VIBRATION OF TURBINE ENGINES AS THE CRITERION OF PRODUCTION QUALITY OF ROTOR ASSEMBLIES AND THEIR CURRENT TECHNICAL STATE

Stefan SZCZECIŃSKI
Włodzimierz BALICKI
Institute of Aviation

The article deals with the problem of vibration of the aircraft turbine engines with special attention devoted to the unbalance stages of rotors. This problem is rarely mentioned in the specialist engine reference books. It was shown that the amount of information contained in the data got from the permanently made measurements of amplitudes and frequencies of the transverse vibration of engines, particularly in case of the multi-rotors, can not be overestimated. The relationships of these parameters to: the constructional form of rotors, the quality of their realization and assembling, the effect on the clearance changes in supports and the possibilities to detect emergency states of engines have been presented.

1. INTRODUCTION

In all technical devices (mechanical, hydraulic, pneumatic and so on) operating in the changing conditions of loading, especially cyclically ones, it is the probability to occur the resonance phenomenon. Sometimes this phenomenon is used, for example, to improve the filling up the cylinders of piston engine, by suitable tuning of the frequency of vibration of air column in the inlet pipe according to the rotational speed of engine. In other way, this phenomenon can be used to estimate the crack magnitude of compressor or turbine blades of the jet engine during operation. The determining of the natural frequency range (scatter of this frequency) of engine units can be the measure of the production quality of the given manufacturer.

Frequencies and amplitudes of vibration of the running aircraft turbine engine depend on the rotational speed of rotors and flight conditions. In the dynamic (transitory) states of operation so typical in the case of combat helicopters and airplanes, it is difficult to determine the operation ranges in which the resonance can occur particularly.

A lot of polytechnic textbooks relating to aircraft powerplants advise to avoid the even numbers of blades in the blade ring of rotating machines (rotors and guide rings of compressors and turbines) and it would be the best if they have the prime numbers of blades. It is also thought that it is unacceptably to apply the equal numbers of blades in the several stages of rotor assembly. Indeed, in the first „Dervent" and „Nene" jet engines (and their tracings) produced in series by the Rolls-Royce the number of rotor blades of compressor was equal to 29 when the number of guide veins of diffuser was 18. Such solution effectively protected compressors of these engines against the appearance of dangerous resonance of vibration. However, nowadays the two-spool jet engines are in operation (such as the D -30KU powered the Ił-62M and Tu-154M airplanes), in which the four-degree turbine driving the ventilator assembly has the identical rotor carrying disk with equal numbers of rotor blades and guide veins in all stages. Undoubtedly, this is the result of tendency to reduce the production costs of engines (to obtain the unification of assemblies). Also in the most modern engines such as the General Electric CF-6 families, the even number of ventilator blades have been applied. In this case, the constructors made possible to replace the broken-down blade (ventilator blades are particularly subjected to damage by the bird strikes or other „alien bodies" suction through engine, for example, the crumbs of concrete from landing strip) together with the second one, fixed on the opposite side of carrying disk, by the set of reserve blades. It is done without necessity to demount the engine and check the rotor balancing. These examples illustrate the fact that rotor blades and guide vanes are the source of disturbance of stream flowing through the „gas track" of engine and can force the dangerous vibration, particularly when the forcing frequencies (most often dependent from a rotational speed of rotors) are close to the natural frequency of any engine units (usually blades).

In engines of the combat airplanes these forced input functions come from the changes of flight condition (density and temperature of air and combustion gases are changed considerably due to the considerably changes of the height and speed of flight), the changeable modes of engine operation, and the action of mechanization assemblies (such as: air bleeding valves, adjustable guide rings of compressors, and mechanism to control cones and claps of the air inlets and the cross-section of escape nozzles) in the fluid-flow channel. Particularly dangerous are the cases when inlet channels of compressors are not axially-symmetrical placed. The source of similar problems can be the non-uniform temperature field of gases before the turbine or into the engine afterburner.

The above mentioned factors create the necessity to carry out experimental researches to obtain the information about the real values of vibration amplitudes which can appear during the flight. Such study can be performed with the laboratory precision in the stationary, so called, „height" test bed. The engine subjected to such tests should be placed in the complete airframe and controlled exactly in the same way as in the specific flight task, e.g. the combat mission.

It seems that both the specialist textbooks and polytechnic studies devote too little attention to effects of the unbalance degree of the aircraft turbine engine rotor assemblies on their loading and dynamic deformation. These unbalance also cause the loading of bearings, engine frames and the fixing engine units to the airframe. As well, it has not been any attempt to estimate the influence of these loading on the construction durability and the flight safety. The determination of effects of the rotor unbalancing degree on the amplitude of transverse loading of airframe assemblies is relatively simple, but it is difficult to estimate the changes in airframe durability due to this unbalance. All of this put the stress on the research workers to intensify the cognitive works in this field.

The basic problem to determine the value of rotor unbalance degree is to supply the necessary power to reach the suitable rotational speed of the entire rotor during measurements. In the manufacturing process, measurements of the rotor unbalancing are practically made at the rotational speed typically from 10% to 20 % of its nominal value. Hence, if we assume that measuring error obtained in these conditions is only ± 1%, then in a operation range the inaccuracy of a loading estimation due to the rotor unbalancing grows up to over ±25% – 100 % (because centrifugal force is proportional to the square of rotational speed). It is possible to increase the measurement accuracy of a rotor unbalance degree by location of this tested rotor in the vacuum chamber (or helium filled chamber) and increase its rotational speed to the operation value.

Starting from time of the origin of turbine engines, the compressor and turbine rotors are statically and dynamically balanced. The case of the turbine of the Armstrong-Siddley Viper ASV - 8 engines was perhaps only one exception, which the balancing relied on the shortening tongues (leading edges) of rotor blades and according to this the requirement of a static balancing was only fulfilled. This solution comes maybe from the following reasons: disk of this turbine was exceptionally flat, the number of blades was relatively large (about 100), and the balancing machines being at that time in disposal of producer were busy to perform the measurements of rotors of radial compressors used for the supercharging of piston engines.

