ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

NUMERICAL MODEL OF AIRCRAFT GAS TURBINE ENGINE (BASED ON EXPERIMENTAL DATA OF DYNAMICAL COMPLIANCES OF ENGINE BODY)

Viatcheslav Baklanov, Stanislav Denisov

Tupolev Design Bureau
Tupolev embankment,17 105005 Moscow, Russia
moto@tupolev.ru

Abstract

Compliance of aircraft to new noise standards defines the tendency to switch to extra-high bypass ratio engines (from 4…6 to 8…11). The introduction of high by-pass ratio engines has caused broadening of the spectrum of power plant vibration impact and increase of structure-borne noise contribution to the acoustic field of the pressurized cabin. Facilities for reduction engine vibration intensity and vibration transfer along structure come first by selecting of vibration protection for pressurized cabin and integration vibration protection units into engine mounting attachments seems to us the most effective. But whatever vibration protection means are used to select parameters of vibration isolation units, calculated model is required which is based on real dynamic characteristics of engines and airframe in mounting points. The long-term investigations directed to dynamical characteristics definition for number of engines bodies (with different by-pass ratio) and airframe design of airliner allow to significantly specify calculation models of advanced aircraft design constructions in engine's rotor frequency range. And it allows to define tendency of engine's dynamic characteristics variation with by-pass ratio increasing. Analysis of experimental data makes it possible to divide the frequency range of investigation into several sub-ranges characterized by certain dynamic behavior of the engine and com be presented by simple and clear mathematical model. Taking this into account the analytical function was found, which describes change of engine compliances frequency characteristics. Within a wide range of rotor frequencies the dynamic behavior of engine body corresponds to the model of elastic-inertial system or to an elastic-dissipative element. It differs substantially from the idealized rigid-body model of aircraft gas turbine engine both by the value of dynamic compliance module and by the type of dynamic behavior.

INTRODUCTION

Vibration spectrum of high by-pass ratio turbofan engines shifts to low-frequency region (caused by possible use of reducing gear schemes gear scheme it is possible to introduce, low rotational speed of rotor fan etc.), what will define character of engine dynamic impact on airframe via mounting points.

This will require a new approach to design of vibroisolation engine mounts in view of realistic dynamic characteristics of the advanced structure (engines, airframes).

If calculation models of the airframe, the pylon and the cabin take into account several thousands of freedom degrees, the engine is usually considered as rigid-body, its mass and moments of inertia taken into account only.

This is the result of an old tradition of successful flutter calculations, as the rigid-body engine model is still true in that range (low frequency range, below 15 Hz).

Using long-term investigations the numerical model for real engine dynamic compliances was proposed.

EXPERIMENTAL DATA ANALYSIS

Performed experiments allow define more exactly calculation models of turbofan engines and airframes in rotor frequency range. And also to define tendency of dynamic date changing at by-pass ratio increase.

Such characteristics were determined for a number of by-pass turbofan engines distinguished substantially both in thrust and by-pass ratio (from 0,5...1,1 to 2,5...5,0), and for airframes of trunk-route aircraft.

