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INVESTIGATION OF DYNAMIC STRESSES OF TAIL BEAM OF AN MI-8 HELICOPTER

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Abstract. Results of theoretical and experimental research of stresses in the skin of a tail beam of an Mi-8 helicopter at resonant dynamic loading are discussed in this paper. It is shown that resonant tests of the object with an appropriate choice of configuration of internal and external constraints are one of the most effective and economical ways to achieve the necessary result. It is also obvious that the choice of a configuration for the complex dynamic system should be solved on the basis of a careful preliminary analysis of expected dynamic properties. First of all, the research that was performed shows the very significant effect of the elasticity of internal and external constraints on the dynamic properties of real aircraft components at full-scale dynamic testing. On other hand, it is shown that the correct modelling of a complex elastic system and its constraints allows one to adequately describe the dynamic properties of the real structure and to obtain good coincidence with experimental data.

Keywords: aircraft, full-scale testing, dynamics, stress.

1. Introduction

In recent years, research and development to create structural health monitoring (SHM) integrated into the structure of aircraft has progressed rapidly (Health… 2004, Ladda *et al.*1993, Kudva *et al.1993,* Lemistre *et al.* 2000). One of the major stages in this research is the practical evaluation of the efficiency and working capacity of similar kind of systems in full-scale tests or under operating conditions. In the frame of the European project AISHA, the authors of this paper have been involved in the preparation and realization of full-scale tests of some components of a structure with some elements of the SHM integrated into the aircraft structure. It was therefore necessary to solve a number of problems concerning the rational selection of programs of loading and acceptable means for to realize it. More precisely, the problem of defining the optimum single-channel loading for a tail beam of an Mi-8 helicopter to localize the zone of the greatest stresses was solved. It was necessary to perform the analysis of dynamic characteristics and their experimental definition. In the course of this research, some problems concerning the adequate model of a dynamic system that represents both scientific and practical interest came to light and were solved. By V. Pavelko and S. Kuznetsov, problems involved in the modelling of real dynamic systems, are exposed (Pavelko 2006, Kuznetsov *et al*. 2006). It is shown that the results of the modelling well correspond to the experiment, if the structure is described realistically. In this paper, the results of theoretical and experimental research of stresses in the skin of a tail beam of an Mi-8 helicopter are stated at resonant dynamic loading.

2. Object of dynamic tests and statement of problem

The thin-walled beam 1 has the form of a truncated cone with a length of 5500 mm, i.e. the length of the beam (Fig 1). The base diameters are respectively 1000 and 550 mm. The material of all main elements of the beam is the aluminium alloy D16-T (2024-T4), with a primary skin thickness of 0.8–1 mm. A larger thickness of about 2 mm is reached in the trailer part of the beam. The structural skeleton of the beam consists of 17 frames and 26 stringers. Forced frames on the ends of the beam have increased strength and a lot of holes for bolts for joining with the others components of structure. Stringers with angular cross-section have an area of 1.2 cm^2 in the compressed zone of the section and 0.63 cm^2 in the stretched zone. In a trailer part of the beam, the area of the cross-section is reduced to 0.63 cm^2 and 0.48 cm^2 respectively. The skin is point-welded to the stringers along most of the length with a glutinous layer. The sheets of skin have riveted joints with power frames. The skin sheets of aluminium are connected to one another with rivets and are also riveted to the frame. The surfaces of the skin, stringers, and frames are anodised. The outer surface of the beam is covered by a protective paint covering about 0.2 mm thick.

The problem was formulated as follows. There is a thin-walled beam with one fixed end. A mechanical vibrator 2 is attached on the opposite end and allows exciting the harmonious vibrations (Fig 1). Some zone of the structure with typical rivet joints of skin with elements of a basic frame is selected for monitoring. This zone of structure marked for non-destructive inspection (NDI) is the technological joint of stringers by the riveted joint between frames 7 and 10 (Fig 2). It is an ideal place for the evaluation of the working capacity and efficiency of the new technology of detecting fatigue damage during full-scale dynamic tests. For this purpose, the type of loading at which the maximum level of stress will be localized in the chosen zone must be defined. On the other hand, the loading system should be as simple as possible and consume little power. The preferable resolution of this problem is the choice of a resonant way of loading with one of the natural frequencies of the beam so that the maximal curvature of bent axes of a beam would be in the zone of localization of expected fatigue damage. At the same time, having the resonant frequency as low as possible is preferable to decreasing excitation power.

