity component in the flow. Streamlines are not present in the meridional planes, their form have variable curvature so the oscillation frequencies of flow parameters depend on φ. It leads to deformation of the shock wave at the outer diffuser shell.

In Fig. 5, 6 there are represented some results of the model diffuser flow calculations for $0.5 < \varepsilon(\varphi) < 0.68$ (regime A) and $0.7 < \varepsilon(\varphi) < 0.88$ (regime B). The main peculiarities of axisymmetrical non-stationarity are seen also in the non-axisymmetrical case.

For regime A, as the most loaded one, the oscillations of flow parameters are registered with periodically varying amplitude. Such nonstationary phenomena also are known as beats. The period of beats in this regime is equal to 9.75Hz. Analysis of inlet and outlet flow rate oscillations for sections $\varphi = 0$ and $\varphi = \pi$ shows the presence of two near modes with the same difference which causes beats. The spectral analysis of integral flow rate oscillations points out the existence of two modes simultaneously with main harmonic frequencies near 146.5Hz and 156.25Hz. For the family of Maier's diffusers (No. 1, 2) the beatings were recorded in experiment too, at the frequency of 10Hz.

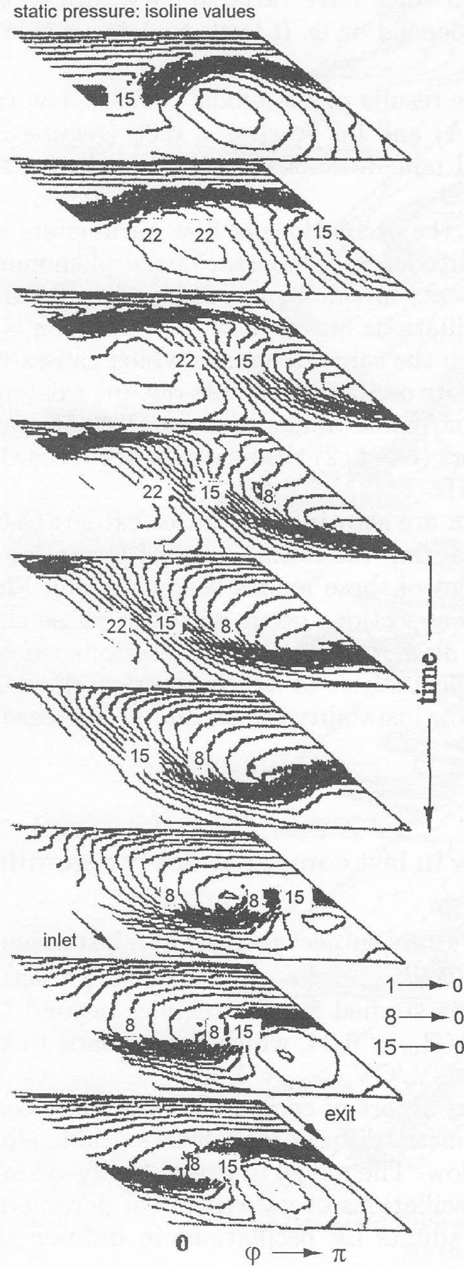
For regime B the described phenomena are showing to a lesser extent. Oscillations with different frequencies appear during the initial stage of formation of oscillatory process and then the frequencies of these modes become equal. Flow structure oscillations in different meridional sections occur with the phase shift which increases with the rise of φ and for diametrically opposite sections reaches one half of the period. So, the comparative analysis of axisymmetrical flow and 3D calculation suggest the expansion of flow instability region and an increase of the oscillation frequencies range.

6. Phenomena of the oscillatory flow in last compartment "stage-diffuser"

The exhaust compartment chosen as a study subject consists of a last stage of LP cylinder of SKODA, Plzen turbine 500MW and the exhaust hood (Ŝt'astný, 1989). Flow through the compartment was studied for the regimes beyond the operation point in range $0.16 < \varepsilon = P_{cond}/P_{tot} < 0.24$, which corresponds to the range of volumetric flow rate $1.0 < \mathrm{GV} = \mathrm{GV}/\mathrm{GV}_{design} < 1.33$.

