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Vibration activity characterized by vibration parameter in a chosen place of the casing is an important operational parameter of the helicopter powerplant engine.

As a rule in the beginning the engine batch production faces the problem of the rejection by vibration.

Thorough research of this issue has led to the development of the parameter design method minimizing the rejection and permitting the creation of the construction with the designed vibration level.

Vibration parameter image of the engine type in a form of probability distribution of all the possible vibration values in the products manufactured from the parts without any deviation from the documentation is a starting design point.

Permittable percent of parts rejection by vibration parameter (vibration level) in the process of manufacturing serves as an objective function. The objective function is achieved in the process of top - down design where the Parametric Activity Function (PAF) stipulating the dynamical behaviour of a construction is a core.

PAF is built as a chain of argument - function dyads. Each function of the below-located dyad is an argument of the above located dyad. The construction unit of an engine corresponds to its dyad.

The procedure is expected to use at design temp.

Vibration activity, characterised by vibration parameter level in the selected point of the casing is an important operation parameter of engine helicopter of power plant.

Harmonic movements of aviation engine casing masses are brought off both according to trajectory-closed oscilations mechanism under the action of the rotary vector forces of and at the expense of mass kinetic energy transition to construction deformation potential energy with mass return to the initial position due to it's stiffness. Accordingly, oscillation movements of engine masses participate in general and local vibrations.

In Russia vibration level in places of engine attachment to airframe is requilated by the state standard requirements to limit excitation transition to the airframe.

These levels, according to the existing documentation, shall not exceed 30...50 mm/sec depending on development stage, signal frequences and engine operation mode (rated and unsteady).

Analogous protective requrements are valid for the accessories installed on the engine: vibration shall not exceed - 90 mm/sec ($V \leq 90$ mm/sec) in the place of accesories attachment in the rotor components frequency range (because vibrations are main sourse of accesors excitation from the engine).

Thus, vibration level is introduced into the list of engine output parametres and, consequently an engine designer shall have the program of design, technological and production maintenance of the given vibration level.

At the very begining of gasturbine engines design in the USSR this problem was considered in the following way: decrease of rotors action to supports at the expence of rotors balancing. For helicopter engines it was necessary to look for the solution for rotors serviceability provision, because rotors are featured by higher levels of rotation speed, which increased greatly in comparison with the engines of previous designs as a result the problem of rotors failure due to critical rotation speeds occured.

Some helicopters engine rotor systems parameters, designed by our company are presented in the table 1, fig 1a, 1b.

Table 1

Model	GTD-350	TV2-117	TV3-117	TV3-117VMA	TV7 117	TVa-3000
•turbo compressor, rpm	25000÷45000	13350÷21400	11375÷19620	12100÷19792	20381÷30711	20514÷30193
•free turbine, rpm	22200÷24000	4320÷12250	9000÷15450	9000÷15300	17150÷17850	15809÷19656 13110÷15936
•turbo compressor, kg	12,7·	49,4·	51,6·	48,1·	74,9·	61,6·
•free turbine, kg	7,41·	27,2·	24,5·	32,2·	35,6·	33,4·
•helicopter power plant	Mi-2	Mi-8	Mi-14, Mi-24 and other	Ka252, Ka50 and other	Mi-38	Mi-38, Ka40

It's obvious from the tab.1 that the first helicopter engines designed (1956-1962) by our corporation incorporated turbocompressor rotor of 45000 rpm. Our company has carried out a great scope of theoretical and experimental works by rotor dynamic behaviour investigation, in special rigs with vibrographing and flexure and force on supports measurement.

As a result, the method of complex form rotors natural frequencies estimation, the rotors stiffness requirements securing the given compliance and damping, were developed.

In particular the design of flexible bearing supported "squirred cage" instalation with compressed oile film as damper was introduced by us on the engine for the first time.

One of the main results of the reserch is introducing into practice the rotors operating in supercritical revolutions zone with organization of self-centring regime i.c. creation of conditions, when in revolution operation zone, rotor is rotating around the main central inertia axis. It allows to limit loads on the bearing by the acceptable values and to have them constantly and operation area. The danger of rotors critical modes developing, which are now avaible on the rotations up to the idle, is reduced to minimum at the expense of damp introduction.

Efforts undertaken and chosen constructive solutions have alloweed to transmit disturbances to the helicopter which are not more than 5÷7 g for GTD 350 and then not more than 7 g for TV2-117 with turbocompressor harmonic.

Rotors dynamic behaviour of the TV2-117 product was created on the mind of experience obtained in the process of GTD-350 the product development. In the revolutions operation zone rotors of this engine are also rotating in the self-centring mode, table 4.

Design-basis and experimental methods of loads, determination, generated by rotor in supports, and determination of rotor behaviour by engine and helicopter gauge mesure have been worked out.

Design actions, substantiated in serial production by balance and measures assembly, have allowed to get vibration parameter stable values for the engines in operation.