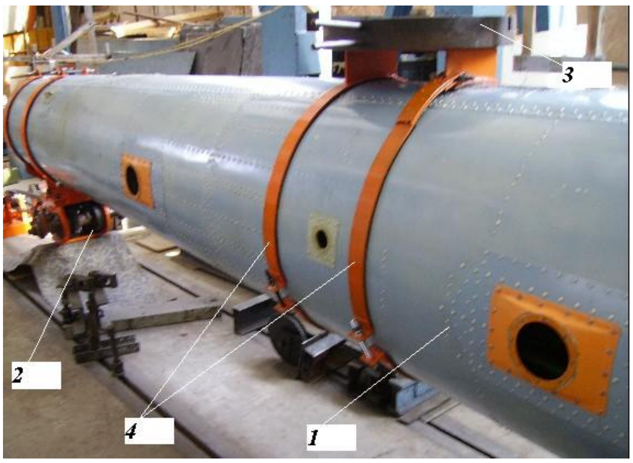

Fig 1. General view of stand

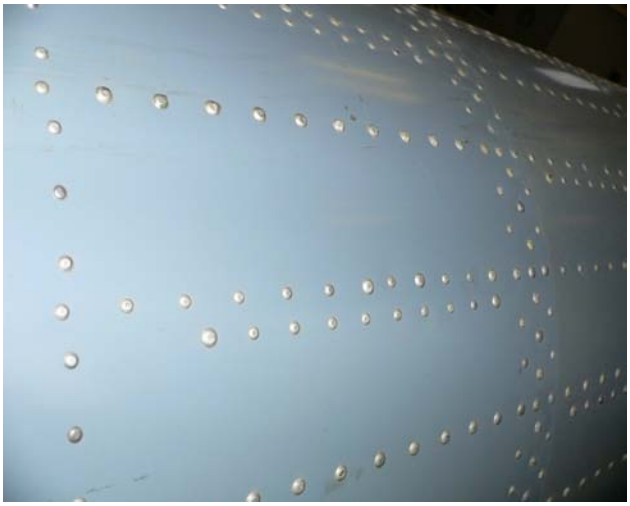

Fig 2. Picture of technological rivet-joint of stringers between frames 7 and 10 (outer view)

3. Estimation of nominal stress of beam under its form of bent axes

There are many means of analysis the strain and stress state in dynamic systems with continuous parameters (Kuznetsov *et al.* 2006, Timoshenko *et al.* 1974, Ananijev *et al.* 1972, Tse *et al.* 1983, Smith *et al.* 1989). The simplest of them is the beam system. Within the limits of the theory of the bending of beams, internal forces and direct stress in their cross-sections are unequivocally defined by the form of bent axes of a beam (Kuznetsov *et al.* 2006).

Let $v_i(z)$ be the form i of the bent axes of a beam. Then the bending moment $M(z)$ in this section is defined by curvature of this axis in the given section

$$M(z) = EJ_x \frac{d^2 v_i}{dz^2}, \tag{1}$$

and maximal direct stress is defined by the formula

$$\sigma_{\max} = \frac{M(z)}{W(z)}, \tag{2}$$

where $W(z)$ is the section modulus.

If the amplitude of movement of the free end of the beam is designated as A_0, then formula (2) of maximal direct stress can be written in the following form:

$$\sigma_{\max} = \frac{EJ_x(z)}{W(z)} \frac{d^2 v_i}{dz^2} = \frac{1}{2l^2} EDA_0 \frac{d^2 \overline{v}_i}{d\overline{z}^2}, \tag{3}$$

where $J_x(z)$ is the moment of inertia of the section, D and l are the external diameter of the section and length of the beam, and E is the module of elasticity of the material of the beam. The form of vibrations is expressed in the relative form $\overline{v}_i = v_i / A_0$ as a function of the relative coordinate of section $\overline{z} = z / l$.

It is obvious that the constant $\sigma_0 = \frac{1}{2l^2} EDA_0$ has the same dimension as the stress, and its value is defined by the amplitude of the movement of the free tip of the beam. As a result

$$\overline{\sigma}_{\max} = \frac{\sigma_{\max}}{\sigma_0} = \frac{d^2 \overline{v}_i}{d\overline{z}^2}. \tag{4}$$

Thus, the maximal distribution of stress along the longitudinal axis of the beam at a bend in a beam under form i is completely defined by the relative curvature of the bent axes. To define the absolute value of stress, it is necessary to set the amplitude of the movement of the free tip of the beam and to calculate a scale constant σ_0.

4. Essential experimental results

The main goal of dynamic testing has been to define the stress state of the tail beam.