In the extended (along axis) axial rotors of multistage compressors and turbines the preliminary static balancing of individual stages and shaft sections are made, and next after their assembling, the static and dynamic balancing of complete rotors are conducted. During engine operation on the nominal range, the real unbalance degree of rotors strongly depends from their transverse stiffness. For these reasons, the drum-disk constructions, with the possibly small span of supports, are most profitable. The majority of aircraft turbine engines have the separate rotors of compressors and turbines with own bearings, but often with the common central bearing. Such solution makes possible to relief the internal bearing from reactionary unbalancing forces of every separate rotor through the suitable „adjustment" of the mutual angular position of compressor and turbine rotors.

The forces due to the unbalancing depend only from the square of a rotor rotational speed, of course at assumption that the mass centre of rotor does not change its position in relation to its rotation axis. The change of a mass centre position during the engine operation can be caused by following factors: the small transverse stiffness of rotor (and its deformation), the decrease of mass (e.g. to tear the part of rotor blade off), and the clearance increase in the rotor supporting bearing.

The loadings due to the unbalancing of rotors occur as a result of the action of mass forces during the rotor rotation; in case of the gas-dynamic forces depend not only on the mode of engine operation (so indirectly from a rotational speed) but also from the altitude and speed of flight (exactly from the air density in an engine inlet).

The continuous measurement and recording of amplitudes and frequencies of the engine frame vibration in the chosen plane (e.g. perpendicular) makes possible to create the special diagnostic parameter which can be called the „coefficient of flight safety".

2. BALANCING OF ROTORS

It was known from a long time, that in some constructional cases (e.g. bicycle wheels) the static balancing is quite sufficient because of their small circumferential speeds and flat structure. The units rotated at greater speeds and having the considerable „along axis" sizes have also in addition required the dynamic balancing (e.g. wheels of personal cars). The principles of the static and dynamic balancing of rotated blocks were explained in Fig.1. For simplicity, unbalanced masses were imported to the common plane. Hence, it clearly follows that the static balancing can be recognised to be sufficient only for the flat axial symmetrical disks (such as e.g. rotor disks of low pressure turbines).

The multistage assemblies of these turbines and multistage compressor rotors are the axially extended rotated blocks having a lot of the mutually connected disks, drums and shafts with the palisades of rotor blades severally placed on the external rings of disks.

The process of balancing, both static and dynamic, depends on the removal of the mass from the „heavier" places (by milling, polishing or scraping) or the addition of the mass (e.g. in form of suitable screws) in the „lighter" places of rotors (as was explained in Fig. 2).

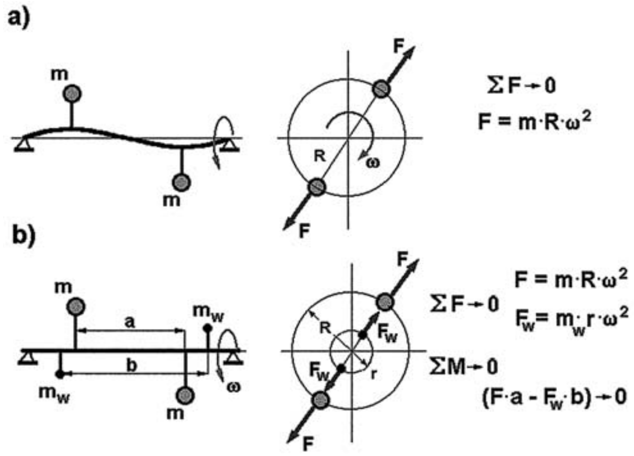

Fig. 1. Balancing principles of axially-symmetrical blocks a) static balancing; b) dynamic balancing

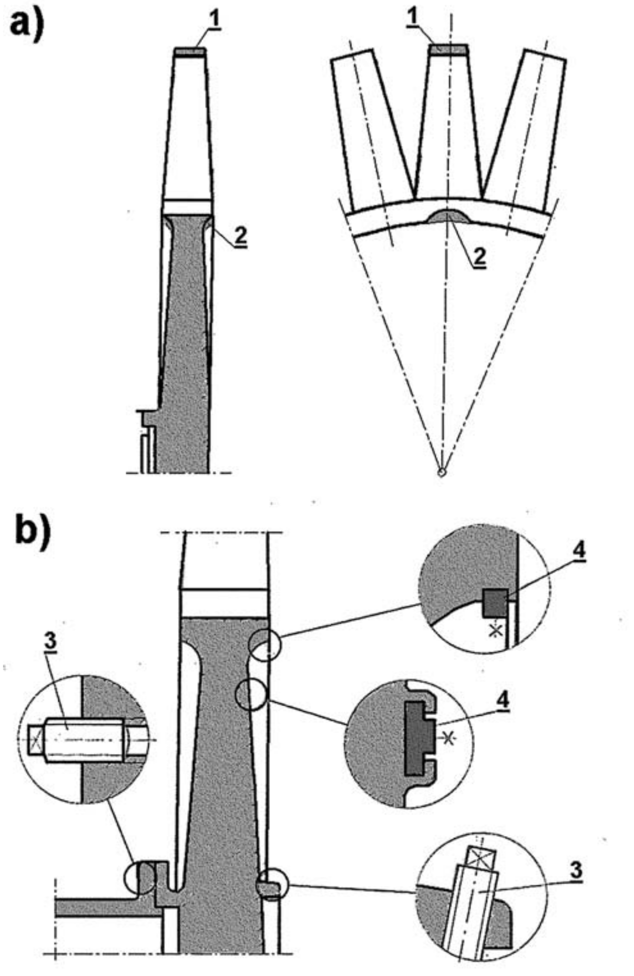

Fig. 2. Methods of rotor balancing: a) removal of mass; b) addition of mass: 1 – shortening of blade top; 2 –removal of material from the side surface of disk; 3 – balancing screw; 4 – balancing mass

The process to join the compressors to turbines in the entire rotor assemblies requires the thorough analysis. The best precision is obtained for the integral constructions with bearings placed in two frame supports. The schematic diagram of such supported assembly was shown in Fig. 3.