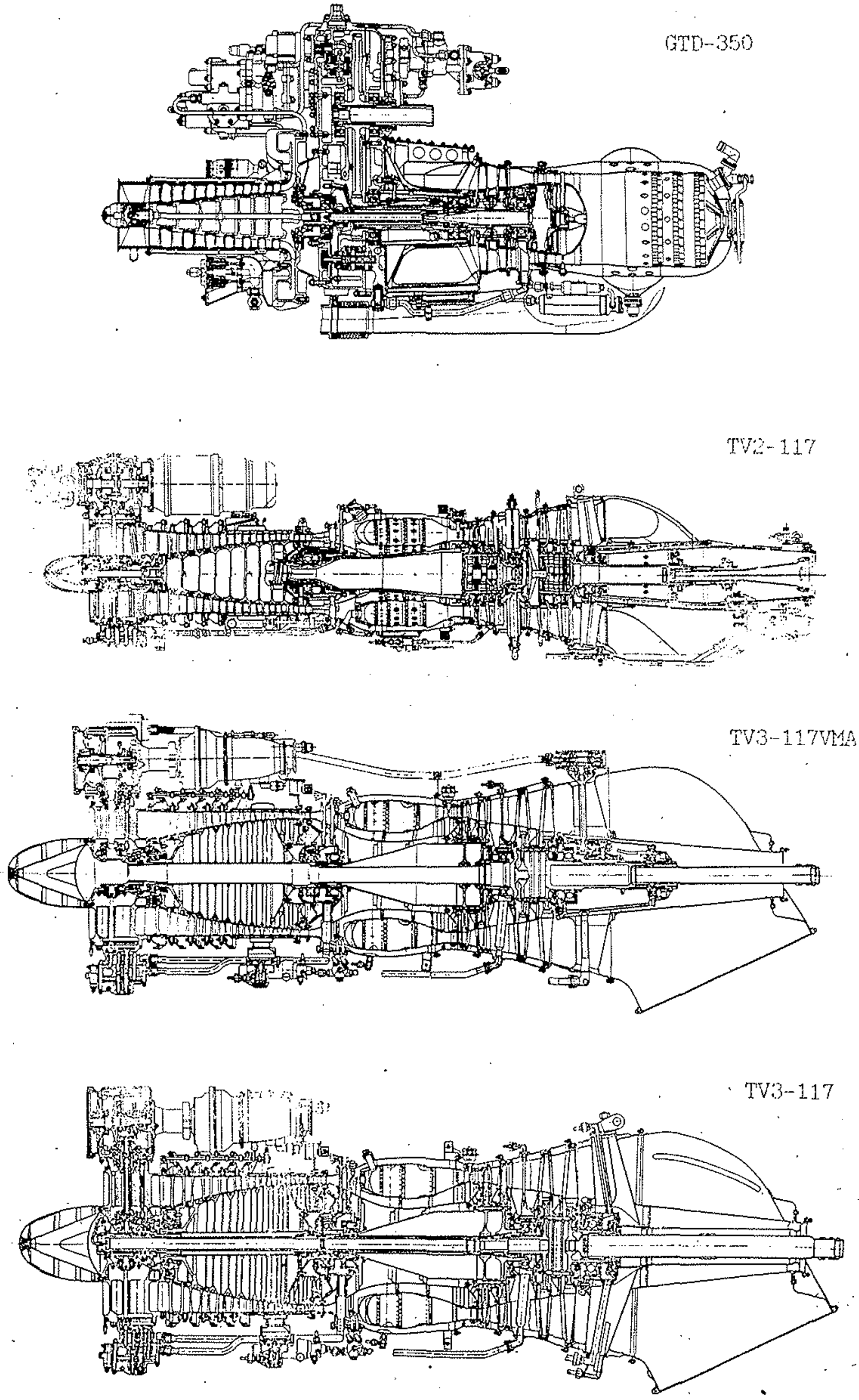


Fig.1 a

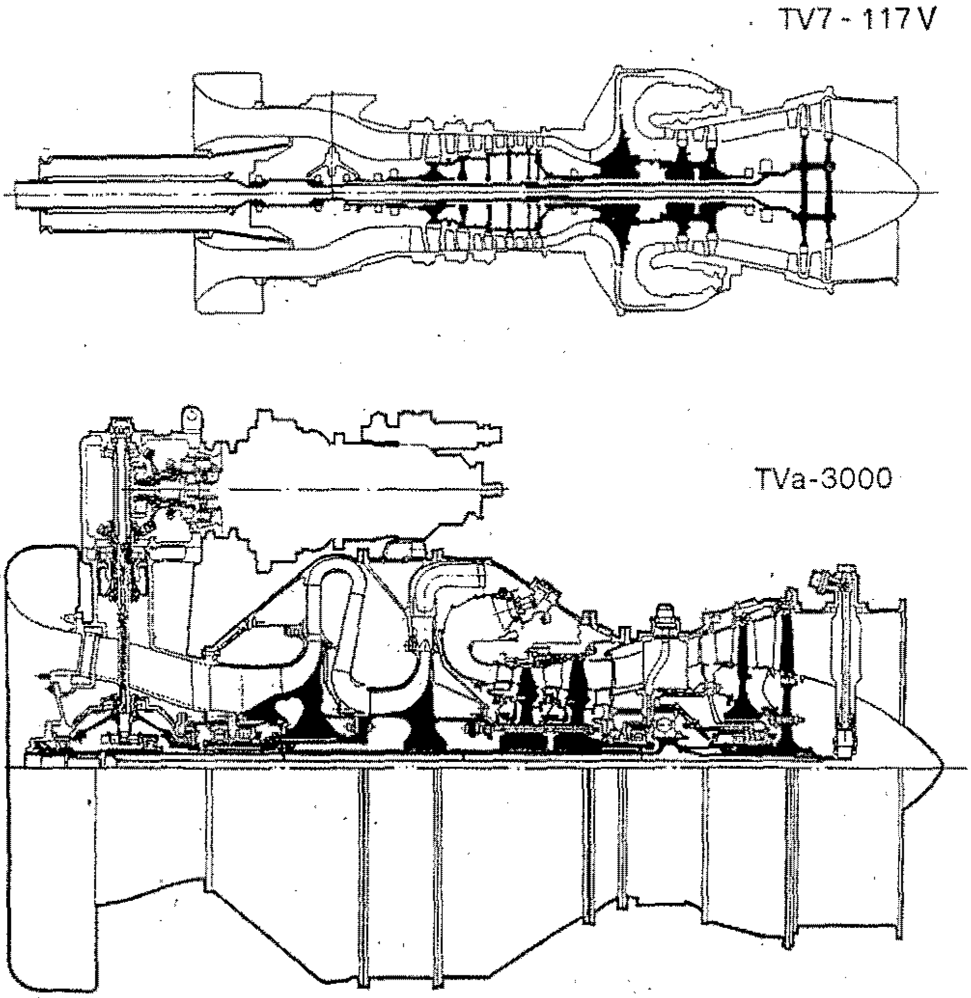


Fig.1 b

TV2-117 vibration level

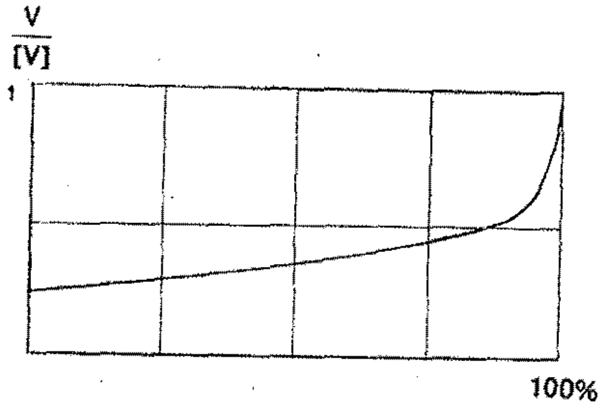


Fig.2

TV2-117 engine vibration level statistic is presented (Fig. 2) alony axes: vibration level in relation to normalized level - relative engines quantity. 85 % of engines have vibration level less then 50 % of standard level.

TV3-117 is charactirized by a sharp increase of gravity performances: engine dry mass is 14 % less than for TV2-117 , and power is 47 % more than for TV2-117.

A considerable casing mass decrease has made vibrations interpretation measured at different construction points complicated.

In this connection in the process of TV3-117 enginering more adaptable method and critical rotation speed calculation program were developed.

At allowed to increase greatly the rotors alternative development. Nevertheless the serial production has shown that some number of produced engines do not meet the requirements of non-increase of normalized vibration level.

All engines were assembled from high-quality parts and according to the drawing documentation requirements.

Detailed investigation of this problem has shown, that our understanding of engine vibration processes does not adequatly reveal the problem essence.

The essence of non-adequacy has been brought to the next aspects:

1) attention was paied to the rotor serviceability providing;
2) forces reduction on bearings down to the level, providing the bearings serviceability considered to be sufficient:
3) estimation of isolated sample was used for estimation of engine type;
4) control of the vibration parameter on the next engine life cycle stages was not included into the project development concept.

As a result of theoretical reserch available criteria and presentations were corrected. Vibration performance was expanded in this correction by some arquments.

At first, vibration parameter depends on rotation speed , where "i" is a number of rotor system.

At second, vibration parameter depends on time at specified rotation speed as parameters.

At third, the field of vibration characteristics, got for machine batch, conforms to the definite vibration parameter dependence specified rotation speed and to specified operation time.

As it is seen from the graph Fig. 3. a the spread of different engines samples initial condition by vibration parameter is observed including the point $t=t_o$ conforming to the moment of engine launching.

Thus, it is expedient to introduce the type engine parametric description in the form of vibration le vel distribution, admitted by the natural frames of the engine construction, produced within the technical documentation requirements. This description [distribution] may differ from experimentally, revealed description in the machines batch, in particular, the difference is as follows:

- in non--conformation of real vibration dependence on revolutions referring to expected function;
- in non-agreement of the expected value at the moment $t=t_o$;
- vibration parameter deviation from the expected value, including falling outside beoynd the parameter limit values.