Many strain gauges were installed on the outer surface of the beam (Fig 3). Strain gauges with a base 10 mm and nominal electrical resistance of 200 Ohm were used. Items 5 and 10 are located in the centre of the zone that was investigated. In general there is the possibility to measure the strains in a skin both near a support and along the middle part of a beam. The data acquisition system also included amplifiers, converters of electrical signals by firm L-cards, a PC, and corresponding software. As a result of the static calibration, the constants of all strain gauges were evaluated. Two configurations of a dynamic system were investigated. In the first stage, in the middle part of the beam (near the critical zone) an additional weight of about 95 kg (3 on Fig 1) was fixed. During tests the exclusively strong influence of external and internal constraints on natural frequencies and modes was revealed (Pavelko 2006, Kuznetsov *et al.* 2006). It was shown that the adequate description of the real properties of these constraints in the theoretical analysis allows one to obtain good conformity with experimental results.

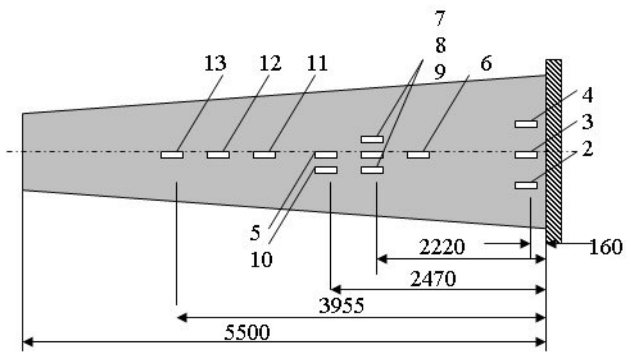

Fig 3. Scheme of location of strain gauges on outer surface of tail beam

However, another problem also came to light during the tests: the concentration of the big weight caused higher more local stresses in the structure and the undesirable fatigue failure of elements of the skeleton (Fig 4). In particular, fatigues damages appeared to the wall of the frame to which loading from one of the belts fastening additional weight 3 were transferred. In further tests the weight in the middle part of the beam was therefore reduced and was accepted to equal 15 kg. In the final configuration, an additional weight equal to 105 kg on the end of a beam was accepted. Experimental first and second natural frequencies of the beam were obtained and were equal to 5.5 and 52 Hz respectively. Because the second resonance of the beam was selected for the main working regimen of dynamic testing, three value of amplitude of external force were used at loading. The distribution of dynamic stress amplitude along the beam was measured and is shown in figure 5. It can be seen that stress increased at forced vibration with larger amplitude. But linear proportionality is conserved only if the amplitude of force is not more than 2.71 kN. This means there is some non-linear effect to a beam vibration of high intensity. It can also be seen that the non-linearity effect is not significant for stress distribution along the beam.

Fig 4. Fatigue cracks in frame as a result of dynamic overload induced by additional concentrated weight in middle part of tail beam

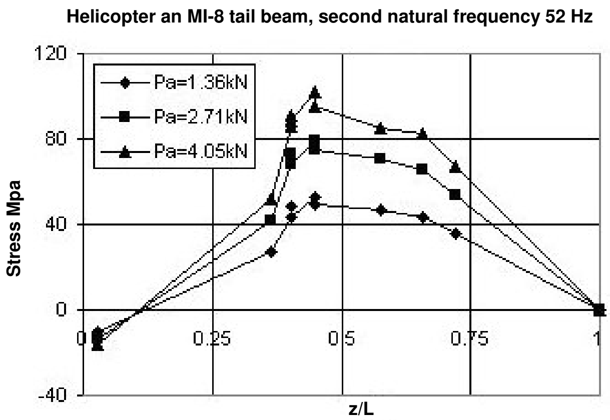

Fig 5. Distribution of dynamic stresses in skin of tail beam at forced vibration with second natural frequency

5. Numerical analysis of stresses at forced vibration of tail beam of an Mi-8 helicopter during dynamic tests

The first two natural frequencies and forms of a tail beam of an Mi-8 helicopter are obtained at the boundary conditions which are approximately the same as in dynamic tests (Health... 2004). It allows to obtain an estimation of the maximal direct stress in the sections of the beam. With this purpose, the second form presented in the relative view was approximated by a polynomial of the fifth degree. There is high enough accuracy of approximation of the form. It is known, however, that differentiation of a polynomial can lead to greater mistakes for derivatives. Approximation by a polynomial of a higher degree was therefore also carried out. A comparison of results really confirms this statement if the form along the length of the beam is considered. Significant differences are characteristic for zones of small stress near the tips of the beam. In the most intense zone, the difference of stresses at the fifth and sixth degrees of a polynomial is insignificant. The concentrated weights method and this technique of defining stress at resonance vibration were therefore used in the following numerical analysis of the dynamic properties of the beam. A parametrical analysis of the dynamic properties of the beam was performed by varying the weight on the free end of beam at the place the vibrator was installed. The first and second natural frequencies of the beam were found to be equal to 5.53 and 60.5 Hz as a result of numerical analysis. Corresponding natural forms of vibrations of the beam are shown in figure 6. It can be seen that the second natural form of vibrations has the maximal curvature in the middle part of the beam. The greatest stresses at second resonance are also there. Comparison of the experimental result of stress measurement at vibration with a frequency of 52 Hz (amplitude of a force of 2.71 kN) and predicted stress is illustrated on figure 7.