Studied phenomenon for the diffuser as a part of compartment is more complicated due to effect of casing axial asymmetry diffuser inlet swirl variable along the radius and the effect of tip clearance flow. The model of stage axially symmetric diffuser compartment contains HF oscillations of gasdynamical parameters with nz_1 main harmonic frequency and admits LF oscillations in diffuser and interstage axial spacing.

The LF self-exited flow oscillations for SKODA diffuser without the tip clearance flow do not arise. On the outer diffuser shell the shock and separation bubble arises in the range $(0.3 \div 0.6)L$, the flow has a diagonal structure due to significant circular velocity component behind the rotor. Near the inner diffuser



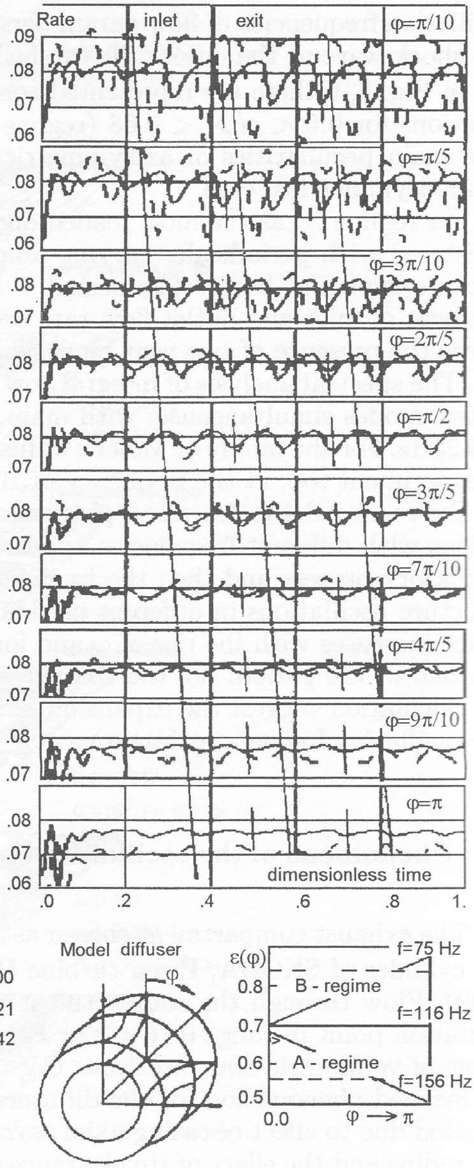


Fig. 5. Shock wave evolution on outer diffuser shell (A-regime).

Fig. 6. Phase shift of rate oscillations for differe diffuser sections (B-regime).

shell the recirculation zone arises which is placed in the diffuser and does not propagate upstream.

The tip clearance flow model is based on the experimental data (Maier, 1989) of total static pressure and velocity angles depending on the radius as functions of volumetric flow rate. Radial dimension of the tip jet reaches 4% of the blade length. Owing to expansion the total pressure in the tip clearance flow drops along first third of the length of the diffuser dimension by 1.5 times, swirl angle is reduced from 70^o to $20^o \div 30^o$. Tip clearance flow modeling in the range $1.0 < GV < 1.33$ provokes self-exited flow oscillations in the diffuser, axial spacing and bevel cut domain of rotor channel. Some data presenting the calculations are given in Fig. 7-10.

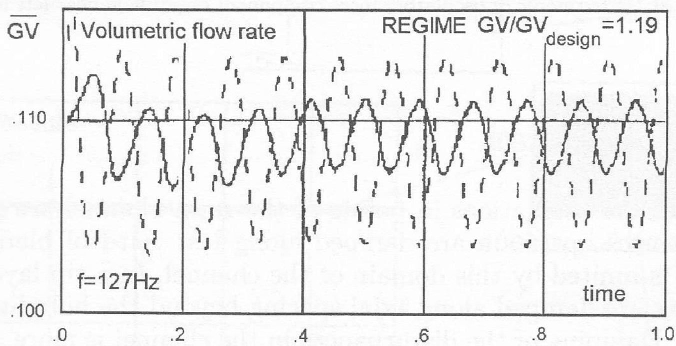


Fig. 7. Oscillation of diffuser volumetric flow rate for inlet and exit sections.

