Considerations regarding optimization of low speed balancing of high speed flexible rotors

Eng. Alexandru Tudorache
INCD Turbomotoare COMOTI
Scientific Research Assistant
Iuliu Maniu Blvd., No. 220D, Bucharest, Romania
alexandru.tudorache@comoti.ro

Prof. Dr. Eng. Ion Fuiorea
INCD Turbomotoare COMOTI
Senior Scientific Researcher
Iuliu Maniu Blvd., No. 220D, Bucharest, Romania
ifuiorea@yahoo.com

ABSTRACT

The research conducted revealed some particularities of the rotors manufactured and/or assembled at INCD Turbomotoare COMOTI. Different phenomena were encountered, named, and described, using data recorded while dynamically balancing the rotors, like the phenomenon where a rotor spins with progressively lower amplitude of vibration when increasing the rotation speed (named self-centering phenomenon), to a point where the rotor stabilises, and the phenomenon where the heavy point of a specific correction plane appears to move angularly while balancing at progressively increased speeds (phenomenon usually encountered in the case of flexible rotors). The recorded data was used to create six graphs that give a clear image of a rotors' behaviour on a specific range of speeds. After gathering a lot of experimental data, certain patterns emerged that helped formulate a balancing procedure based on these graphs. The procedure was tested successfully and it noticeably reduced the time required for dynamic balancing with better results while operating in normal conditions, at the designed nominal speed.

KEYWORDS: *dynamic balancing, high speed rotors*

NOMENCLATURE

a	The distance between the left bearing and the left correction plane
b	The distance between the two correction planes
c	The distance between the right correction plane and the right bearing
d	Distance between bearings, a + b + c
gmm	Unit of measurement for unbalance (mass multiplied by the radius where the correction is made)

1 INTRODUCTION

In all the work related to this field it has been concluded that the dynamic balancing of the rotors is a fundamental necessity for the correct operation of any machine and that the residual unbalance of a rotor is one of the major factors leading to incorrect operation or even the destruction of a machine.[1][3]

This has prompted specialists in the field to work continuously to improve both the dynamic balancing process and the balancing machines, which has led to a high degree of competitiveness.

During this project, materials related to the rotor dynamics, dynamic balancing and documents on the operation and use of dynamic balancing machines were studied in order to design and carry out dynamic low-speed balancing experiments of high speed flexible rotors.[1-5] As an introduction to rotor dynamics and dynamic balancing, the two best references are [1] and [2]. They both give a detailed description of the phenomena, math and methodology behind rotor dynamic balancing.

The methodology used in the experimentation and the results are briefly described in this paper.

2. EXPERIMENTATION PLANNING

The experiments were designed to be performed on the main types of rotor configurations most commonly used in COMOTI: centrifugal compressor rotors and bladed rotors, pinions, gears and helical screw compressor rotors.

Main parameters to be varied:

a. **Speed** - we attempt to measure the unbalance of each rotor at various speed intervals to see how important data acquisition resolution is and whether a broader range is acceptable or not.

   Planned measuring intervals are 100 RPM, 200 RPM, 300 RPM, 400 RPM, and 500 RPM, starting from a speed of 500 RPM, to the maximum speed possible of the balancing machine.

   In general, balancing speed is chosen in terms of several aspects:

   - Must be far enough away from critical or harmonic speeds;
   - If the balancing machine has "soft" bearings, it must not be too large so the vibration amplitude will not damage the bearings or the transducers of the machine;
   - Must be high enough so that the self-centering phenomenon (explained in section 3.1) occurs.

   An optimum in choosing the balancing speed can be found by studying the evolution curve of the residual unbalance in relation to the speed resulting from the unbalance measurement (see section 3.3).

b. **Correction Planes**

   - For **long** and **symmetrical** / **asymmetrical** rotor configuration (Figure 1)
   - b > a, c;
   - Will vary a and c, keeping a fixed distance between the bearings, if and as long as each rotor permits;
   - Will vary d (= a + b + c), as long as each rotor permits.