In order to work out the criterion, suitable for using during designing, it is important to at the expence of what structural features this non-agreement may take place.

Already this initial analysis shows, that for the article quality determination in comparison with forecast quality, it is neccessary to introduce additional indicator characterizing the articles vibration condition.

We have introduced the next indications Fig. 3 :

- realization (vibration parameter) - parameter's value on the article sample - Y(t);

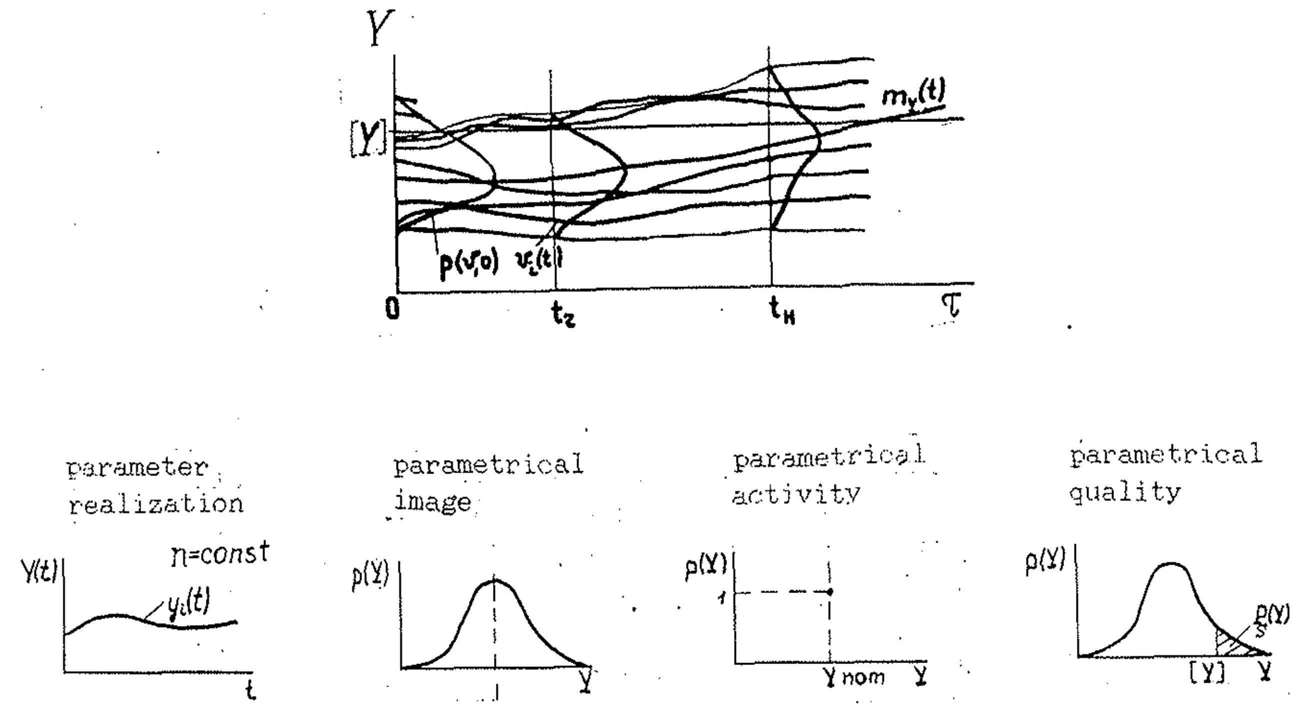


Fig. 3

- engine type parametrical activity (vibration activity) - expected parameter realization in the article, made according to so called nominal project, i.e. on the ideal article with sizes without any deviations from nominal sizes - $V_{nominal}$ PAF (parametric activity function;
- construction parametrical image (vibration description) - parameters values in machines batch in the form of realisation distribution, admitted by the article PIF (parametric image function) of the expanded design, i.e. the article within the drawing tolerance - p(V);
- design parametrical quality - (vibration quality) - is a function of parametrical image - probability of the given level increase by the parameter Ps(V).

As it is seen the engine type model estimation according to the enumerated parameters kit does not coincide with the estimation by the fact of non-increasing of the norm by the vibration parameter.

So, vibration activity is a value péculiar to the design.

Having in mind the details of vibration parameters measurements on the casing, the level of vibration charachteristics at the given point is convenient to define by force and response on it as a sum of twine-term components (dyada)

$$\bar{Y}_{N\nu} = w\sum_{\nu=1}^{K} \bar{R}_{\nu} \cdot A_{N\nu} + w\sum_{\nu=1}^{m} \bar{R}_{\nu} \cdot B_{N\nu} \qquad (1)$$

where: R_ν - forces activy on the casing from rotor side.

$A_{N\nu}$ - effect coefficients at the casing point from the action force applied to the **y** point and causing the casing total vibration.

$B_{N\nu}$ - effect coefficients in N point of the casing from the force action applied to the ν point and causing the casing local vibration.

The second part (1) might be essential for interpretation of vibrations measurements results, however it does not characterize the engine as an aircraft exitation source.

The first part (1) describing the general vibrations is convenient to PAF.

Rotor has two parameters effecting the engine operability: the forces in the supports and the outline deviations from the axis.

The force effects the vibration state; deviation from the axis effects operatility.

It is natural to wish to describe PAF by the simplist possible way.

The desire to clear PAF from not always exactly defined in practice terms permits to define the expression reasonably for the component responsible for design response to force action:

$$A_{N\nu} = \sum_{j=1}^{N} \frac{Y_j(x_\nu) \cdot Y_j(x)}{w_j^2 (1 - \xi_j^2)},$$

where: $Y_j(x_\nu)$ - an ordinate of the "j" orthonormal natural vibration form of the casing at the point with ordinate x=x ;

w_j - "j" natural frequency;

ξ_j - w_j/w detuning coefficient;

w - excitation frequency.

The effect coefficient expression purposely does not contain a term with resistance.

We prefare that it is necessary to carry out an integrated damping evaluation which may be performed with less expenses than the reliable damping evaluation in the local part.

It is evident that the casing estimated model is accepted in the beam form that fully sufficiently reflects the helicopter gas turbine engine systems dynamics, made without the intermediate gear box introduction to the engine arrangement.

First approach in PAF component responsible for R_1 forces with are generated by the rotor in the supports, follows from the oscillation process form in which the rotor is involved at the engine operating conditions.

In other words to define the loads it is necessary to know the rotor behavior and the conditions in which the contruction operability is not broken.

Rotor-stator shift function may be an integral indicator of the rotor-stator quality system.

The factors, effecting this function are assembled into the block-diagram fig. 4.

In figure 5 there are given the blocks decodings fig. 4.

The factors indicated on the PAF are actually found out and described according to the results of GTD finishing, icluding the results of our company experience by the helicopter engines.