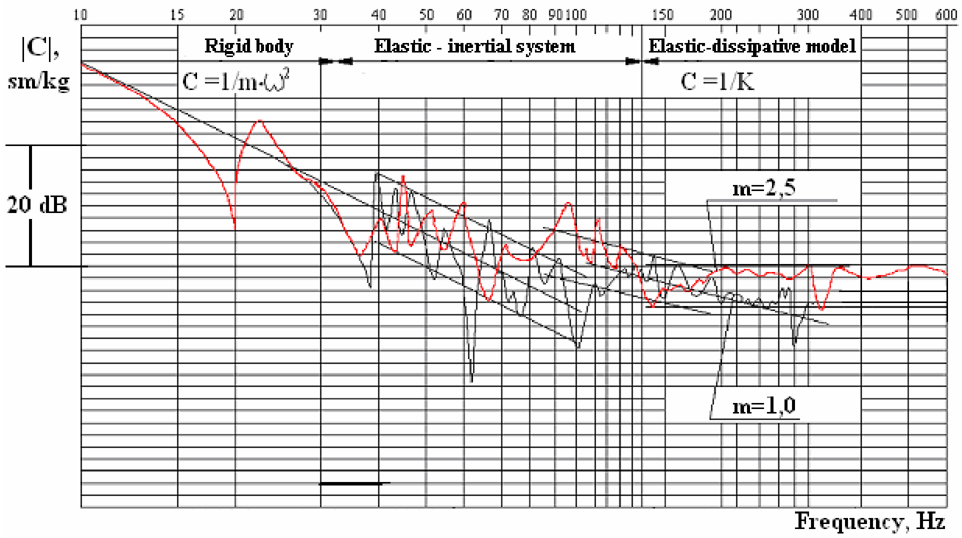

Figure 1- Modules of dynamical compliances of engines with different by-pass ratio.

Compliance values of such sub-systems as engine and airframe were determined by method of test effect within 10…500 Hz frequency range.

A well-known impedance testing technique was used: for definition of these characteristics of structures were excited by an electrodynamics shaker at constant harmonic input force amplitude and its frequency varying automatically within the studied range.

The investigated system linearity was verified by changing the applied force by several times; reciprocity principle observation was investigated by changing locations of effect application and response measuring.

Dynamic compliances of engine casings at attachment points has revealed that casing of engines with pass-by ratio (*m*) of 0.5…1.1 correspond to rigid body model up to 40 Hz, while for engines with pass-by ratio of 1.5…2.5 upper boundary of the rigid body model is shifted to 20 Hz.

It has been defined that at frequencies higher than 20…40 Hz up to 120 Hz the engine casing behaves as elastic-inertial system with large amount of resonances of various damping degrees.

Identification (Fig. 2) of said resonances (by comparison of different investigations results) made it possible to associate them with natural frequencies of a number of engine components (rotors, drive gear boxes, certain components fixed on casing etc.).

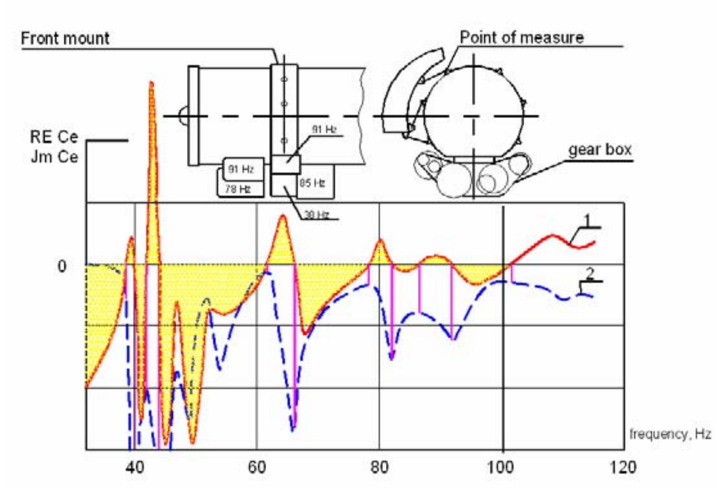

Figure 2 - Identification of engines resonance
1 –Real component of dynamic engines compliance;
2 – Imaginary component of dynamic engines compliance

Within frequency range of 120…500 Hz engine casing corresponds to the model of elastic – dissipative element.

Generalization of performed investigations has revealed that dynamic behavior of advanced gas turbine engines casings corresponds to rigid state model up to 20…40 Hz according to pass-by ratio (Fig. 3).

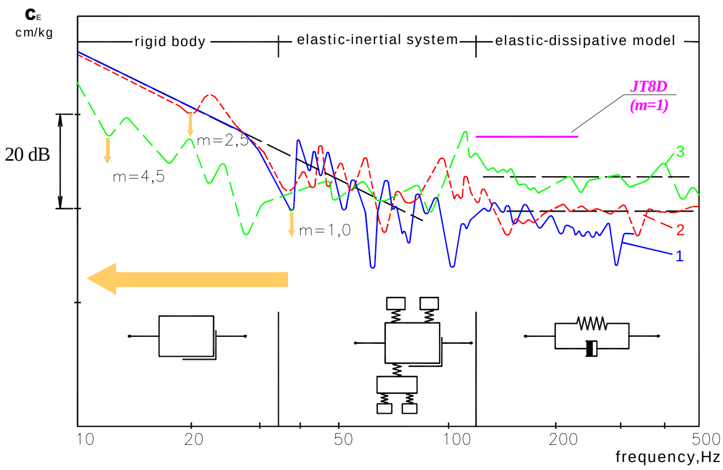

Figure 3 - Dynamic compliances of engine body at attachment points:
1 – m = 1; 2 – m = 2,5; 3 – m = 4,5.