The stiff structure guarantees the balancing stability during the operation and it can only be disturbed by the erosive losses (chipping) of rotor blades. On the same figure was also shown the „classic" position of supports of the single-stage ventilators rotors applied in the constructions of huge thrust jet engines. The location one of supports „inside" the ventilator rotor, almost in plane of its centre of mass allows to apply only the static balancing and causes that the critical rotational speed of this rotor theoretically aims to infinity.

In each turbojet engine the compressor rotor is mechanically coupled with turbine rotor (according to the principle of operation). In coupling of these assemblies two methods are applied. The first one depends on the factory adjustment of their common position, e.g. by the special asymmetrical splined coupling. Using this method the Rolls-Royce Dervent and Nene family engine rotors (and their developmental versions) were coupled. In the second method, called as American one, compressor and turbine rotors are coupled in such a way, that vectors of residual unbalancing of compressor and turbine rotors have the opposite directions. This was graphically shown in Fig. 4.

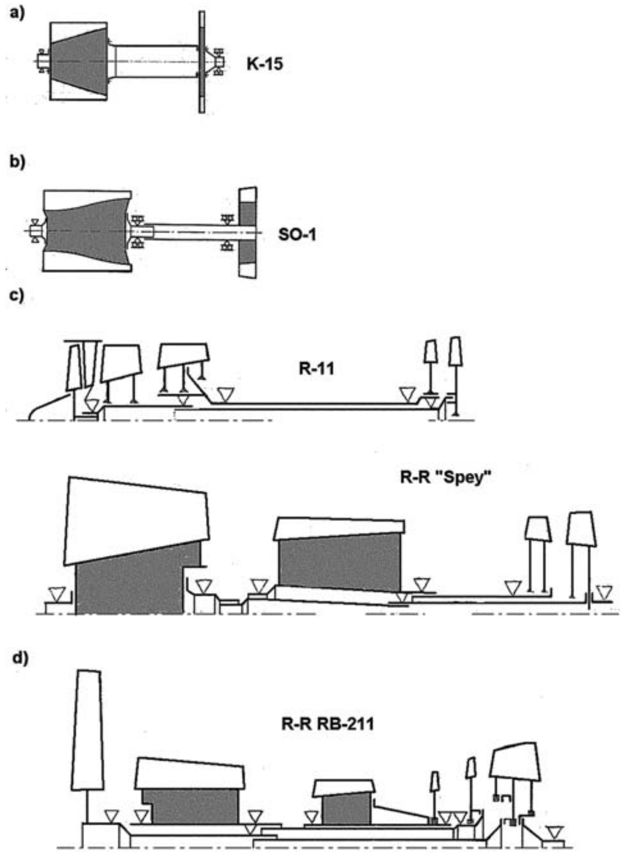

Fig. 3. Basic schematic diagram of rotor bearings: a) two-point support; b) three-point support; c) bearings of two-spool structure; d) bearings of three-spool structure

Thanks to such method of the rotor assembly the unbalancing vector of an entire assembly is the smallest (smaller than each unbalancing vectors of component rotors). However in this case, it is necessary to ensure the possibility to couple the turbine rotors with compressor rotor in the various mutual angular settings. It should also be possible to perform this operation in the engine repairing shops. This method of rotors coupling also allows to compensate the action of unbalancing forces on extreme supports by using the additional balancing mass.

The essential restriction in the rotor balancing process (especially compressors) is the value of necessary power to keep their rotation with the sufficient speed to precise measure the unbalancing forces in the planes of rotor supports. The values of power taken by compressors of present-day jet engines reach the hundreds of kilowatts (on maximum power ranges even many megawatts). Although the necessary external powers to keep turning rotors of compressors, ventilators and turbines during the testing are smaller than in case of the operating engine but these necessary external power is the serious restriction for the present-day balancing machines. From this reasons the border possibilities of the applied balancing machines practically allow to make the measurements on rotational speeds reaching only just a dozen percentage of the nominal rotational speed of rotors. It is possible to get the somewhat higher rotational speeds placing the tested rotors in vacuum chambers or in chambers filled with helium (density of helium is about seven times less than density of air).

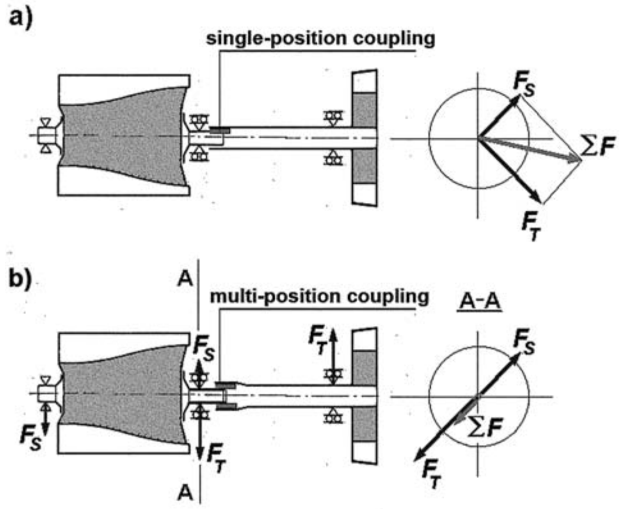

Fig. 4. Effect of assembly method of rotor assemblies on their balancing: a) factory adjustment of mutual angular position of rotors; b) optimum position set in repair shop

The measurement of the unbalance degree value of rotors in the factory or repair shop is the important criterion to estimate the quality of assembly, repairs and manufacturing. Observation of changes of the unbalance degree value during rotor operation makes possible to estimate the current technical state of engine and to predict the safe usage period. The reducing of the unbalanced degree of turbine engine rotors has the significant influences on its structure durability.