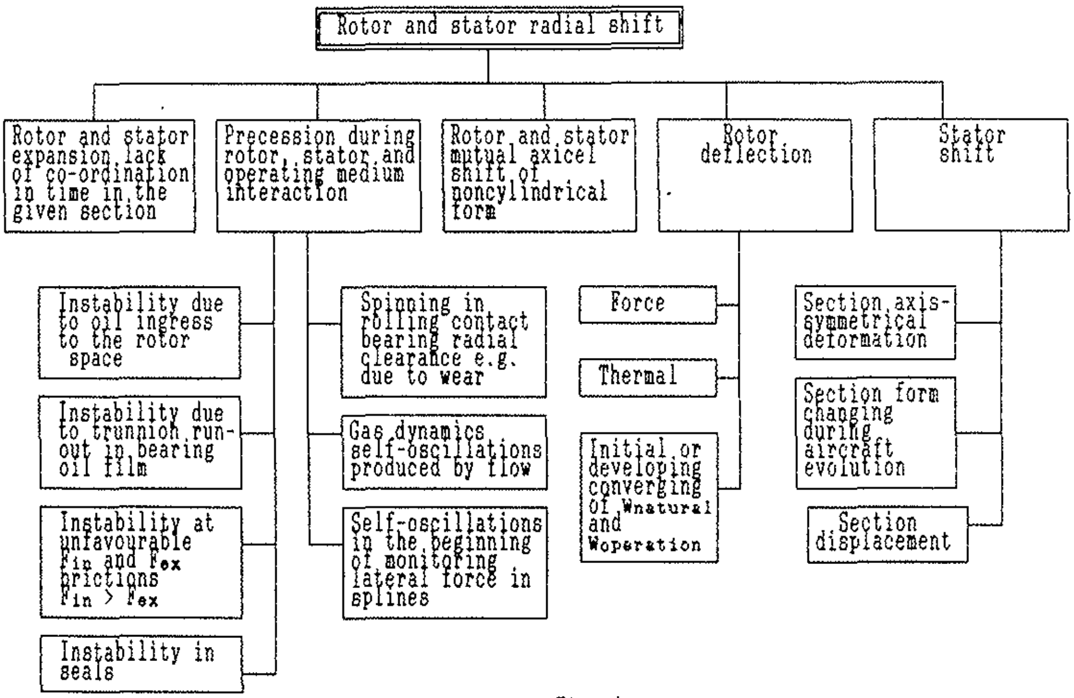
It is to avoid their simptom on the engine if at the start of the engine development constructive decisions which prevent undesirable processes will be taken into account.

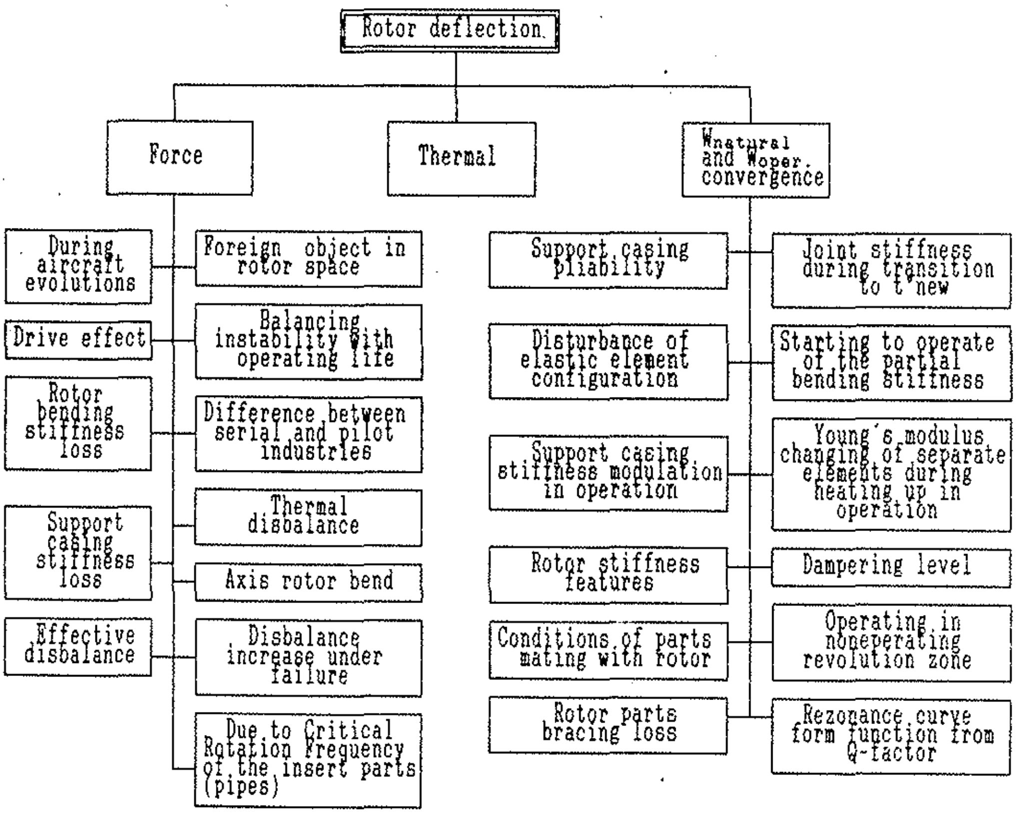
These factors may appear at any stage of the engine creation: designing, construction, manufacture, finishing, operation if they outstay the engineer's mind.

From the point of view of the creative process at each of these stages an engine presents different products of the designer's activity.

In fig. 6 in elaborated terms the scheme of design function on engine development is given where for each intermediate product the scheme of its production and the way of its decription trough the design evaluation are designated.

At operation the engine as a type is presented by a hardware complex, where each device is characterized by a number of output parametrs including realization vibration level.


Rotor and stator radial shift
Rotor and stator expansion lack of co-ordination in time in the given section
Precession during rotor, stator and operating medium interaction
Rotor and stator mutual axicel shift of noncylindrical form
Rotor deflection
Stator shift
Instability due to oil ingress to the rotor space
Instability due to trunnion run-out in bearing oil film
Instability at unfavourable Fin and Fex frictions Fin > Fex
Instability in seals
Spinning in rolling contact bearing radial clearance e.g. due to wear
Gas dynamics self-oscillations produced by flow
Self-oscillations in the beginning of monitoring lateral force in splines
Force
Thermal
Initial or developing converging of Wnatural and Woperation
Section axis-symmetrical deformation
Section form changing during aircraft evolution
Section displacement
Fig. 4


Rotor deflection
Force
Thermal
Wnatural and Woper. convergence
During aircraft evolutions
Drive effect
Rotor bending stiffness loss
Support casing stiffness loss
Effective disbalance
Foreign object in rotor space
Balancing instability with operating life
Difference between serial and pilot industries
Thermal disbalance
Axis rotor bend
Disbalance increase under failure
Due to Critical Rotation Frequency of the insert parts (pipes)
Support casing pliability
Disturbance of elastic element configuration
Support casing stiffness modulation in operation
Rotor stiffness features
Conditions of parts mating with rotor
Rotor parts bracing loss
Joint stiffness during transition to t°new
Starting to operate of the partial bending stiffness
Young's modulus changing of separate elements during heating up in operation
Dampering level
Operating in noneperating revolution zone
Rezonance curve form function from Q-factor
Fig. 5

STRUCTUREL OF DESIGN ACTIONS

Principles:

- Engineering material object conservatism
- Reproduction with deviations
- Engeneering evolution continuosity
- Multiaspect structure
- Adoption of solution in real time
- Reaction to fast changing medium