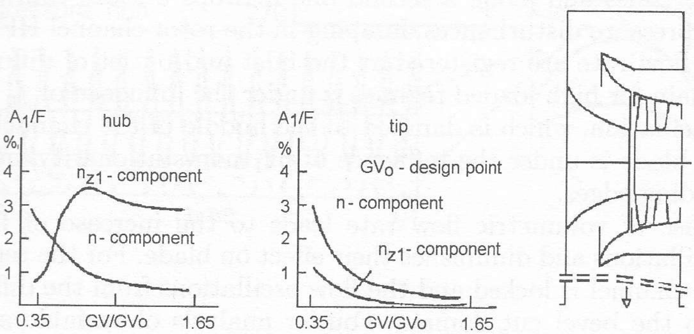


Fig. 8. Amplitude of 1^{st} harmonic of HF and LF force component on blading of SKODA LP last compartment.

Near the regimes $GV = 1.18 \div 1.19$ the range of shock movement is $(0.25 \div 0.35)l$ with frequencies $127\text{Hz} < f_p < 137\text{Hz}$ and flow rate amplitudes $0 < \Delta G <$

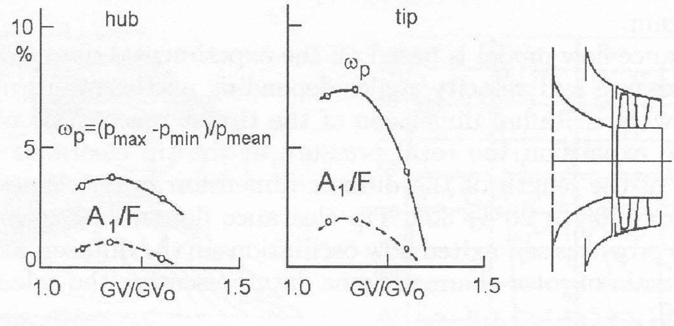


Fig. 9. Amplitude of 1st harmonic of oscillatory force component on a blade near left horizontal joint.

10%. Static pressure oscillations in points of the rotor channel are given in Fig. 10. Its disturbances upstream are damped along last third of blade chord and their influence is limited by this domain of the channel. In a tip layer of blading the disturbances are damped along axial spacing beyond the hub due to high velocities of flow. Damping of the disturbances in the channel is more strong in the case of the hub because of active blade profile in this layer. High frequency oscillations of flow parameters for rotor inlet are most strong for hub layer because of small relative stator-rotor clearance. In a section of leading edges the amplitude of 1^{st} harmonic of nz_1 oscillation is 5.3%, for first third of the blade chord it reduces by 5.5 times and along a second one it drops 5 more times. Despite of such a strong pressure disturbances damping in the rotor channel HF oscillations of volumetric flow rate are registered at the inlet and outlet of diffuser. So, the bevel cut domain for high loaded regimes is under the influence of f_p – frequency parameters oscillation, which is damped to the middle of the channel and simultaneously the blade is under the influence of nz_1 non-stationarity, mainly in the vicinity of leading edges.

The increase of volumetric flow rate leads to the increase of frequency of self-exited oscillations and diminishes their effect on blade. For the regimes GV > 1.33 the rotor channel is locked and the flow oscillations from the diffuser do not penetrate into the bevel cut domain. Fourier analysis of variable aerodynamic blade force for different layers points out the growth of the amplitude of HF components and the existence of maximum amplitude of f_p components in vicinity of GV = 1.18 when the GV increases too. It may be explained by two factors: the presence of own extremum for the process of shock oscillation in a diffuser (see, for example Fig. 4) and gradual channel, bevel cut and diffuser inlet locking when GV increases.