2. ENGINE TRANSVERSE VIBRATION

The main source of the transverse vibration of a turbine engine are inertia forces acting on rotors when, due to the unbalancing, the mass centres of rotors do not coincide with their rotation axes.

The general schematic diagram of engine, together with the location of sensors and the measurement of vibration directions, was shown in Fig. 5. It was also presented the drafts of movements of vectors of unbalancing forces (F) as a function of the temporary angular position of rotors of two-spool jet engine and the resultant forces of both rotors. The rotational speeds of both rotors are different so the values of unbalanced forces are different too. In operation conditions, in case of rotors balanced with the similar accuracy - the unbalancing forces of the high pressure rotor (gas producer) are several times greater than the low pressure rotor.

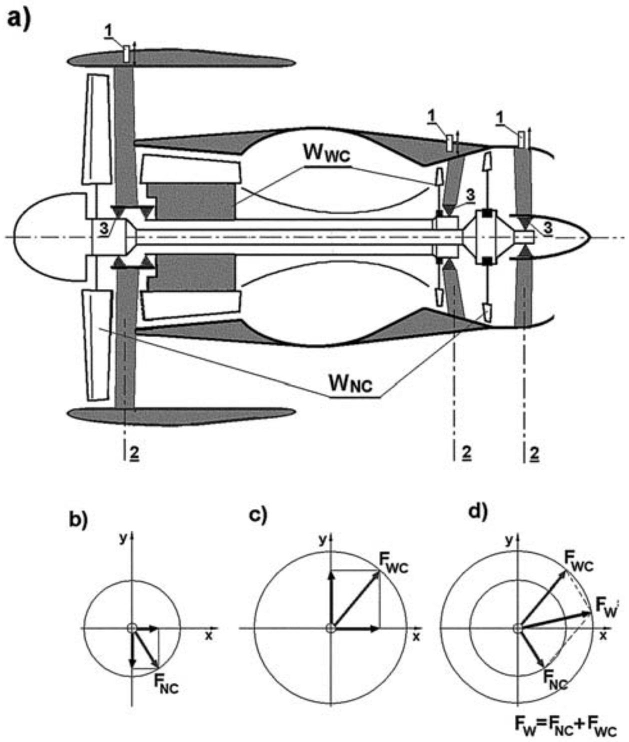

Fig. 5. General schematic diagram of engine and temporary angular position of vectors of unbalancing forces: a) engine; b) rotor of low pressure (WNC); c) rotor of gas producer (WWC); d) resultant force of both rotors: 1 - vibration pick-ups; 2 - plane of supports; 3 - supports

In the two-spool engine (with rotors coupled only gas-dynamically) the function graph of total vertical component of forces, coming from both rotor unbalancing, imposed on the frame of engine is cyclically changing and depends from the temporary slip of rotors. Figure 6 shows: the temporary values of vertical component of inertia forces, coming from the unbalancing, as a function of the angular position (separately for both rotors) and the resultant force as a function of the angular position of gas producer rotor.

The above introduced considerations allow to calculate the value of forces coming from the unbalancing. The experimental verification of these calculations is made through processing of electronically recorded displacements, accelerations or (usually) the speeds of displacement as a function of their occurrence frequency.

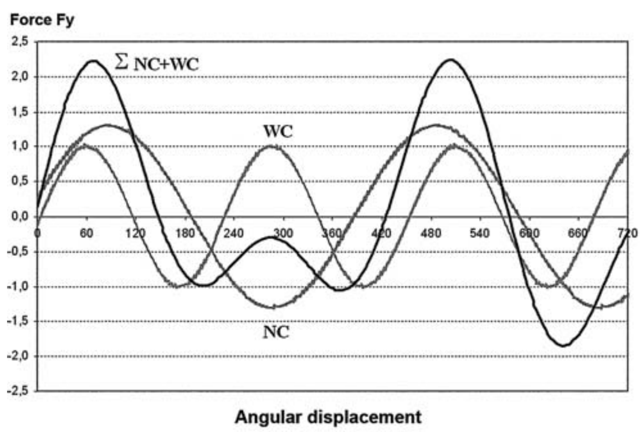

Fig. 6. Changes of vertical components of unbalanced forces Fy vs. angular position of rotors: NC – low pressure; WC – high pressure; NC+WC – resultant forces of both rotors

Figure 7 shows the characteristic dependences of vibration amplitudes (as a function of frequency) due to the unbalancing forces of the bypass two-spool jet engine during operation on the maximum power. The frequency of occurrence of the largest amplitudes (peaks) allows to identify, by comparison of rotational speeds of individual constructional units, which rotor or unit creates these peaks (e.g. the rotor of oil froth breaker or the rolling bearing cage). The predominant amplitudes are generated by rotors which unbalancing is changed during operation. Monitoring of these changes allows, in time, to notice the damages which can cause the break-down of engine e.g. the excessive wear of bearing, or chipping of part of rotor blade. Particularly essential, it is suitably early to detect the appearance of the vibration amplitude having the frequency twice larger than the basic frequency resulting from the rotational speed of rotor ($f = n/60$ [1/s]). This symptom points that the circumferential clearance of a bearing race already reached the value causing the loss of contact point (separation of the rolling elements from a bearing race). As a result, the striking elimination of clearance occurs - twice during one turn. This threatens that the bearing wear and the liquidation of tip clearance of rotor blades (friction between blades and a frame occurs) will be accelerated and the break-down of engine will come shortly.