It can be seen that experimental points are mainly concentrated near the theoretical curve.

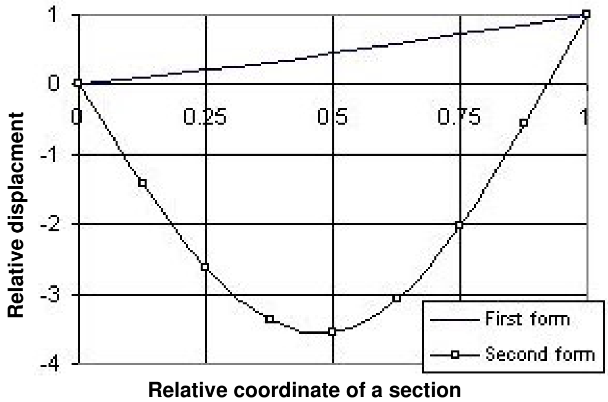

Fig 6. First and second natural forms of beam

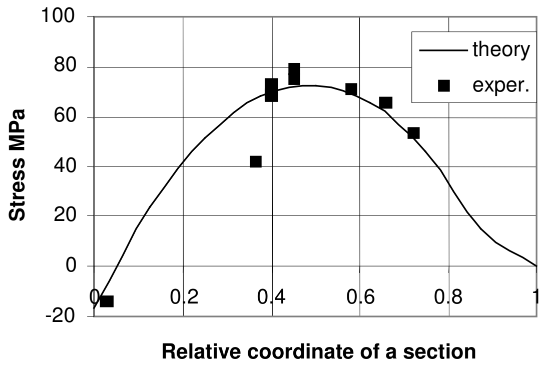

Fig 7. Comparison of predicted and measured stresses in skin of beam

5. Conclusions

This research allows allows to declare that appropriate use of dynamic properties of an aircraft component at planning of the full-scale dynamic tests it is possible to solve many problems concerning the optimization of a mode of loading. Thus, resonant tests of an object with an appropriate choice of a configuration of internal and external constraints are one of the most effective and economical ways to achieve the desired result. It is also obvious the problem of choosing a configuration for a complex dynamic system should be solved on the basis of the careful preliminary analysis of expected dynamic properties. And this problem is not simple.

First of all, this research shows the very significant effect of the elasticity of internal and external constraints on the dynamic properties of real aircraft components at full-scale dynamic testing. On other hand, it is shown that the correct modelling of a complex elastic system and its constraints allows one to adequately describe the dynamic properties of a real structure and to obtain good coincidence with experimental data.

Another important result of this research concerns the choice of a method of analysing the dynamic properties of elastic systems. The method of concentrated weights in a combination with classical theory of the bending of beams appeared to be a good tool for the research of the lowest natural frequencies and forms of vibrations of an MI-8 tail beam, and also for the distributions of strain and stress in of such a tail beam.

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SRAIGTASPARNIO MI-8 GALINĖS SIJOS DINAMINIŲ ĮTEMPIMŲ TYRIMAS

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Santrauka

Straipsnio tikslas – ištirti ribinių sąlygų įtaką nominaliems įtempimams atskirose plonasienės sijos vietose ir parinkti racionalų dinaminės apkrovos režimą planuojant realų eksperimentą. Nominalūs įtempimai įvertinti pagal sijos dinaminių charakteristikų analizės rezultatus koncentruotų masių metodu. Buvo modeliuojamos neklasikinės ribinės sąlygos, kurios tiksliau apibūdina realius sijos ryšius. Įvertinant sijos įtvirtinimo elastingumą ir vidinių ryšių elastingumą galima gauti tikslesnes laisvųjų svyravimų pirmųjų dviejų dažnių reikšmes, kurios gerai sutampa su bandymų rezultatais. Kaip rodo analizė, vidinių ir išorinių ryšių elastingumas turi didelę įtaką sijos dinaminėms charakteristikoms, tarp jų ir laisvųjų svyravimų formai ir dažniui.

Reikšminiai žodžiai: orlaivis, išsamūs tyrimai, dinamika, įtempimai.