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AEROELASTIC SELF-OSCILLATIONS OF PLANE CHANNEL WALL

J.M. Temis[1]
A.V. Selivanov
Central Institute of Aviation Motors
Moscow, Russia

ABSTRACT

Plane channel with wall supported by springs and damper is considered to describe seal aeroelastic oscillations. One of the channel walls has two degrees of freedom and other wall is stationary. The investigation method is based on simulation of the non-stationary gas flow in a channel to determine the aerodynamic forces, followed by the analysis of the aeroelastic stability.

Transient gas flow models are developed to obtain aerodynamic loads acting on the channel wall for two seal type (with smooth and finned channel). Corresponding rigidity and damping gas layer parameters obtained from these loads are included into the dynamic model of the seal for self-oscillations analysis.

The effect of structural parameters on the implementation of convergent oscillation and self-oscillation modes is shown; a picture of the aeroelastic stability boundary is given. A paradox of destabilization of the system with the increasing damping is observed for a certain parameter set.

INTRODUCTION

Turbine engine performance, specific fuel consumption and service life are strongly defined by non-stationary processes, which may take place in seal ducts and channels between rotor and stator. Modern aircraft gas turbine engine have about 50-100 seals. Some of them have a smooth flowing channel (annular seals), other have a finned flowing channel (labyrinth seals).

Gas flow influence on seal/channel elements and rotor dynamic behavior is of particular interest. Possible aeroelastic vibrations, especially self-oscillations of seal walls may cause fatigue failure of seals. As a result, lifetime of structure will be reduced, and operating costs will increase.

The main subject of this paper is the simulating of seal wall aeroelastic oscillations induced by transient gas dynamic loads. Fig. 1 shows overall algorithm of the aeroelastic analysis.

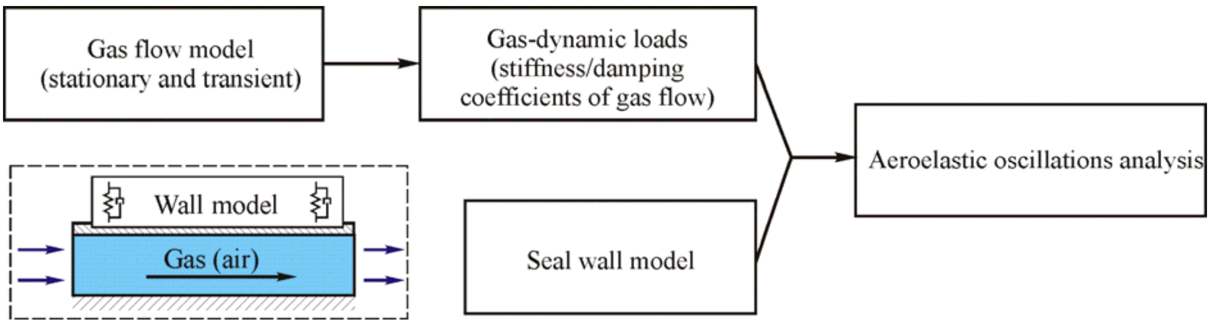

Fig. 1 The aeroelastic analysis algorithm

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Plane seal models with two degrees of freedom are considered to describe aeroelastic vibrations. Channel height δ is considered to be small in comparison with its length L. Spring dampers

[1]Corresponding author. Email tejoum@ciam.ru

with stiffness (k_0, k_1) and damping (c_1) coefficients imitate seal structure characteristics. External pressure p_e is constant. Moving wall is considered to be absolutely rigid.