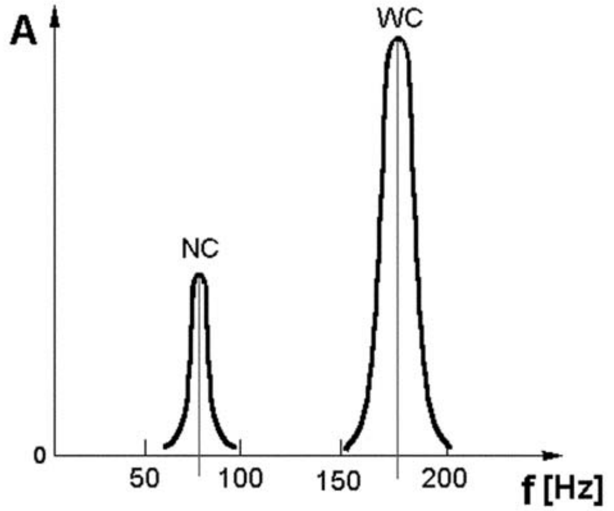

Fig. 7. Amplitude (A) of engine vibration vs. imposed frequency (f): WC – from rotor of high pressure; NC – from rotor of low pressure

In order to practically use the above mentioned considerations to diagnose the specific engine and the entire population of engines, it is necessary to find the effect of the operation conditions, constructional and assembly features on the formation of vibration.

The position of mass centre of rotor placed in the frame bearing has been effected by their radial clearances. In the state of rest of rotor and its small rotational speed they are eliminated in the direction of the action of gravitational forces. When the rotational speed increases, the transverse displacement of mass centre of rotor increases as well to reach the value of its deflection. After crossing the rotational speed, at which the centrifugal force acting radially on the moved mass will cross the gravitation force it will happen the circumferential „reeling" of rotor on external bearing races to eliminate the radial clearance of bearings accordingly to the temporary direction of the resultant vector of centrifugal force. The bearing races can also to be reeled in range of the flexibility of supports in which are placed. Fig. 8 shows how the mentioned factors have an influence on the radial displacement of mass centre of rotor in relation to its rotation axis.

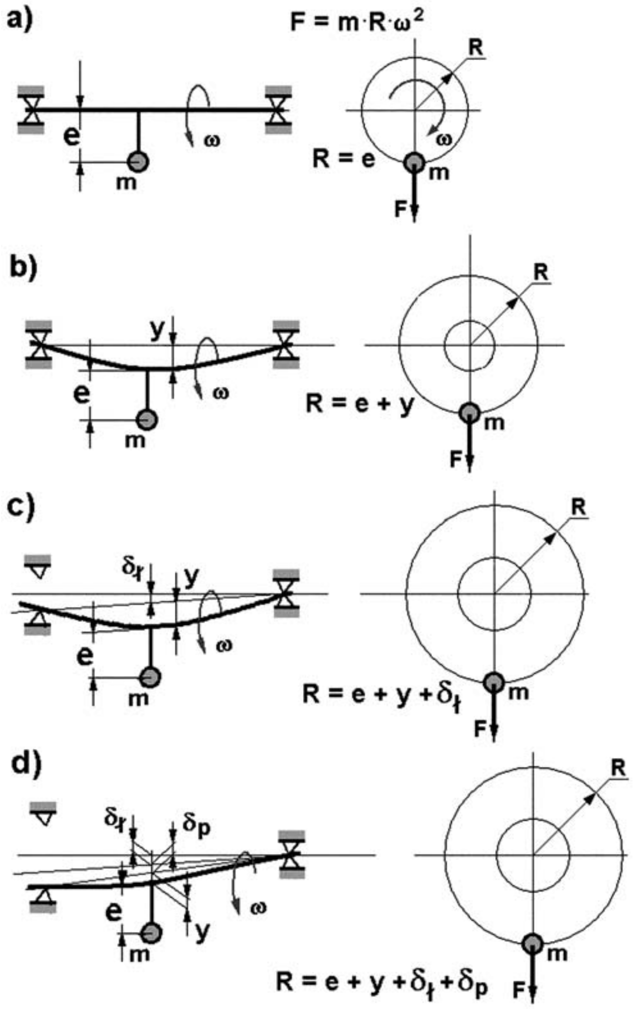

Fig. 8. Movement of mass centre of rotor due to the action of unbalancing forces taking into consideration: a) eccentric of mass centre; b) deformability of rotor; c) radial clearance in one bearing; d) deformability of one support

In all of these cases it was assumed, that one of rotor supports is undistortable and the bearing placed in it has no radial clearance. The radial movement of the mass centre of rotor (m) initially has the value of the eccentric (e) to which in turn are added: the rotor deflection (y), the displacement (δ_l) coming from the bearing clearance, and the displacement (δ_p) due to a susceptibility of support.

The above mentioned description of main factors influencing on the radial movement of a mass centre of rotor in relation to its geometrical axis explains, that the constructional (rotor stiffness and stiffness of bearing frame supports) and assembly (bearing clearance) features have the similar (comparable) influence on the position of mass centre in operation conditions of engine as the residual unbalancing of rotor obtained during its manufacturing and assembly process. Besides the above enumerated factors, the position of mass centre of rotor has also been changed by the thermal deformations of rotor component units, frame supports and bearings (not always the axially-symmetrical ones). These thermal deformations strongly depend from the current loading of engine (thrust or power), the height and speed of flight, and even the climatical zone where the airplane or helicopter are used.

During the engine operation occurs: the bearings wear (abrasion of races and rolling units), the increase of bearing fitting in frame supports and the erosive decreasing of a material in rotor blades. All of these have the influence on the change of position of rotor mass centre and its unbalancing degree. For each specific engine the changes of the unbalancing of rotors depend on the kind of mode of its operation. Nowadays almost universally, it is used the method in which the safe usage period of engine is estimated according to its current technical state. In this method it is necessary to compare the initial values of engine (new or repaired) diagnostic parameters with the current ones. This should also be related to the vibration profile determined on the engine built-up in the airframe.