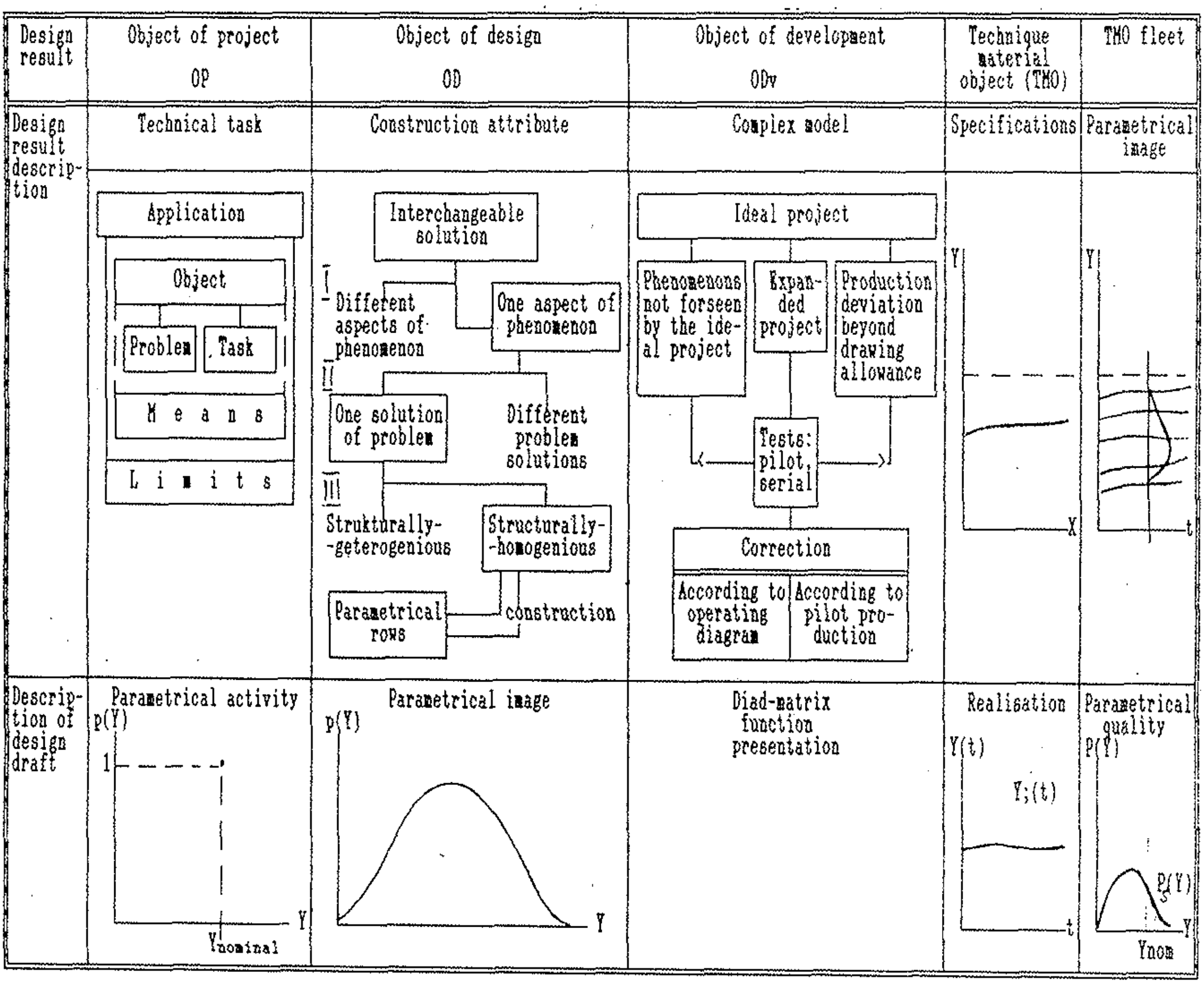


Fig.6

The block diagram presents the isolated structure which is the exponent of the chosen parametrs.

The scheme is universal becouse it may be adapted to each of designe part at hierarchic engine construction.

Some explanations are required block-diagram of the activity form in the development of OP - object of project, OD - object of design, ODv - object of development.

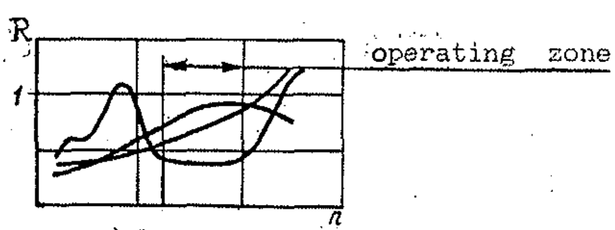


Fig.7

Disregard of difference in designe objekt interpretation leads to such cases when one try to disigne the objekt according to rules of other type.

This is the mostly frequent reason of noncoordination, blunders, mistakes and unattainability of results.

For OP - it is the conception development and the diagram corresponds the formal signetificance of the invention as it is required by the patent formula of German and Russian type.

Activity by OD creation is subordinated to the constructive decisions optimisation for what they, are classified in conformity with the principles of interchangeable decisions.

By this the form (type) of the outlet parameter is defined by action of I - horizontal and its numerical parameters - III. At this stage evaluation is a paramethrical image.

ODv product - finishing object is born in the result of validation tests.

There are revealed some accepted technical decisions and design execution which are separetely stable in reference to the production, combined in the Power Plant system, etc.

As a result the complete model of interaction of the engine assemblies and systems and helicopter Power Plant is revealed.

So the system appoach based on the introduced criteria and the given schemitization of the oscillatory motion presumes:

- creation of the analytical models sets of the structure dynamic behavior;
- each of the models communication with constructive features;
- model selection of the desired dynamic behavior;
- itroduction only those constructive features which correspond to the desired model of rotor behavior into diagram the engine dynamic;
- analytical model design of specified construction;
- check of correct tolerance by criterion of parametrical quality.

In other words during the design synthesis we are "dressing" the analytical model constructively.

During selection of the proper criterion - special objective function the consecutive chain of the estimated criteria is all intermediate design objects being formed up from top to bottom, up to the detailed dimensions.

Let's follow flig succession on the example of the rotor forces formation in the supports at unbalanced source exitation.

The view of this rotor load characteristic is presented in fig. 7.

Resonance speeds location in respect of the operating zone is defined by the rotor support, elastisity value; efforts value - by passage speed of rotations of resonance zones and their locations, damping ratio, dynamic system reconstruction at the possible clearances selection and etc.

Let's consider the selection of the elastic element structure of the support (fig.6 OD). The II gorizontal, big as a result of selection: pliancy at the expense of elasticity of the mechanical structure, in the process of selection structures based on the physical princeples: hydrodynamic, electromagnet, gasdynamic, magnetic etc were rejected. The possible variants are presented in the table 2.

Compliance value (support output parametr) is specified from location condition of critical rotor r.p.m. at at required rotational speed.

Within the model frame of the unbalanced forces source load value on the support is the function of construction factor:

$$R_v = \sum_{i=1}^{s} K_{vi} \cdot Z_{vi}$$

$$Z_{vi} = f\,(D, \boldsymbol{\delta}, \boldsymbol{\delta}_k);$$

$$K_i = f_1\,(M,\ A,\ B,\ \frac{l_{c.g.}}{L},\ w);$$

where: D – rotor residual unbalance;
δ – radial clearance in a bearing;
δ_k – dynamic pliancy of the elastic support;
M – reduced rotor mass;
A, B – rotor inertia mass moments
w – operational revolutions;
$\frac{l_{c.g.}}{L}$ – relative length of rotor dimensions;
S – quantity of variables effecting transverse load in supports y.