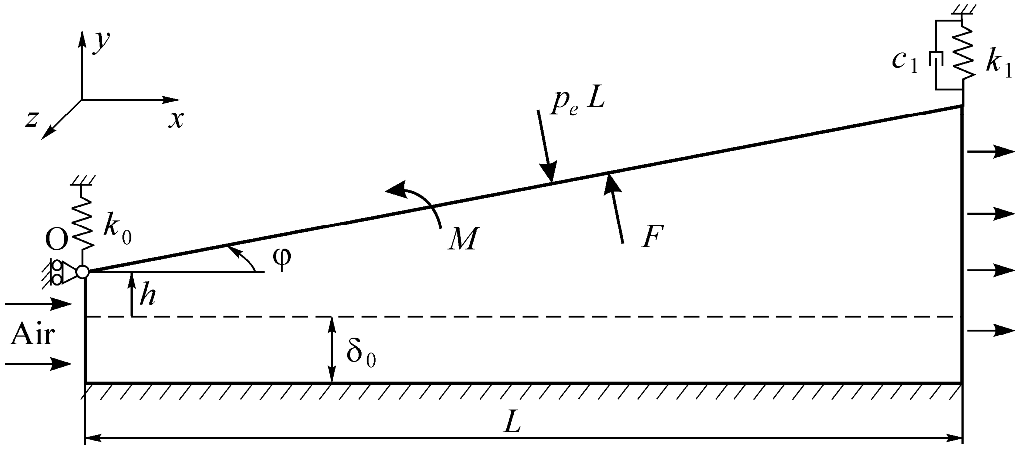

Fig. 2 Plane channel model with smooth walls

If φ is sufficiently small, then wall oscillations are described by equations

$$\begin{cases} m\ddot{h} + \dfrac{mL}{2}\ddot{\varphi} + c_1(\dot{h} + \dot{\varphi}L) + k_0 h + k_1(h + L\varphi) = \Delta F \\ \dfrac{mL}{2}\ddot{h} + \dfrac{mL^2}{3}\ddot{\varphi} + c_1 L(\dot{h} + \dot{\varphi}L) + k_1 L(h + \varphi L) = \Delta M \end{cases} \tag{1}$$

Here m is the wall mass; h and φ are deviations from the static equilibrium position; ΔF and ΔM are the aeroelastic force and moment deviations from their values at the static equilibrium. These deviations can be represented as

$$\Delta F = \frac{\partial F}{\partial h}h + \frac{\partial F}{\partial \varphi}\varphi + \frac{\partial F}{\partial \dot{h}}\dot{h} + \frac{\partial F}{\partial \dot{\varphi}}\dot{\varphi},\ \Delta M = \frac{\partial M}{\partial h}h + \frac{\partial M}{\partial \varphi}\varphi + \frac{\partial M}{\partial \dot{h}}\dot{h} + \frac{\partial M}{\partial \dot{\varphi}}\dot{\varphi} \tag{2}$$

Derivatives in equations (2) are called stiffness and damping gas seal coefficients. We have

$$F(t) = \int_0^L p(x,t)dx, \quad M(t) = \int_0^L xp(x,t)dx \tag{3}$$

Transient gas flow model is developed to obtain aerodynamic loads (force F and moment M) acting on the seal. Turbulent gas flow is generally described by a system of partial derivatives differential equations. This system consists of continuity equation, momentum equations, and energy equation. It also contains some equations, used to describe the turbulence model.

At the same time, there is a lot of experimental data that allows us to define friction coefficients, depending on Reynolds number. Thus, the problem can be simplified to one-dimensional model that reduces calculation time. For 1D gas flow model, we can write continuity equation (4), momentum equation (5), energy equation (6), and state equation $p = \rho RT$ [3].

$$\frac{\partial(\rho S)}{\partial t} + \frac{\partial(\rho u S)}{\partial x} = 0 \tag{4}$$

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + \frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{2\tau}{\rho S} = 0 \tag{5}$$

$$\frac{\partial T^*}{\partial t} + u\frac{\partial T^*}{\partial x} - \frac{1}{\rho c_p}\frac{\partial p}{\partial t} = 0 \tag{6}$$

Here ρ is a gas density, u is a flow velocity, p is a pressure, and T^* is a stagnation temperature. Shear stress τ is equal to $\tau = f\rho u|u|/2$, where wall friction factor f for a turbulent flow is equal to $f = 0.187 \cdot \mathrm{Re}_x^{-0.333}$.