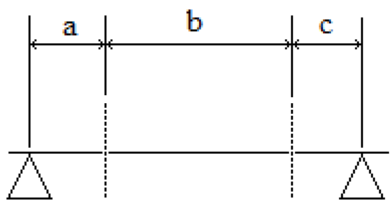

Figure 1 Long symmetrical/asymmetrical rotor configuration schematic

- For **narrow** rotor configuration (Figure 2)
- b < a,c;
- Will vary b, keeping a fixed distance between the bearings, if and as much as each rotor permits;
- Will vary d (a + b + c), if and as long as each rotor permits.

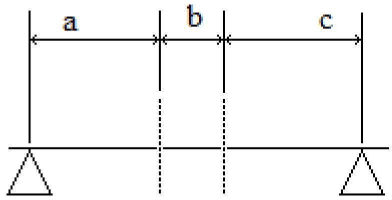

Figure 2 Narrow rotor configuration schematic

- For **console** rotor configuration (Figure 3)
- will vary the position of the entire rotor on the bearings (provided that the centre of gravity is maintained between them in order not to overturn), if and as long as each rotor permits;
- d will be varied if and as much as each rotor permits.

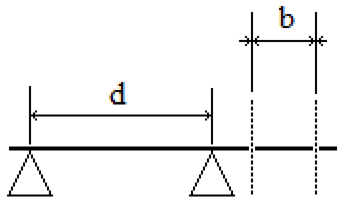

Figure 3 Console rotor configuration schematic

- For **dual console** rotor configuration (Figure 4)
- the position of the entire rotor on the bearings will be varied;
- d will vary, if and as much as each rotor permits.

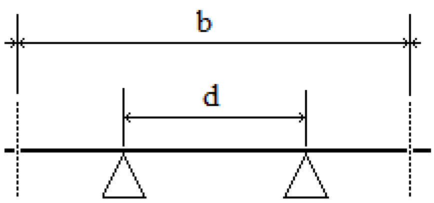

Figure 4 Dual console rotor configuration

c. **Balancing technique**[7][8][9]

- **"left-right" -** involves the elimination of residual imbalance in the left or right plane, either one at a time or simultaneously.
- **"Static-couple" -** means eliminating usually static residual unbalance first, then the couple unbalance, but it is also possible vice versa.

The approach to balancing a rotor depends on its configuration. For example, a helical rotor from a CU90G screw compressor falls into the category of long and symmetrical rotors. In the way it is seated on the balancing machine bearings, the correction planes are far from the opposite bearings, so the interference of the correction planes is minimal, and the best approach to balancing is the "left-right" technique. In the case of balancing the gear in the multiplier of the same compressor, which falls within the narrow rotor category, the two correction planes are very close to each other and a balancing approach using „static-couple" technique is recommended in order to bypass the correction plane interference effect.

Computerized data acquisition was decided to be made in intervals of 100 RPM, with plans to try 50 RPM intervals for even better resolution. We cannot use a shorter speed interval because the accuracy of the speed measurement does not allow this (the sensor measures from 60 RPM to 300 RPM with a +/- 1 RPM error, from 300 RPM to 6000 RPM with a +/- 10 RPM, and from 6000 RPM to 60000 RPM, the maximum speed of the IRD 246 balancing device, with an error of +/- 100 RPM).[3]

Connecting the IRD 246 balancing device to a computer was done with an RS232 communication cable.

Fig. 5 shows the schematic of the existing IRD 246 balancing machine and in the dashed line the added equipment for automated data acquisition.