The nature of the load proceeding by the rotor rotation frequency is different depending on either these rotations are taken place before critical rotation frequency in the rotations operating zone – for our designs this is a zone of rotors selfcentralizing.

However the model of the unbalanced nature source the PAF has the following appearance:

$$\bar{Y}_N = w \cdot \sum_{\nu=1}^{K} \cdot R_\nu \cdot A_{N\nu}$$

As it is known the parts and units dimensions have a scaffer and the real values for example: length, diameter, unbalance do not correspond to the ideal design.

Table 2

INTERCHANGEABLE DECISIONS
AT THE GIVEN LEVEL OF DISBALANCE FORCES

Used side of phenomenon	Proplem decisions	Technical solution	Execution
1. Rotor stiffening	1. "Flexible" shaft	1. Discret elastic inclusions	1. "Allison" ring
2. Nonlinear "clearance"	2. Linear elastic supports with elasticity above bearing	2. Elastic element "squirrel ring"	2. Ring with millings and plains
3. Energy dissipation	3. Linear elastic supports with elasticity below bearing	3. Elastic ply (metal-rubber)	3. Smooth ring
4. Self-alignmen	4. Nonlinear elastic supports with modulation by disbalance forces	4. Pressed-out oil film	4. Ring with welded on bosses
5. Nonlinear stifiness in support	5. Nonlinear elastic supports with monitoring system	5. Annubar elastic element	5. Rings kit with millings
6. Spreading of forces along		6. Beam or annubar section jacket	

Therefore the engine vibrotype is presented in a form of p(Y) parameter distribution – analytical description taking into consideration the effect of the constructive parameters scattering and named the parametrical image function PIF.

The Comparative investigations was shown that $A_{N\nu}$ sufficiently stable to disturbances and they may be considered constant from sample to sample of the engine.

It is different with the loading system.

In spite of the fact that each sample has the same quantity of forces, acting to the casing nevertheless there is its own forces system acting on each sample. As each vector in the support may be turned by φ angle, the same time it differs by $R_y(x)$ value.

The total system of rotor loads the casing has been formed from the random vectors with the random location by the angle

$$\overline{R}_v = R_j(x)\cdot e^{i\varphi_v}$$

R_y and φ_v values are changing from one sample to another.

Below there are given distributions of the intermidiat variables (arguments):

- radial clearance in bearings

$$P(\delta) = \frac{6}{\sqrt{2\pi}\,(\delta_2-\delta_1)}\cdot e^{-\frac{18\left(\delta-\frac{\delta_1+\delta_2}{2}\right)^2}{(\delta_2-\delta_1)^2}}$$

- the residual unbalance in the support

$$P(D) = \frac{D}{[D]^2\left\{1-\left(2+\frac{\Delta}{[D]}\right)\cdot e^{-\left(1+\frac{\Delta}{[D]}\right)}\right\}}\cdot e^{-\frac{\Delta}{[D]}}$$

using the test data for small size gas turbine engines

$p(D) \approx 1,02\cdot D\cdot e^{-D}$, where

Δ - residual unbalance change value from [D] value admitted by the design documentation;

- pliancy of the elastic support with the allison ring

$$p(\delta_K) = \frac{1}{\sqrt{2\pi}\sigma_K(\delta_H;q;s)}\cdot e^{-\frac{(\delta_K-\delta_H)^2}{2\sigma_K^2(\sigma_H;q;s)}}$$

according to the serial production experience of the ring restoring elements (RRE) there are obtained:

$q(\delta_H) = 16,03-1,36\cdot 10_4\cdot\delta_H$, where

- the nominal value of support pliancy with RRE;

$q(\delta_H)$ - relative scaffer of elastic support pliancy with RRE from its nominal value;

S - relative quantity of supports with RRE, pliancy of which is within the limits of the calculated tolerance field;

$\sigma k(\delta_H, q, S)$ - midirotractor deviation of the elastic support pliancy with the RRE defined by Laplas function:

$$\Phi\left(\frac{\delta_H q}{\sigma_K\sqrt{2}}\right)=S$$

Distribution of the radial force modulus of the small size gas turbine engines rotor systems supports depending on the combination of the initial effect factors on the support.

The total statistic dependency for the radial load modulus has the following form:

$$P(R_\nu) = \begin{cases} \dfrac{1.02}{K} D_\nu e^{-D_\nu} \\ \dfrac{1.02}{K_\nu} \cdot e^{\frac{R_\nu}{K_{\nu_1}}} \cdot \left\{ \dfrac{R_\nu}{K_{\nu_1}} \cdot e^{\left(m_\nu t_\nu + \frac{\sigma_\nu^2 t_\nu^2}{2}\right)} \cdot I_{11} - t_\nu \cdot e^{\left(m_\nu t_\nu + \frac{\sigma_\nu^2 t_\nu^2}{2}\right)} \cdot \left[\left(m_\nu + \sigma_\nu^2 t_\nu\right) I_{11} + \dfrac{\sigma_\nu \sqrt{2}}{\sqrt{\pi}} \cdot I_{22} \right] \right\} \\ \dfrac{\delta_{k\nu}^2}{K_{\nu_k}} P(\delta_{k\nu}) \end{cases}$$

where:

- the first expression corresponds to the load in the stiff support without running-in;
- the second - in the stiff support with running-in;
- the third - in the elastic support.

$$I_{11} = \frac{1}{2}\left\{ \Phi\left[\frac{\delta_{1\nu} - \left(m_\nu + \delta_\nu^2 t_\nu\right)}{\delta_\nu \sqrt{2}}\right] - \Phi\left[\frac{\delta_{1\nu} - \left(m_\nu + \delta_\nu^2 t_\nu\right)}{\delta_\nu \sqrt{2}}\right] \right\}$$

$$I_{22} = -\frac{1}{2}\left\{ e^{-\frac{\left[\delta_{1\nu} - (m_\nu + \sigma_\nu^2 t_\nu)\right]^2}{\sigma_\nu \sqrt{2}}} - e^{-\frac{\left[\delta_{2\nu} - (m_\nu + \sigma_\nu^2 t_\nu)\right]^2}{\sigma_\nu \sqrt{2}}} \right\}$$

It is possible to anticipate that in serial engine production distribution of the vectors of loads for a batch of engines in each section of the forces application is uniform by angle:

$$P(\varphi) = \frac{1}{2\pi} \qquad 0 \le \varphi_\nu \le 2\pi,\ \upsilon = 1,2 \ldots k$$