Deviations for φ and h are small, so the system can be linearized. One-dimensional modeling of transient gas flow in linear approximation is carried out using finite difference method with implicit scheme. In order to test the results, two-dimensional transient gas flow is analysed. The analysis is carried out using STAR-CD software. The difference between 1D and 2D model results (for aerodynamic force and moment) is less than 1,6%. Therefore it is valid to use one-dimensional linear approximation.

Stiffness and damping gas layer coefficients are included into the dynamic model of the seal for self-excitation vibrations analysis and boundary of aeroelastic stability evaluation.

Let us find the solution of system (1) in the following form

$$\begin{cases} h = He^{i\omega t} \\ \varphi = \Phi e^{i\omega t} \end{cases} \tag{7}$$

where ω is a self-oscillations frequency, H and Φ are complex amplitudes. Combining (1), (2), (7) and writing non-trivial solution existence condition, we can determine parameters of self-excitation oscillations.

Fig. 3 represents safe operating area *1*, unstable area *2*, and stability threshold *3* (harmonic self-oscillations curve). It must be noted, that for some seal parameters structural linear damping c_1 increase may cause oscillations increase and seal instability. This effect is similar to Mansour`s anomaly and can be explained as follows: there is no direct coupling between damping coefficient increase and damping work increase for such systems [1].

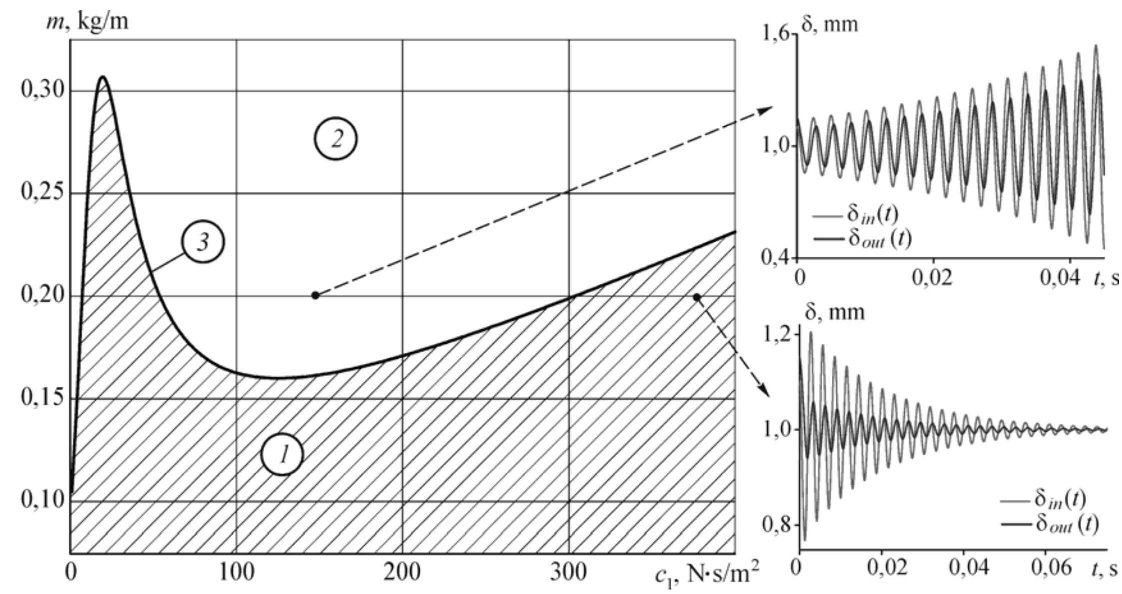

Fig. 3 Smooth seal stability region: 1 – safe operating area; 2 – unstable area; 3 – stability threshold

Let us consider the behavior of characteristic equation roots λ with damping coefficient c_1 vary. As a result of characteristic equation numerical solution, two pairs of complex conjugate roots (λ_1, λ_2) and (λ_3, λ_4) are obtained.