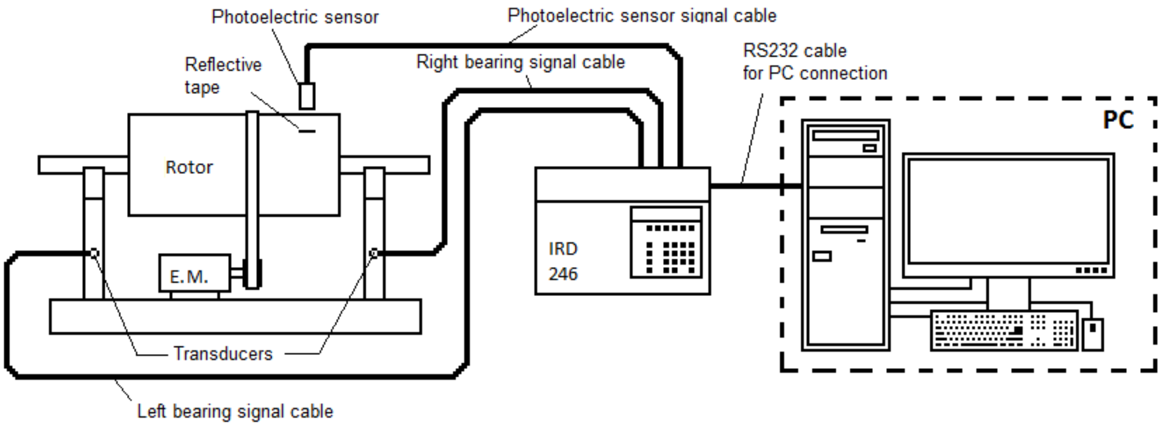

Figure 5 IRD balancing machine with IRD246 balancing device schematic

3. DATA ANALYSIS

Experimental data has been accumulated to identify common elements that appear in rotor behaviour (depending on their type) have been analysed and some phenomena that have occurred frequently in each type of investigated rotor have been identified, which helped taking decisions to balance the rotors.

With these data, graphs have been developed that show the rotor behaviour over the entire speed range at which it can be tested. Thus, these graphs provide a fairly clear picture of the phenomena taking place and ease the decision-making process on the rotor balancing approach.

3.1. THE SELF-CENTERING PHENOMENON

We call "self-centering phenomenon" the phenomenon that occurs when gradually raising the speed from a very low one. From the graphs (Fig. 6), where we represent the residual unbalance and the vibration amplitude measured according to the speed at which the measurement is made, it is observed how the effect of the vibration decreases, the vibration amplitude of the rotor becoming less and less.

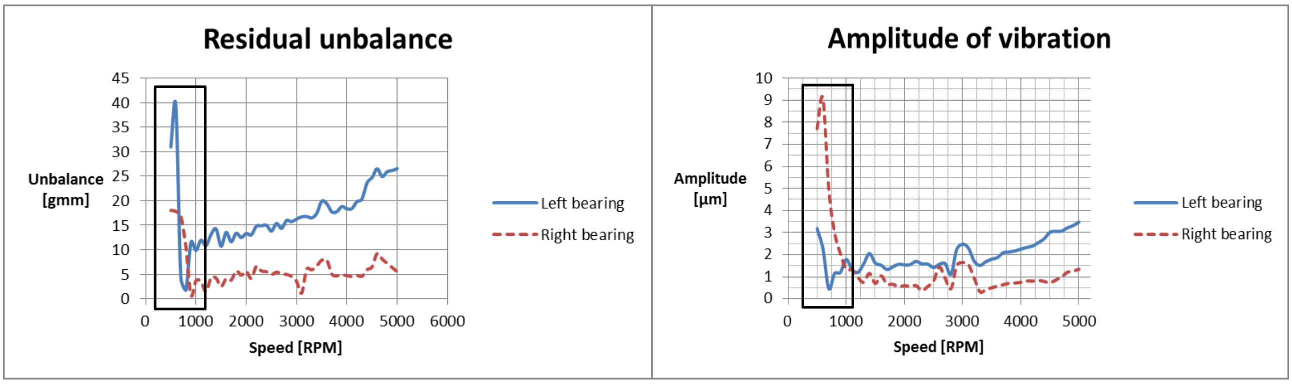

Figure 6 Example for self-centering phenomenon

3.2. HEAVY POINT „MIGRATION" PHENOMENON

The heavy point is the place where the mass of material is causing the imbalance.

At each data recording made by the balancing machine, it specifies the angular position of the heavy point in the correction plane chosen, as well as the amount of mass to be removed. In the graphs (Fig. 7), we represented the angular position of the heavy point in polar coordinates, the radius representing the residual imbalance value (left), and the angular position relative to the speed (right).