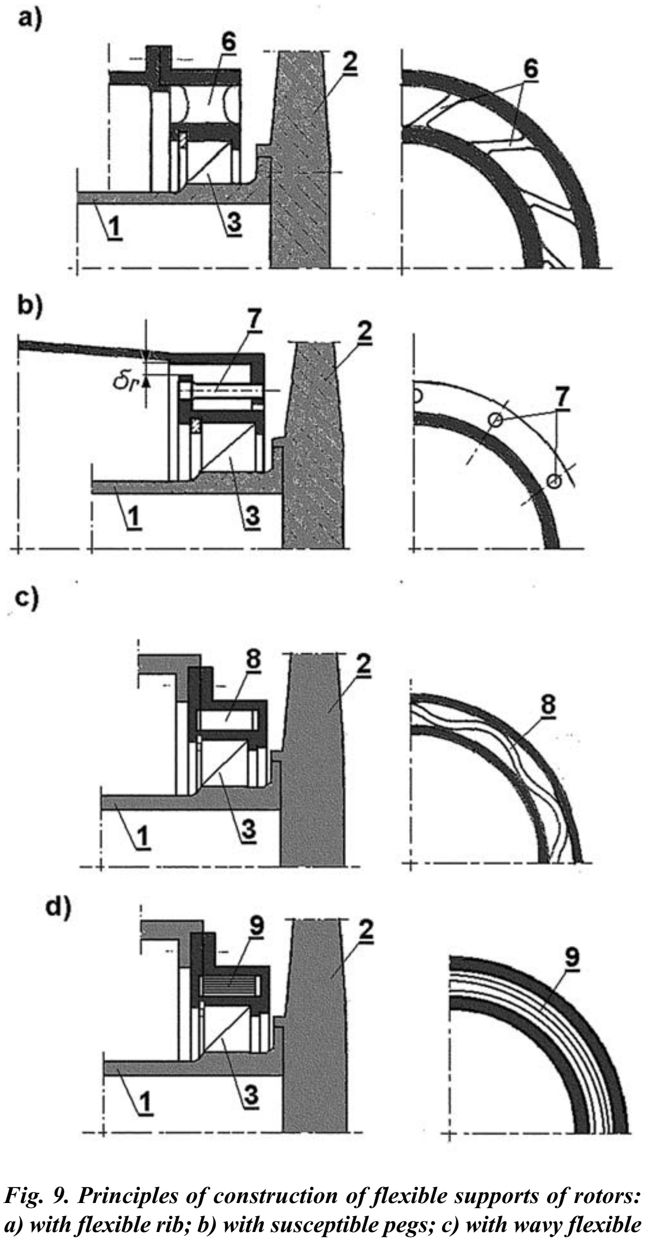

Fig. 9. Principles of construction of flexible supports of rotors: a) with flexible rib; b) with susceptible pegs; c) with wavy flexible insertion d) with packet of tin insertions 1 - shaft; 2 - bearing disc of rotor; 3 - bearing; 6 - flexible rib; 7 - flexible plugs; 8 - wavy flexible insertion; 9 - packet of tin insertion

In order to soften the vibration transferred from rotors through bearings on: the supports, the different units of engine and its suspension in airframe - the flexible supports are universally applied. Accordingly to their light constructions all supports of aircraft engines are flexible in the notable degree, but in this case the flexibility was intentionally introduced. In Fig. 9 was presented the typical constructional methods to get the required flexibility of supports of aircraft engines. The first of them (Fig. 9a) – the oldest one, was applied in the Rolls-Royce Dervent and Nene family engines, the second (Fig. 9b) was also worked out at this firm and was applied among others in the Spey engines. The special attention deserves the solutions shown in Fig. 9c and 9d. In these solutions the vibration was damped by the insertion of oil into the closed spaces between the external race of bearing and its support in a frame. This oil, under the action of changing loading forces of the bearing, squeezes through the crevices in the flexible insertions.

The introduction into the structure of aircraft turbine engines the flexible bearing supports for ventilators, compressors and turbines rotors has considerably changed the dynamic characteristic of rotor assemblies and the engine vibration spectrum transferred on the airframe. The favourable effect, which comes from these, is the quite good reduction of vibration amplitudes transferred on the structure of airframe or even their total removing. This mainly relates to the vibrations created by the residual production unbalancing of rotor. This unbalancing increases in time during the long-lasting operation as a result of the formation of erosive losses of rotor blades and also due to the dusts settled down on them. The flexibility of rotor supports significantly reduces the ranges of critical speeds of rotors but it seldom, if ever, is the desirable feature of engine. The rotors of aircraft engines are usually the subcritical ones. It is thought that operation range of their rotational speeds should be remote from the critical range at least of 20 to 25%. These values come, first of all, from the experience obtained on the following base : the determination of causes of the breakdown of engines, the analysis of parameter changes recorded during the operation conducted according to the technical state of engines and the manufacturing tests conducted at the extreme loading of engines.

3. CHANGES OF AMPLITUDES AND FREQUENCIES OF VIBRATION DURING OPERATION OF ENGINES

In aviation, the use of turbine engines is unusually diversified because their changing loading coming from: the airframe demands on power or thrust, the operation conditions dependent from the climatical zone, a time of the year, the dustiness degree of air and other factors relative to the kind of mission realized by airplane or helicopter. These mentioned factors, first of all, have the influence on the progressive erosive and corrosive wearing of parts located in the fluid-flow channels of engines and the bearings which support the rotors. It comes from the experience that the largest loadings of engine occur in case of the fighter airplane during the air fight. In this case the transverse overloads of rotors reached even 8 to 10g. The repeated overloads can cause the stiffness changes of rotors due to permanent deformations of the screw joints of drums and shafts collars and the changes of bearing clamps placed in supports. As a result, the deflection of rotor and the bearing clearances increase so in effect the progressive growth of the rotor unbalancing degree has been noted.