The curves on the complex plane, represented in Fig. 4, show λ_1 and λ_3 versus c_1 (increasing c_1 is indicated by arrows). For Re(λ)<0 oscillations decrease exists. The intersection points of curves λ with ordinate axis Re(λ)=0 correspond to periodic self-oscillations mode and the motion is unstable in area Re(λ)>0.

Fig. 5 shows complex amplitude ratio H/Φ for $\lambda_{1..4}$ (self-oscillations points are marked by circles or triangles).

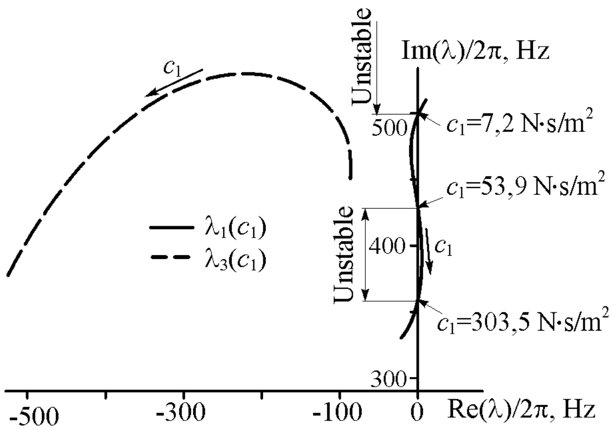

Fig. 4 Roots λ versus c_1

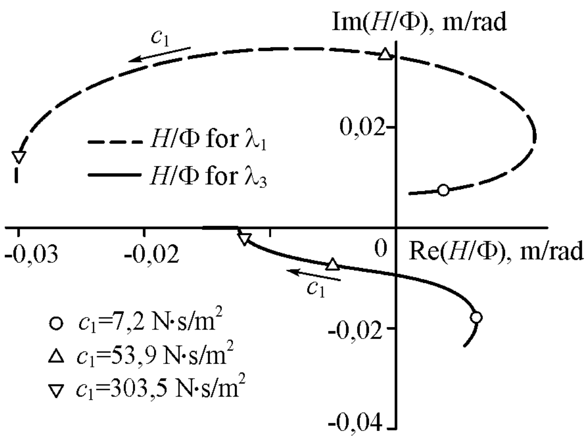

Fig. 5 Amplitude ratio H/Φ versus c_1

Boundary of stability may qualitatively change, with variation of stiffness k_0, k_1. If k_0, k_1 are "small" values, then damping increasing turns the system from oscillations increase to oscillations decrease. Nonlinear effects appear with stiffness increasing (fig. 6).

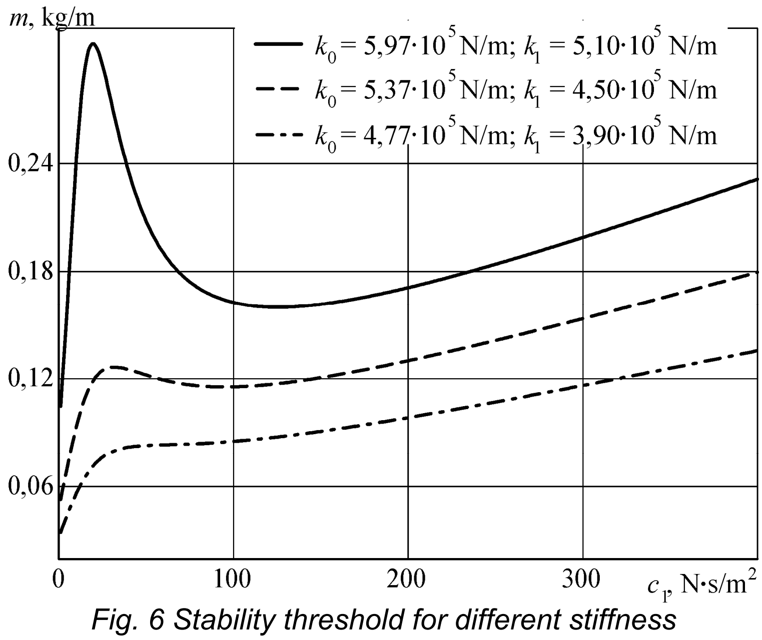

Fig. 6 Stability threshold for different stiffness

So self-oscillations may be appearing in similar gas seal structures. For more complete stability analysis and flow condition influence on the seal dynamic behavior see [2, 3].