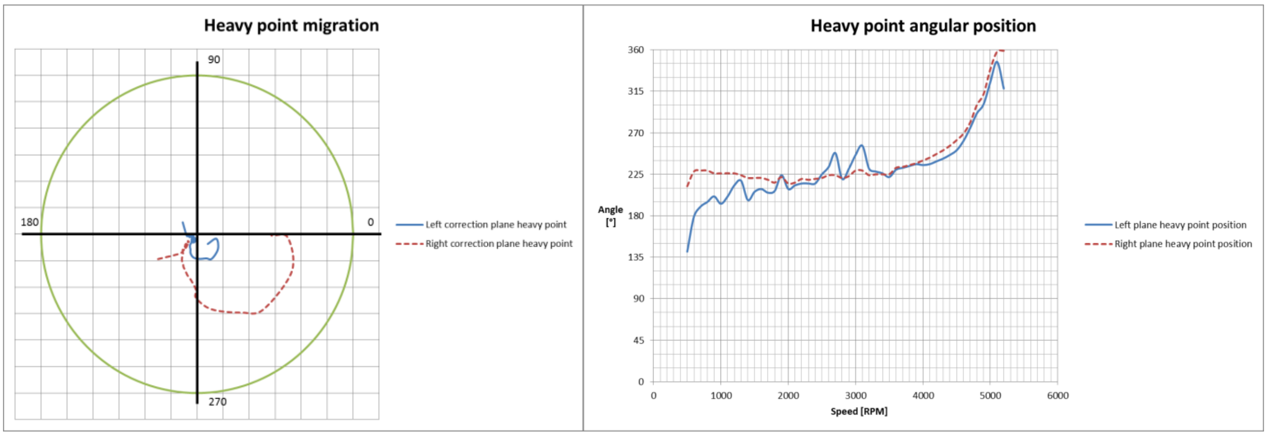

Figure 7 Example for the heavy point migration phenomena

It is noted that, most of the time, with increasing speed, the angular position of the heavy point changes. We call this change of position "heavy point migration".

3.3. SELECTION OF BALANCE SPEED

In the balancing standard, a certain value or a way of choosing the balancing speed is not specified. Several sets of recommendations are defined depending on the rotor configuration, the number of components, or whether it is an assembly, in which case the individual balancing of each component is recommended. The balancing speed must be large enough for the self-centering phenomenon to manifest itself completely, but below the speed at which the heavy point begins to migrate strongly (the two above-mentioned phenomena).

For example, the measured values for a test rotor assembly (Figures 8a / 8b).