The accelerated wear of rotor bearings happens particularly in case of engines of helicopters often started from the accidental landing strips. In this case the stirred up dust is sucked by the running engines (here, it should be noted that the turbine engine demands about four times greater of air than the piston engine having approximate the same power). Similarly, the large amount of dusts is sucked by the powerful bypass jet engines installed on the present-day transcontinental passenger airplanes. First of all, it is due to the huge intensity of air flow which reaches from 600 to 800kg/s during take-off and landing when, so called, the thrust reverse is switch on to slow down. From these reasons the rotor bearings of aircraft turbine engines are subjected to the special supervision to escape from the break-down in flight. It is widespread method to control the wear state by the observation of amount of sediment („file dust") on magnetical corks placed in channels draining oil from every bearing. The different method applied in the bearing diagnostics used the addition of the various chemical element into the material of rolling elements (rollers and balls) to permit the identification of wear products coming from the specific bearing (during testing of the oil samples).

In one of the leading American manufacturer the „shrewd" method, to diagnose the particularly heavy loaded and hard to reach rotor bearing, has been worked out. In this method, on the internal bearing race, the flat surface of about 2mm width was made. This flat surface makes up the transverse „groove" for the rolling over rollers (draft of this solution was shown in Fig. 10). When the every roller goes over this groove the acoustic signal (and not only) is generated. The decrease of these signal frequency in relation to the early registered signals of rotors of new engines, of course for the same of rotational speed, indicates the growth of slip of the rollers in bearing races (here in the internal race) due to the increase of the bearing clearance. The further development of phenomenon leads to state shown in Fig. 8c.

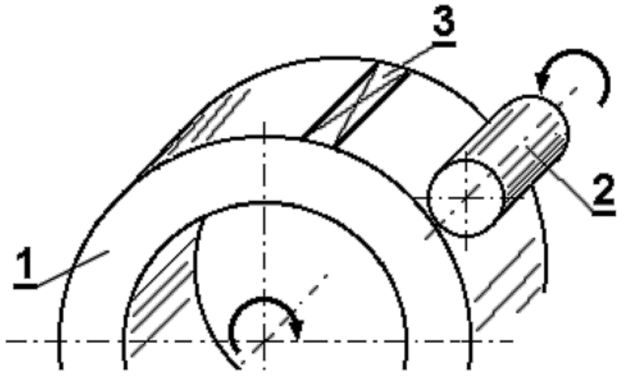

Fig. 10. Inner race of roller bearing prepared for diagnosis: 1 – inner bearing race; 2 – bearing roller; 3 – groove on bearing race

The described cases belong to the group which has the „soft" influence on the wearing process of engines. The continuous observation of enlarging amplitudes of vibration connected with rotational speeds of rotors and the appearance of vibration amplitudes with diversified frequencies indicates the wear degree of the engine. Fig.11 shows the growing amplitudes of vibration (mainly identified as „rotor") recorded in function of the engine operation time.

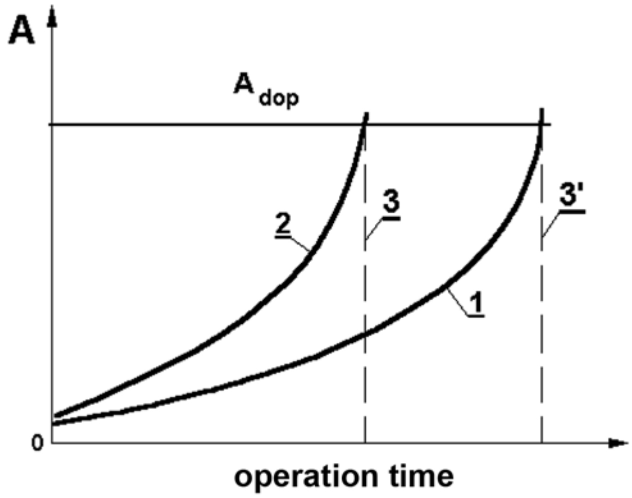

Fig. 11. Observed amplitudes of vibration(A) vs. operation time of engine: 1 – „soft" condition of engine operation; 2 – „hard" condition of engine operation; 3, 3' – ranges of safe operation

In order to easily interpret of such records the hypothetic extreme curves were marked: for the „soft" condition of engine operation (e.g. in case of a transcontinental passenger airplane) and for the „hard" condition of engine operation (e.g. in case of operation in a military aviation training school). The shown differences indicate the necessity to conduct individual observations of the technical state of every engine in order to lengthen its operation time to the borders of its durability keeping in the same time the required margin of safe flying.

In companies using in similar way the considerable number of airplanes of one type (in the similar climatical conditions) – as it occurs for example in case of the big airlines – it is possible to conduct the continuous observation of the current technical state of the entire population of engines. Figure 12 is a sketch of the relationships of dynamic features area of engines as a function of the operation time. The quantitative character of the distribution of its features in population was marked. On both figure (11 and 12) the admitted by manufacturer level of vibration amplitudes and resulting from this the limitation of the safe operation time were marked. Conducted in this way, the continuous monitoring of dynamic features of the entire population of engines can make up the basis to predict the overhauls, replace the modules and pass the engines to be repaired. All of these make possible to achieve the operational availability of the possessed group of airplanes.

Recording and the continuous observation of the amplitudes and frequencies of engine vibration give the possibility to detect in time, for example, the growing wear of bearing, the break of part of rotor blade and due to this it will be possible to prevent the serious break-down of engine.

5.SUMMARY

It is quite obvious that the values of amplitudes and frequencies of engines transverse vibrations are the most important (and the most sensitive) diagnostic parameters, which the changes (the growth of amplitudes and the appearance of new frequencies) indicate the threat of engine break-down. However, they are not only ones, which warn the user about the progressive wear (damage) of rotor assemblies, their bearings and supports in the engine frame.