The law of the vibration velocity vector modulus distribution in the standard point at the casing excitation by loads PIF, being trasmitted by means of "k" rotor supports:

$$p(V_{\Sigma R}) = \begin{cases} 0 & V_{R_1R_2\ldots R_K} \le 0 \\ \dfrac{2V_{R_1R_2\ldots R_{K-1}}}{\pi\omega\left|A_{NK}\right|} \displaystyle\int_0^\infty p(V_{R_1R_2\ldots R_{K-1}}) \cdot \left\{ \dfrac{\pi}{4}\left[\dfrac{p_{R_K}\left(\dfrac{V_{R_1R_2\ldots R_{K-1}} + V_{R_1R_2\ldots R_K}}{\omega\left|A_{NK}\right|}\right)}{V_{R_1R_2\ldots R_{K-1}} + V_{R_1R_2\ldots R_K}} + \dfrac{p_{R_K}\left(\dfrac{V_{R_1R_2\ldots R_{K-1}} - V_{R_1R_2\ldots R_K}}{\omega\left|A_{NK}\right|}\right)}{V_{R_1R_2\ldots R_{K-1}} - V_{R_1R_2\ldots R_K}} \right] + \right. \\ \left. + \dfrac{1}{2} \displaystyle\int_{V_{R_1R_2\ldots R_{K-1}} - V_{R_1R_2\ldots R_K}}^{V_{R_1R_2\ldots R_{K-1}} + V_{R_1R_2\ldots R_K}} \arcsin \dfrac{V_{R_K}^2 - V_{R_1R_2\ldots R_{K-1}}^2 - V_{R_1R_2\ldots R_K}^2}{2V_{R_1R_2\ldots R_{K-1}} \cdot V_{R_1R_2\ldots R_K}} \times \dfrac{p_{R_K}\left(\dfrac{V_{R_K}}{\omega\left|A_{NK}\right|}\right) - \dfrac{\partial p_{R_K}\left(\dfrac{V_{R_K}}{\omega\left|A_{NK}\right|}\right)}{\partial V_{R_K}}}{V_{R_K}^2} \cdot dV_{R_K} \right\} dV_{R_1R_2\ldots R_{K-1}}, & V_{R_1R_2\ldots R_K} > 0 \end{cases}$$

where V_{Rk} – the vibration rate vector modulus at the standard point on the case of engine when it is subjected by the one R_k load vector;

$V_{R..Rk} = V_{\Sigma R}$ – the sum vector modulus of k random vectors;

$P_{Rk}(a)$ – the distribution law of sum vector of "k" random vectors by "a" argument;

A_{NK} – the structure responce at the specified point – influence coefficient of case constructure at N vibration by N_{own} shapes at the point with k coordinate.

The built vibration velocity distribution reflects the vibration state of the small size gas turbine engine with one frequency excitation.

Probability density of the generalized vibration velocity parameter in the standard point on the engine casing with two frequency excitation:

$$P(V) = V\int_0^V P_2(V_2)\cdot P_1\left(\sqrt{V^2 - V_2^2}\right)\cdot \frac{dV_2}{\sqrt{V^2 - V_2^2}}$$

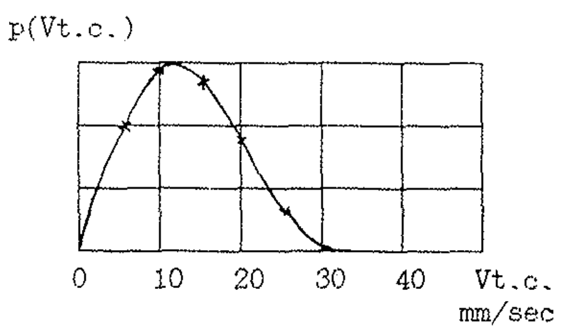
Data illustrating the influence of the force system change on the level of vibration in the engine casing standard point are given in the table 3. The results for force systems, which vector modules are constant and only their mutual angle location is beeing change.

Table 3

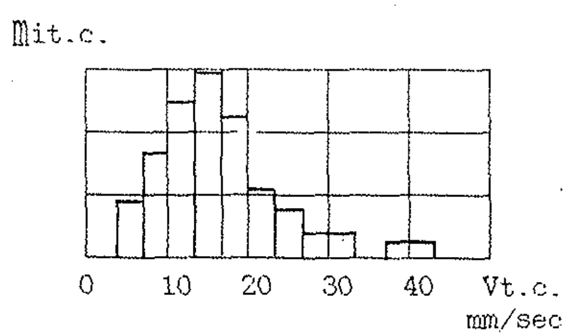
Rotor	Engine type 1 Vmax	Engine type 1 Vmin	Engine type 2 Vmax	Engine type 2 Vmin
rpm	mm/sec		mm/sec	
21180	27,2	16,4	2.0	19,4
(max) 31165	0,3	0,3	0,26	2.76

It means that scatter of realization is possible at the same type engine rating by several times and also at different rotation speed at the same engine.

Evalution of the procedure of comparison of the design parametric descriptions with the vibration distribution bar chart in the standard point of the serial produced casing excited by rotors of different structures is presented below. Fig. 8.



Calculation



Test

Fig. 8

Methodological provision of works yering to obfain turbine engine improved vibration perfomances by GTD, using self-alignment effect

Table 4

Method essence: Designing of conditions for rotor system self-alignment in the revolution operating zone.

Abstraction: Multisupport beam with concentrated masses; supports dynamic elasticity is taken into account.

Calculating facilites: Computer.

No	Action stages	Target	Methodical provision
1	2	3	4
1	Construction preliminary analysis.	Excluding of potential vibration sources.	Comparison of design decisions with available ones.
2	Subsystem estimation and rational placement of stiffness along rotor length.	Rotors' potential serviceability in system.	Natural frequencies and forms calculation; alternative diagrams taking into account natural forms.
3	Placement determination to self-alignment area revolution of rotor system on rigid supports	Insight regading rotor system properties.	Natural frequencies calculation, natural form construction.
4	Overlapping of self-alignment area with operating revolutions, zone; choice of supports elasticity design values.	Provision of engine operation without elevated loads.	Calculation of natural frequencies and forms and critical modes diagram.
5	Borders determination of allowable spread in design elasticity values.	Influence estimation of spread in stiffness parametres of rotor system on its dynamic behaviour.	Calculation of natural frequencies and forms and critical modes diagram.
6	Parametres selection of elastic supports elements: a) quantity determination of casing support elasticity; b) value calculation of bearings linear elasticity; c) value determination of additional elasticity which provides support elasticity design value and calculation of elastic element structural sizes.	Obtaining of supports elasticity design values on engine. Comparison of construction (structure) stiff performances with design value. Provision of design parametres and dynamic properties of rotor system.	Experimental installation. Method of linear features receival. Calculation of methods of elastic elements parametres according to the given support elasticity value.
7	Full scale engine testing.	Receivaly of vibration perfomances.	Test bench.

Positive aspects of the method.

on design stage: forced oscillations calculations and system damp properties investigation are not required;

on production stage: it is possible to be limited to rotor balancing as a solid body;

on operation stage: bearing increased life and increased engine reliability are provided.

It is revealed that

- a certain percent of pickoff by vibration is caused by the constuction itself;
- estimated calculation coincides with the results of the statistic level processing of the produced samples of the same design;
- values of the absolute maximum vibration velocity and values of vibration quality criterion, calculated by the design documentation data and obtained by the test results practically coincide.

Parameter / Source	Vmax p.t. mm/sec
Calculation	65
Series	68

Work on vibration begins since the moment of engine preliminary outline.

In practice the phase of construction preliminary synthesis precedes the work on vibration activity control scheme. The aim of this phase is to eliminate phisical effects the action on which is hindered. The mentioned phase proceeds according to PAF fig. 4, 5. Construction realization of these measures is varied due to engine type.