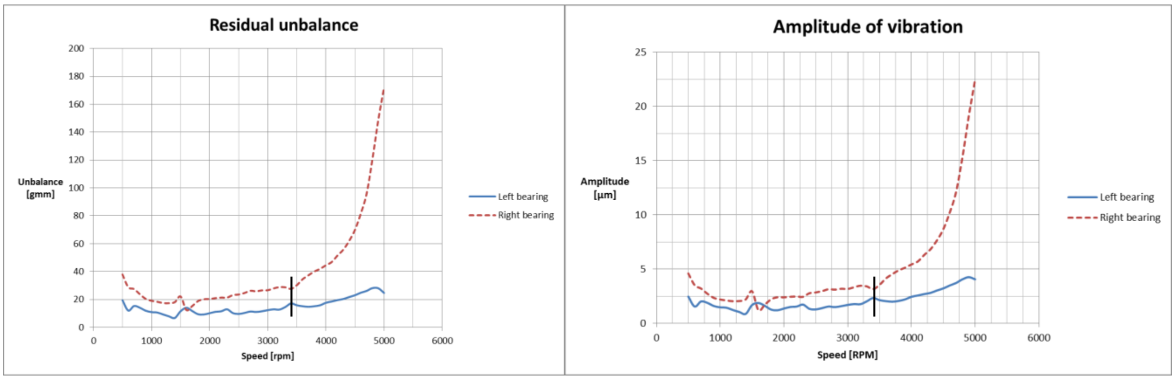

Figure 8a Example for choosing the balancing speed, test rotor assembly

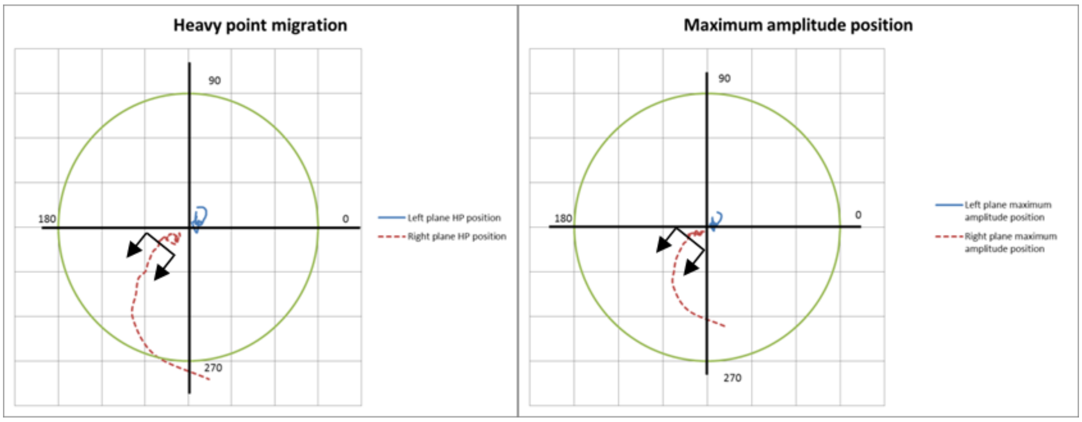

Figure 8b Example for choosing the balancing speed, test rotor assembly

In this case, we notice that we find the two above-mentioned phenomena, so we have to choose the speed taking these into account. Thus, the balancing speed must be large enough for the self-centering phenomena to manifest completely (at approximately 1000 RPM), but below the speed at which the heavy point begins to migrate strongly.

After analysing the values obtained, it was observed that from the speed of 3300-3400 RPM the heavy point strongly changes its position as the speed increases, and the amplitude and the unbalance grow directly proportional.

If we choose the balancing speed above this value, the mass the balancing machine indicates will be larger than the real one, because although the imbalance increases, the mass does not change, but only the effect of the mass on the flexible rotor, which introduces an error in the machine. It is not completely wrong to balance at a higher speed, but it should be taken into account that the mass needed to be removed is much smaller than indicated, the amplitude values must be measured again after the removal, and the process should be repeated several times as necessary, especially if too much mass is removed and the values indicated are opposite to the previous ones. Experience has shown that it saves time and you intervene as little as possible if this is true: the balancing speed is above the speed of the full manifestation of the self-centering phenomena and below the speed of migration of the heavy point.

3.4. LEFT-RIGHT OR STATIC-COUPLE

Also from the graphs previously elaborated, one can see whether a certain type of unbalance is dominant. For example, in the first graphs (Figure 9) it is observed that the two imbalances remain approximately on the same side through the entire speed range after the self-centering phenomenon manifests. Thus, it can be concluded that there is a predominantly static unbalance and that it is a priority to be eliminated first.

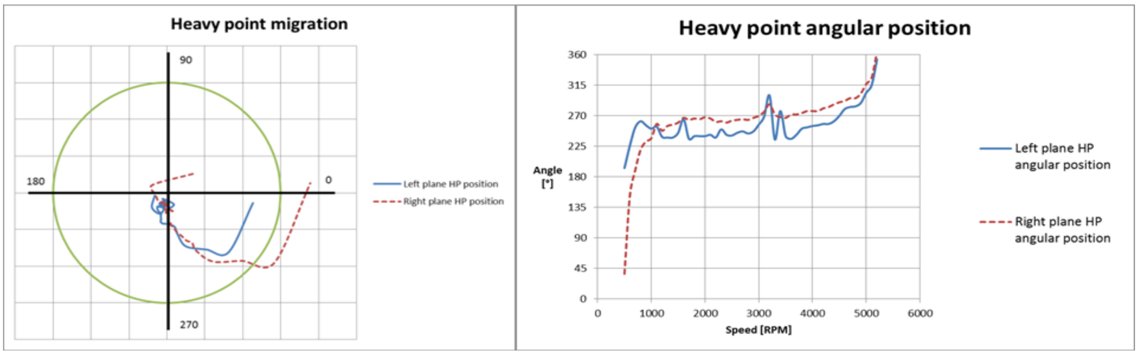

Figure 9 Example of static unbalance

In this other example (Figure 10), it is observed that the two unbalances remain on opposite sides and we can conclude that the predominant is the couple type of unbalance, which is a priority to be eliminated first.