If pass-by ratio is increased up to estimated 8…12 we should expect that upper boundary of dynamic behaviour of the engine, as a rigid body does not exceed 10 Hz.

Within wide range of rotor frequencies dynamic behavior of engine casing corresponds to model of elastic-inertial system or to elastic-dissipative element; differing substantially from idealized model of aircraft gas turbine engine as a rigid body both by value of dynamic compliance module and by type of dynamic behavior.

NUMERICAL MODELLING OF ENGINE DYNAMICAL CHARACTERISTICS

The multi-connected model of a system “engine - mounting - airframe” was elaborated by using generalized dynamic characteristics (such as dynamic compliance, mechanical impedance…) in attachment points [2].

The multi-connected dynamic model of the system «Engine-mount-airframe» can be studied by dividing it into independent subsystems, reaction forces being applied in the separation points. Then the differential equations for the displacements of separation points are written down, where the generalized dynamic characteristics (for example, dynamic compliance) are used as factors of proportionality between dynamic displacement and forces [3].

The frequency function of engine dynamic compliances squarte matrix elements in attachment points were defined:

$$C_E = \begin{bmatrix} C_E^{11} & C_E^{12} & \cdots & C_E^{16} \\ C_E^{21} & C_E^{22} & \cdots & C_E^{26} \\ \vdots & \vdots & \ddots & \vdots \\ C_E^{61} & C_E^{62} & \ldots & C_E^{66} \end{bmatrix}. \tag{1}$$

Analysis of experimental data makes it possible to divide the frequency range of investigation into three sub-ranges characterized by certain dynamic behavior of the engine consequently each of said ranges can be provided with its special mathematical model – simple and clear enough.

When developing time function it was assumed what are rigid-body compliance is described by the relation: $C(f) = -\frac{1}{m(2\pi f)^2}$, wherein $C(f)$ - dynamic compliance, f – frequency, m - mass; compliance of elastic element is $C(f) = \frac{1}{K}$, which doesn't depends on frequency.

Since engine in frequency range from 20…40 Hz to 120…150 Hz behaves as an elastic-inertial system with large number degrees of freedom, what allow to consider this elastic-inertial system as finite set of linear independent not interacting oscillators described by equation [4]:

$$\ddot{X}_i + 2\gamma_i \dot{X}_i + \omega_{0i}^2 X_i = 0 \qquad (2)$$

where X_i - relative displacement, γ_i – phenomenological damping, ω_{0i} - eigenfrequency (fundamental frequency).

Summarizing contribution of said models into general pattern of engine and airframe behavior the numerical models of engine and airframe were developed wherein fitting parameters were selected and optimized to harmonize with experimental engine and airframe date.

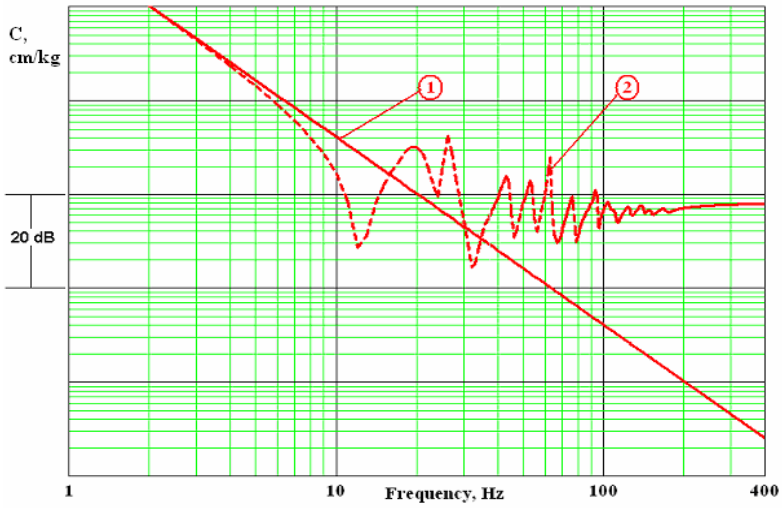

Figure 4 - The compliance module of the engine case at the attachment point.
1 – model of engine, corresponding rigid body.
2 – numerical model of engine case into account of experimental data results.