In fig. 9 vibration activity control scheme within the limits of disbalance excitation model is given. Designation on scheme are as follows:

$\alpha_r, \beta_r, \gamma_r$ - rotor stiffness coefficient.
$\alpha_c, \beta_c, \gamma_c$ - casing stiffness coefficient.
α - coefficient characterizing beam end angular shift. Unit moment acts on the beam end with (at) the embedded other beam end.
β - coefficient characterizing beam end angular shift, on which unit force acts the other beam end being embedded.
γ - coefficient characterizing beam end radial shift, on which unit force acts the other beam end being embedded.
ω_r - rotor natural vibration frequency.
ω_c - casing natural vibration frequency.
Vn - assigned vibration speed value.
Ar, Ac - engine rotor and casing relative vibration amplitudes at the point under investigation (vibration forms).
l - engine casing length.
Vc - engine vibration at the point under investigation along casing length.
Ri - radial force modulus variable component. The radial force is excited in "i" support.
p(Ri) - radial force modulus variable component distribution in "i" support.
Δ - tolerance (distribution value allowable by drawing documentation).
p(V) - vibration output parameter distribution in the standard point of the casing (vibration activity description of the designed construction).
Ps(V) - probability of serviceability level "S" by vibration speed realization. $P_S(V) = 1 - \int_0^S \rho(V)dV$
[Ps] - the given value of quality criterion.
$n_r \geq n_{min}$ - checking of margin satisfying by the given critical rotation speed.

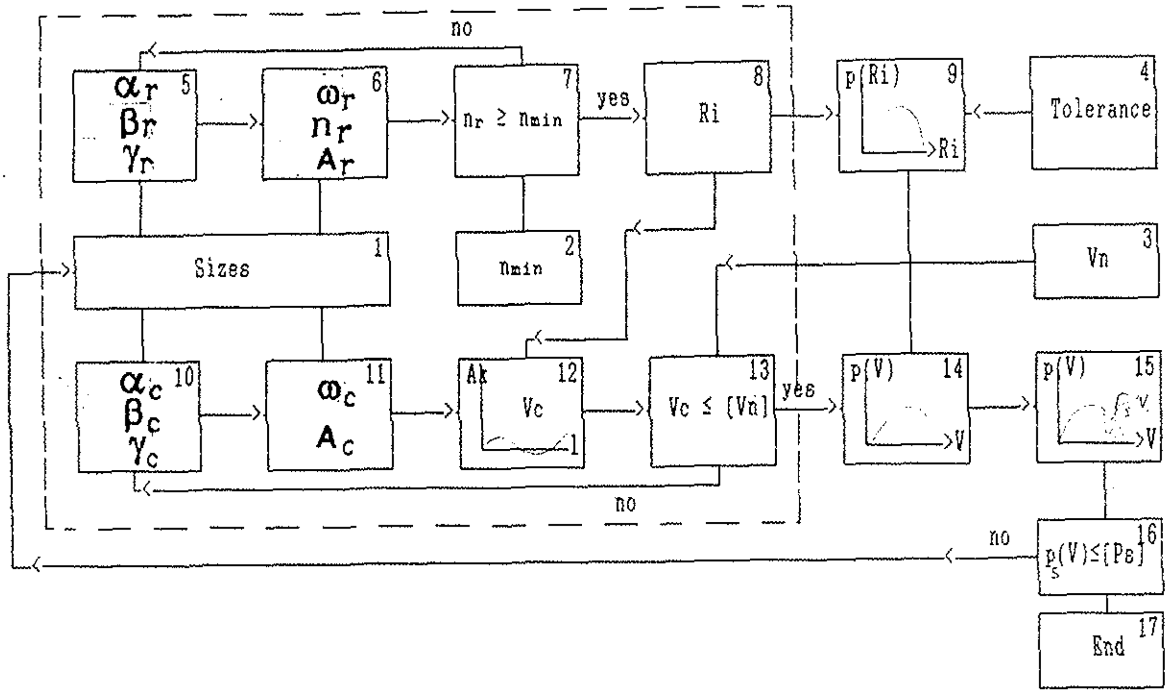


Fig. 9 Vibration activity control block diagram.

In accordance with PIF standardized value of limiting percent of product removal due to vibration is entered in 15 evaluation block.

Estimating block 15 comprises normalized value of allowable per sent of article rejection due to vibrations.

Network is closed within this block by comparison of criterion value given and obtained by construction variant under consideration in block 16.

Information coming to the block of estimation is generated in block 14 of vibration speed spread calculation. Forces spread data in rotor supports are introduced into this block as initual data from block 9. Influence coefficients values are introduced into this block from block 11.

Influence coefficients calculating block is the end-point in a group of blocks, which describe casing natural vibrations. Block 10 (casing stiffness coefficients calculation) and block 11 (casing natural vibration frequency and vibration forms calculating) are included into this group.

Block 9 is an end-point in the group of blocks determining forces in rotor supports depending upon rotor system structural configuration.

Information regarding rotor received from group comprising blocks 5,6,8 is the initual data for block 9 operation.

Rotor stiffness performances (block 5) are calculated in this group of blocks; rotor natural vibration frequences and vibration forms (block 6) are determined in this group of blocks; margins by rotor system critical revolution frequency are calculated and in case $n_r \geq n_{min}$ is not met then the cycle of initual data change and calculation is repeated untill the restriction is satisfied. Then block 8 initiates its operation to calculate forces.

The other internal cycle is closed by comparison of vibration speed value in the standard point of casing (block 12) produced by nominal forces action from block 8 with the normalized vibration speed value from block 13. Output from this cycle is accomplished by comparison with the normalized vibration value (block 13). Parameters distribution recording is done by introducing of drawing tolerance from block 4.

Value of construction response to the casings natural frequences is evaluated with the help of performance coefficient (factor), the value of which is defined on the base of experience.

Limit frequency ($w_i \pm \Delta w_i$), on which the construction response is accepted neglecting damping

$$V_{w_i \pm \Delta w_i} = 0{,}7 \cdot V_{w_i}$$

depends upon system soundness:

$$Q = \frac{w_i}{2\Delta w_i}$$

For each of w_i Δw_i is defined by Q.

In its turn the coefficient effect values in the casing standard point were obtained for wi by the diagram of influence coefficient effect of frequency depending on the individual load in j-point along the casing length

$$A_j\ (w_i \pm \Delta w_i),$$

where:

W_i - casing natural frequency number coinciding with operating revolutions.

Taking into account the construction damping properties the same maximum evaluation for the given product by (2) : 2->$V_{w_i \pm w_i} = 0{,}7 \cdot V_{w_i}$ hase the following meaning at the most dangerous revolutions:

$$V^{\omega_i}_{+\Delta\omega_i} = \sum_{i=1}^{n} R_i^{*}(\omega_i + \Delta\omega_i) \cdot A_{Ni}(\omega_i + \Delta\omega_i) \qquad V^{\omega_i}_{-\Delta\omega_i} = \sum_{i=1}^{n} R_i^{*} \cdot A_{Ni}(\omega_i - \Delta\omega_i)$$

If each of them is less than the service ability $V^{\omega_i} \leq [V]$, i.e. that the location of the abonementioned casing system modes of vibration is allowable in the operating zone of the rotors revolutions.

Conclusion

The presented program of aviation engine vibrationstate development permited Klimov Corporation to designe engines with low level of force excitation to helicopter, to improve essentially the systemizing of engine designe and development at the expense of well-defined recomendations on icrease of resistence to damage monitoring and vibration diagnosis systems, mastering of product engines in short times.

The program is especially efficient in case it is applied from the moment of engine laying. The programm makes an impressive case in favour of Klimov Corporation policy of transition from traditional design methods to design investigations.