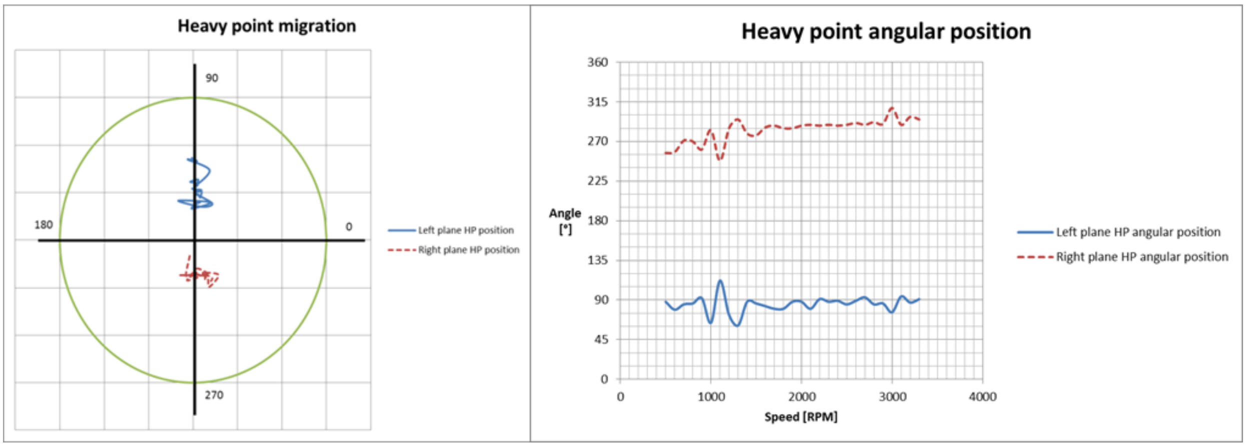

Figure 10 Example of couple unbalance

4.2.5. OTHER OBSERVATIONS

During dynamic balancing, it has been found that in the case of very long and thin rotors, the drive belt may introduce errors in measurements if an arc is formed by tightening it. The best example for this situation is a coupling shaft (Figure 11), about 380 mm long, with a diameter of 24 mm.

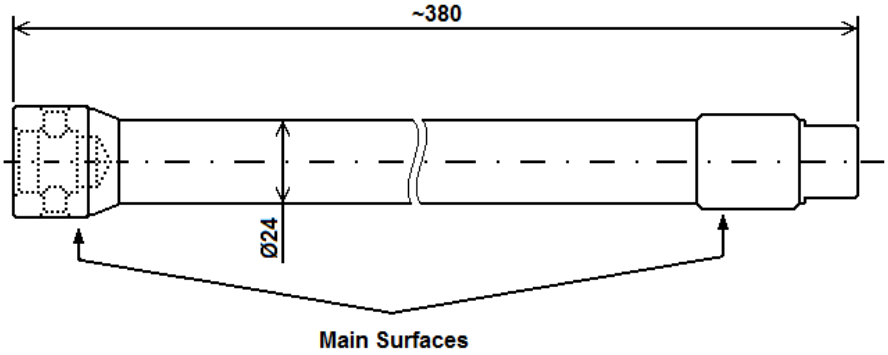

Figure 11 Coupling shaft schematic

Another notable case is that of a jet micro-engine rotor, equipped with a centrifugal compressor stage and an axial flow turbine stage, with a total weight of 1100 grams (Figure 12). The rotor is of small size, too short to be placed on the cylindrical portion of the shaft, 97 mm long, and initially we tried placing it on the inner ring of the bearing near the compressor (left) and on the conical surface near the turbine (right), but the amplitudes were much higher than in the final configuration (and the correct one): setting on both inner bearing rings, clearly indicating that the laying surface is very important, even though the surface is of high quality and within tolerances, small shape deviations significantly affect the measured values, especially for very small rotors.