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This method may be applied for the determination of aerodynamic stability for different seal types, for example for labyrinth seal with finned wall (see the Fig. 7).

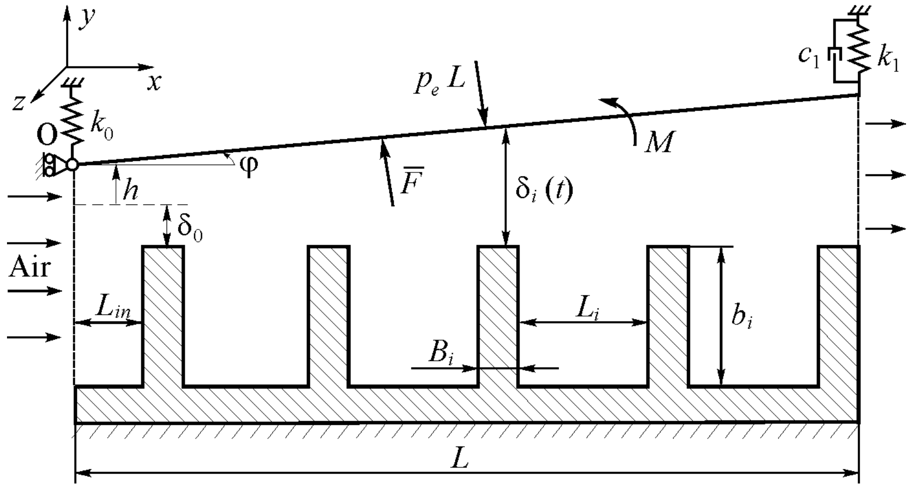

Fig. 7 Plane channel model with finned wall (labyrinth seal)

Gas flow at the labyrinth seal may be described by "one-volume" model [4]

$$\frac{\partial}{\partial t}\left(p_i S_i\right)+\mu_0\mu_{i+1}\delta_{i+1}\sqrt{RT(p_i^2-p_{i+1}^2)}-\mu_0\mu_i\delta_i\sqrt{RT(p_{i-1}^2-p_i^2)}=0, \qquad i=1..N, \tag{8}$$

where N is number of chambers between the fins, S_i is cross-section chamber area, μ_i is discharge coefficient

$$\mu_i=\frac{\pi}{\pi+2-5\beta_i+2\beta_i^2}\text{, where } \beta_i=-1+\left(\frac{p_{i-1}}{p_i}\right)^{(\gamma-1)/\gamma}.$$

For checking the "one-volume" model results, two-dimensional transient gas flow is analysed. As before the difference between 1D and 2D model results is less than 2%.

Fig. 8 shows stability region for finned channel. The effect of seal destabilization with increase damping coefficient isn't displayed for selected seal parameters.