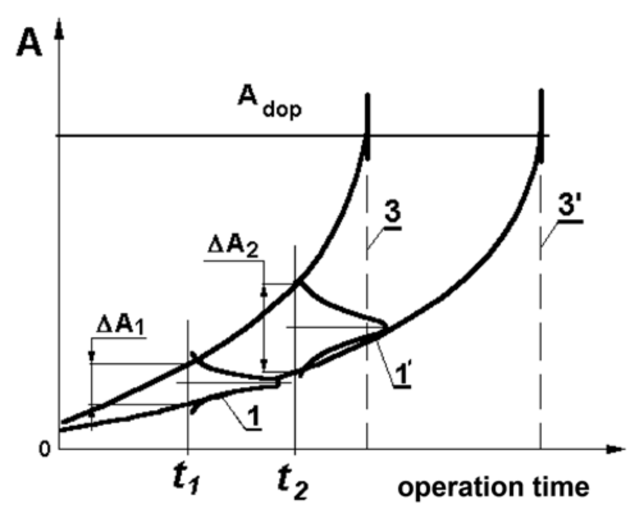

Fig. 12. Maximum amplitudes of vibration of engine population vs. operation time of engine: 1, 1' – quantitative distribution of engines (having dynamic features in range DA1 i DA2) in population after operation time: t1 and t2; 3, 3' – boundary of safe operation

The diagnostic methods are subjected to the continuous development which tempo, first of all, depends on:

– the growing knowledge of scientific workers widened by the PhD and professor theses and the promotion of this knowledge to the next generation of engineers;
– the technical progress in the automatic recording, data acquisition and processing systems using the microprocessor technique,
– the growth of the ability for the close cooperation among expert teams presented the different disciplines such as: physics, chemistry, mechanic, electronic engineering, control system engineering and so on; it is also necessary to set the close co-operation among the research teams, manufacturers and users of engines,
– the popularization of individual and team achievements in the wide accessible scientific literature, popular science periodicals and the widely available internet pages.

REFERENCES

[1] **Balicki W.:** *Wpływ warunków i zakresów pracy oraz cech termodynamiczno-przepływowych turbinowych silników odrzutowych na informację diagnostyczną.* Rozprawa doktorska. Wojskowa Akademia Techniczna 1997 r.
[2] **Den Hartog:** *Tieorija kolebanij.* Wyd. GTTI, Moskwa 1942 r.
[3] **Głowacki P., Łagosz M., Szczeciński S.:** *Drgania silnika, jako wskaźnik diagnostyczny.* Wojskowy Przegląd Techniczny 1989, nr 4.
[4] **Głowacki P., Łagosz M., Szczeciński S.:** *Drgania w dwuwirnikowych silnikach odrzutowych.* Wojskowy Przegląd Techniczny 1989, nr 5.
[5] **Gosiewski Z.:** *Aktywne sterowanie drganiami wirników.* Wyd. Uczelniane WSI w Koszalinie, Koszalin 1989 r.
[6] **Kruschik J.:** *Die Gasturbinen.* Wien 1960 r.
[7] **Kurkiewicz M., Zambrzycki H., Szczeciński S.:** *Przyczyny drgań silników turbinowych.* Prace Instytutu Lotnictwa 1994, nr 137.
[8] **Łagosz M., Nowotarski I., Szczeciński S.:** *Wpływ cech konstrukcyjnych silników odrzutowych na krytyczne prędkości obrotowe.* Prace Instytutu Lotnictwa 1994, nr 137.
[9] **Ponomariov S. i in.:** *Osnovy sovriemiennych mietodov rasciota na procnost' v masinostrojenii.* Wyd. „Oborongiz", Moskwa 1952 r.
[10] **Stodola A.:** *Dampf und Gasturbinen.* Berlin 1924 r.
[11] **Szczeciński S.:** *Lotnicze silniki tłokowe.* Wyd. MON, Warszawa 1969 r.
[12] **Szczeciński S.:** *Studium o luzach wierzchołkowych zespołów wirnikowych lotniczych silników jako parametrze konstrukcyjnym i eksploatacyjnym* – rozprawa habilitacyjna. Dodatek do Biuletynu WAT nr 4/1973 r.
[13] **Szczeciński S.:** *Lotnicze silniki turbinowe* Wyd. MON, Warszawa 1964 r.
[14] **Timoszenko S.:***Tieorija kolebanij v inżeniernom dielie.*Wyd. GTTI, Moskwa 1932 r.

S. Szczeciński, W. Balicki

DRGANIA SILNIKÓW TURBINOWYCH
JAKO KRYTERIUM JAKOŚCI WYTWARZANIA
ZESPOŁÓW WIRNIKOWYCH I ICH BIEŻĄCEGO
STANU TECHNICZNEGO

W artykule poruszono problem drgań lotniczych silników turbinowych ze szczególnym uwzględnieniem stopnia niewyważenia wirników. Zagadnienie to bywa jedynie śladowo poruszane w silnikowej literaturze specjalistycznej. Tymczasem ilość informacji zawarta w danych uzyskanych z permanentnie dokonywanych pomiarów amplitud i częstotliwości drgań poprzecznych silników, zwłaszcza wielowirnikowych, jest nie do przecenienia. Przedstawiono związki tych parametrów z formą konstrukcyjną wirników, jakością ich wykonania i montażu, wpływem zmian luzów w podporach i możliwościami wykrywania stanów zagrożenia bezpiecznego użytkowania silników.

С. Шчециньски, В. Балицки

КОЛЕБАНИЯ ГАЗОТУРБИННЫХ ДВИГАТЕЛЕЙ
КАК КРИТЕРИЙ КАЧЕСТВА ИЗГОТОВЛЕНИЯ
УЗЛОВ РОТОРА И ИХ ТЕКУЩЕГО
ТЕХНИЧЕСКОГО СОСТОЯНИЯ

В статье обсуждается проблема колебаний авиационных газотурбинных двигателей с особым учетом степени дисбаланса роторов. Эта проблема исключительно редко затрачивается в специальной литературе по двигателям. Между тем количество информации содержащейся в данных полученных с непрерывно выполняемых измерений амплитуд и частоты поперечных колебаний двигателей, особенно многороторных, трудно переоценить. Представлена связь этих параметров с конструкторской формой роторов, качеством их исполнения и монтажа, влиянием изменений зазоров в опорах и возможностями обнаружения состояний угрозы безопасной эксплуатации двигателей.