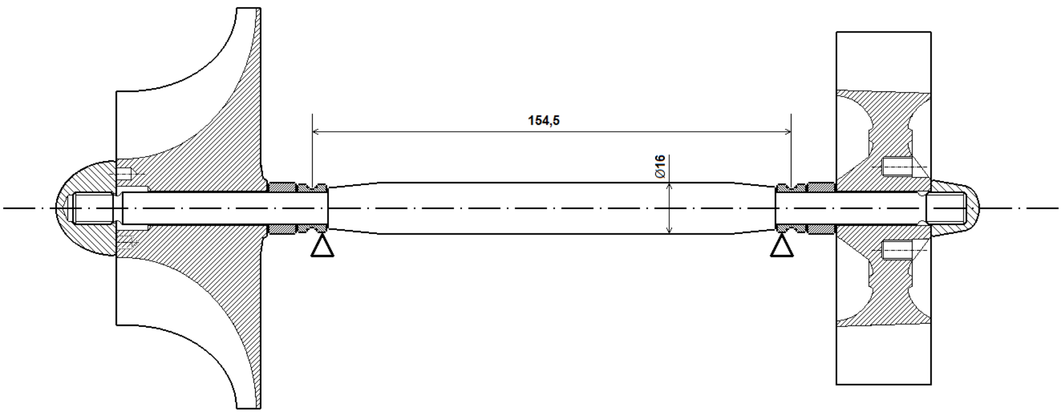

Figure 12 Jet micro-engine rotor schematic

5. CONCLUSIONS

All these observations and experiments were used to create a dynamic balancing procedure, with recommendations and indications, for balancing high speed flexible rotors, but it can be easily applied to all types of rotors, with the purpose of reducing the time required for balancing without sacrificing the quality or reaching higher qualities of balancing in a shorter time than usual.

All rotors balanced using this procedure were monitored while in use for their intended purpose and the feedback received indicated lower levels of machinery vibration than the rotors balanced with normal procedures.

This procedure has been validated for all types of rotors used in INCD Turbomotoare COMOTI, having better results than the previous methods, and will remain as a standard to be used for all future dynamic balancing of any rotor.

REFERENCES

1. M. Radeș, “Dinamica mașinilor”, Ed. Printech 2008;
2. E. J. Gunter, Ph.D., ASME, “Introduction To Rotor Dynamics - Critical Speed and Unbalance Response Analysis”, RODYN Vibration Analysis, Inc. 1932 Arlington Boulevard, Suite 223 Charlottesville, VA 22903-1560, October 2001.
3. “Static and Dynamic Balancing of Rigid Rotors”, Macdara MacCamhaoil, Brüel & Kjær;
4. “Dynamic Balancing Handbook Form #2049”, IRD Mechanalysis Inc., Oct 1990;
5. “Dynamic balancing of rotors”, Dr. Rajiv Tiwari, Department of Mechanical Engineering, Indian Institute of Technology, Guwahati 781039;
6. “Control of surge in centrifugal compressors by active magnetic bearings – Theory and implementation”; Chapter 2 – Introduction to rotor dynamics; Yoon, S.Y.; Lin, Z.; Allaire, P.E.; Springer 2013;
7. “Ref. doc. MI 104 - NOTA TEHNICA – Consideratii privind echilibrarea rotoarelor rigide”, Mobil Industrial AG, 2009;
8. “Ref. doc. MI 105 - NOTA TEHNICA – Echilibrarea dinamica a rotoarelor flexibile”, Mobil Industrial AG, 2009;
9. Standards: ISO 1940-1:2003, ISO 11342:1998 and ISO 19499:1998;
10. Documentation for dynamic balancing machines IRD 246 and IRD 290.