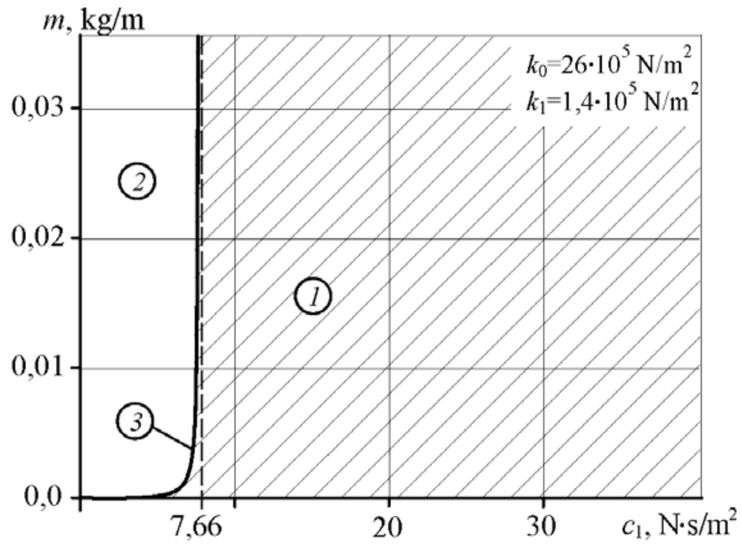

Fig. 8 Finned seal stability region: 1 – safe operating area; 2 – unstable area; 3 – stability threshold

The equations (1) and (8) can be joining to one nonlinear system, where $\Delta F=\int_0^L p(x,t)dx-F(h_0,\varphi_0)$ and $\Delta M=\int_0^L p(x,t)x dx-M(h_0,\varphi_0)$.

Numerical solution of this system for different damping coefficient values is confirm stability threshold (see the Fig. 8). As example, Fig. 9 shows $h(t)$ and $\varphi(t)$ behavior for small damping (point from unstable area, Fig. 8).

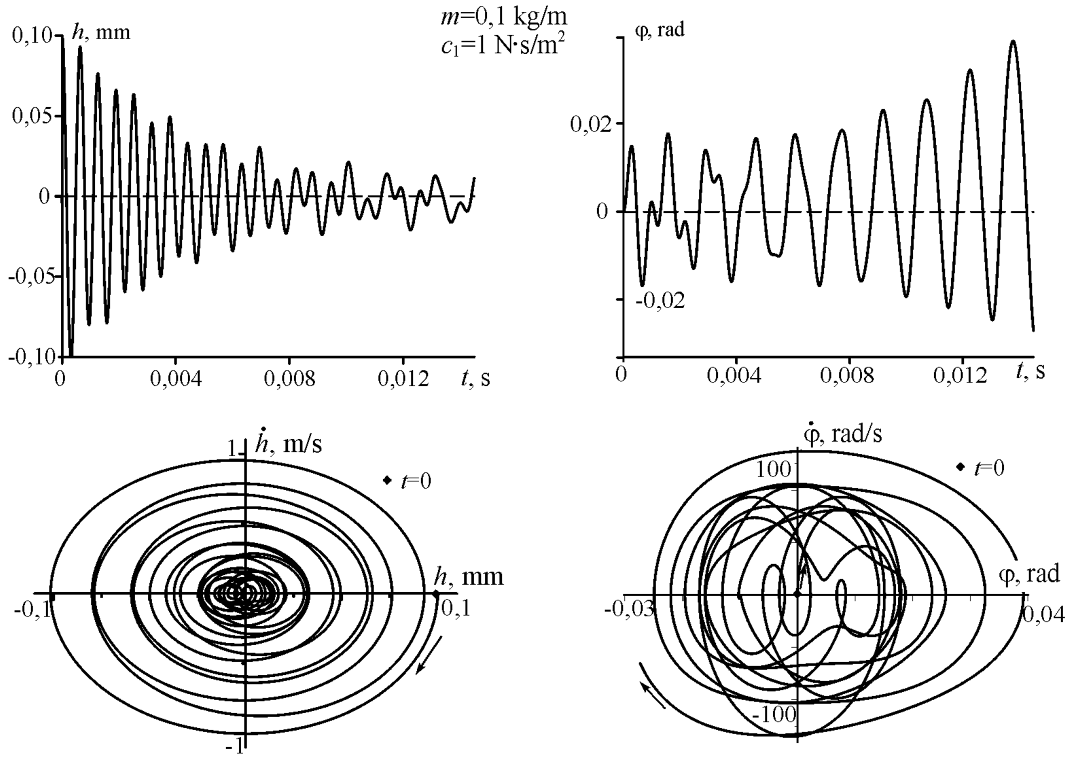

Fig. 9 Wall oscillations for different damping

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