On the Fig. 4 represented in double-log scale (x-axis – frequency, y-axis - compliance) numerical model of the engine (curve 2) in one of the attachment point. Comparison obtained results for numerical calculation with engine model like it rigid body (curve 1) represents significant difference in rotor frequency band.

It can be seen from the graph what the engine model as a rigid-body is corrected is correct up to 10 Hz and supposes much more dynamic effect on the airframe from the engine, when realistic one at frequency range.

The analysis of the characteristics obtained shows that ratio of dynamic compliances (input and transient) differs considerably in the frequency range of disturbing effect.

Taking into account this ratio, it is possible to evaluate possibility of neglection of transient compliances, basing on comparison of the following matrix rates:

$$L=\sqrt{\sum_{i,j=1}^{m}\left|C^{ij}\right|^{2}}\ ;\ L^{*}=\sqrt{\sum_{i=1}^{m}\left|C^{ii}\right|^{2}}. \qquad (3)$$

where L and L^* are the Euclidean rates of the total and diagonal structural matrices.

Then the value of $\alpha=\left(L-L^{*}\right)/L^{*}$ can be identified as a system coupling coefficient.

The made calculations convinced [3] that at its sufficiently little values (less the 0,5 %) the “Engine-Mounting-Airframe” complete system disintegrates to m – disconnected systems, which analysis is substantially simplified.

In case of dynamic independence of engine mounting attachments the equation for dynamic forces, acting from the side of the engine at i-coupling point, can be reduced to the following form:

$$R_E^i(f)=\left[C_{EM}^i(f)+C_{AM}^i(f)\right]^{-1}\cdot\sum_{k=1}^{m}C_{ES}^{ki}(f)\cdot F_E^k(f), \qquad (4)$$

where the expression $\sum_{k=1}^{m}C_{ES}^{ki}(f)\cdot F_E^k(f)$ characterises engine vibration activity and is the engine excursion at attachment points (as a rule, where the standard vibration pickups are installed).

The obtained theoretical relations allow to evaluate level of the dynamic influence engine on airframe at prescribed (1 sm/s) value vibration of the engine case and estimate an expecting dynamical impact level from basic sources (residual disbalance of engine’s rotors) and other vibroactive elements installed on engine (hydropumps, gearbox, perturbations in engine’s gas-air flow duct).

SUMMARY

The performed numerical modeling displayed, that wide used models of airframe and engine are correct in narrow frequency range and not sufficiently exactly describes the dynamical interaction of airframe with engine in rotor frequency range.

The obtained numerical modelling of engine and airframe was harmonized with experimental compliances. These models are necessary for analyze and systematization of the experimental data and for calculation of vibration and noise in the cabin of aircraft at the design stage.

REFERENCES

[1] V. Baklanov, I. Golov, S. Postnov, Vibroisolation of Power Plants for New Generation Airplanes with Engines of High by-Pass Ratio, 10th AIAA/CEAS Aeroacoustics Conference, Manchester, (2004), AIAA 2004-2823.

[2] V. Baklanov, A. Zayakin, E. Orlenko, S. Postnov, The calculation of structural noise in cabin for aircraft with high-by-pass ratio engines, 11th AIAA/CEAS Aeroacoustics Conference, Monterey, (2005), AIAA 2005-3034.

[3] Baklanov V.S, Vul V.M., The influence of connecting fluctuations of complex dynamic systems on the evaluation of vibroisolation effectivenes", X Acoustic Conference, Moscow, Acoustic institute, (1983), pp.65-68.

[4] M. I. Rabinivich, D. I. Trubeckov, *Introduction To The Vibration And Wave Theory*, (Izhevsk, RCD, 2000).