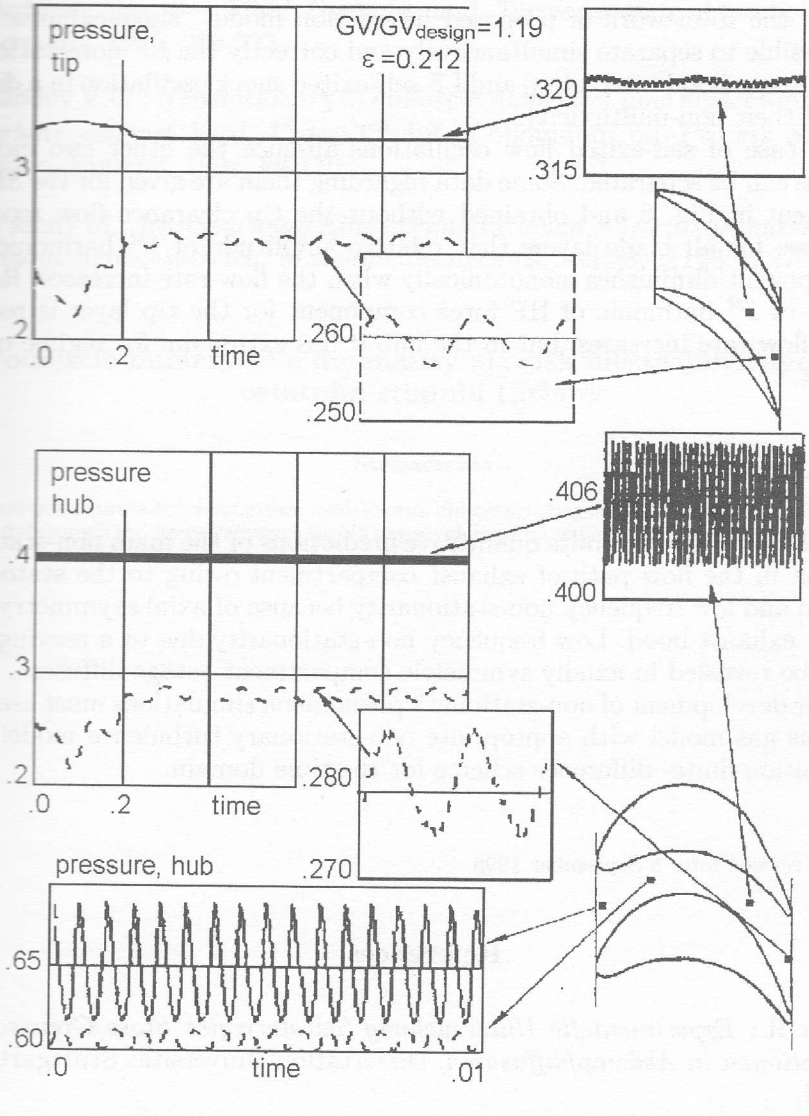


Fig. 10. Pressure oscillations in points of middle surface in rotor channel (nz1 and fp components).

7. Flow non-stationarity in the "stage-exhaust hood" compartment

Within the framework of proposed interaction model "stage-exhaust hood" it is impossible to separate simultaneously and correctly the LF non-stationarity due to the stage-hood interaction and LF self-exited shock oscillation in a difffuser because of their non-multiplicity.

In the case of self-exited flow oscillations absence the other two modes of oscillations can be separated. Some data regarding them are given for the SKODA compartment in Fig. 8 and obtained without the tip clearance flow modeling. One can see for all blade layers that relative amplitude of 1^{st} harmonic for [illegible] force component diminishes monotonically when the flow rate increases. Relative amplitude of 1^{st} harmonic of HF force component for the tip layer is reducing when the flow rate increases, but in the hub it has extremum for regime of axial flow outlet.

8. Conclusion

Inviscid flow model permits qualitative predictions of the main non-stationary phenomena in the flow path of exhaust compartment owing to the stator-rotor interaction and low frequency non-stationarity because of axial asymmetry of the diffuser or exhaust hood. Low frequency non-stationarity due to a moving shock wave can be revealed in axially symmetric compartment «stage-diffuser».

Further development of non-stationary phenomena simulations must use unsteady viscous gas model with appropriate non-stationary turbulence model and a high resolution finite- difference scheme for the time domain.

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Podejście numeryczne do analizy zjawisk niestacjonarnych w ostatnim stopniu turbiny

Streszczenie

Artykuł przedstawia trójwymiarowe modele oraz obliczenia dotyczące oscylacyjnych zakresów przepływu w dyfuzorach wylotowych dużych turbin parowych współpracujących z ostatnim stopniem turbiny.