DYNAMIC BALANCING OF ROTORS WITH MANUAL BALANCERS

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Abstract

An unbalance is the most common issue affecting the turbomachinery rotors. A machine operation at high vibration level due to the rotor disk unbalance will result in the damage to rotor bearings. The vibration level in industrial turbomachinery is hardly ever monitored in online systems to record all exceeded parameters. It is usually monitored in the key process machines. The operation of a supervision system will result in their deactivation at reaching the alarm threshold vibration level. The rotor disk is in almost all cases balanced in its own bearings supporting the shaft using the influence coefficient method. In some cases, the rotor disk is balanced in normal operation using automatic balancers. Although they do not solve the rotor unbalance issue, they reduce the vibration level at any given time. The main disadvantage of the system is its high price, often exceeding the costs of a new pump or fan. If technical conditions allow to stop the rotor for a short period, the disk can be balanced using manual balancers, with a significantly lower cost compared to the automatic balancing equipment. The operation of the manual balancing system is described herein.

Keywords: field balancing, manual and automatical balancer, influence coefficient method, modal balancing.

DYNAMICZNE WYWAŻANIE WIRNIKÓW Z WYKORZYSTANIEM MANUALNYCH ELIMINATORÓW NIEWYWAŻENIA

Streszczenie

Niewyważenie jest najczęściej spotykaną imperfekcją wirnika maszyny przepływowej. Praca maszyny przy zwiększonym poziomie wibracji będącym następstwem niewyważenia tarczy powoduje uszkodzenie łożysk wirnika. Poziom drgań przemysłowych maszyn przepływowych rzadko kiedy jest monitorowany w systemie on-line, odnotowującym przekroczenie wartości parametrów uznanych za dopuszczalne. Tak bywa jedynie w przypadkach maszyn kluczowych dla procesu technologicznego. Działanie systemu nadzoru powoduje ich wyłączenie w przypadku osiągnięcia poziomu wibracji określonej progiem alarmowym.

Tarcza wirnika jest wyważana w łożyskach podpierających wał prawie zawsze metodą macierzy współczynników wpływu. Tylko w nielicznych przypadkach wyważanie tarczy wirnika odbywa się podczas normalnej eksploatacji z wykorzystaniem automatycznych eliminatorów niewyważenia. Choć nie rozwiązują one problemu niewyważenia wirnika, pozwalają w ograniczonym zakresie na zmniejszenie poziomu jego drgań w dowolnej chwili. Główną wadą tego systemu jest wysoka cena, przewyższająca niejednokrotnie koszt zakupu pompy czy wentylatora. Jeżeli warunki technologiczne pozwalają na zatrzymanie wirnika na krótki czas, można jego tarczę wyważyć przy wykorzystaniu eliminatorów manualnych, których koszt jest znikomy w porównaniu do ceny urządzeń wyważających wirnik w sposób automatyczny. Działanie manualnego systemu wyważania omówiono w niniejszej pracy.

Słowa kluczowe: wyważanie w łożyskach własnych, urządzenie do ręcznego i automatycznego wyważania, metoda współczynników wpływu, wyważanie modalne.

1. INTRODUCTION

A source of the turbomachinery vibrations is in most cases a rotor unbalance due to assembly errors and irregular wear of rotor disk vanes.

The effects of those imperfections on the dynamic state of the rotor can be limited by balancing. The balancing method used depends on rotor design and speed. Generally, the rigid rotors can be balanced by the influence coefficient method, which involves the analysis of rotor response to a set constraint, whereas the flexible rotors can be balanced by modal method, considering for the nature of vibrations corresponding to a consecutive natural frequency. The detailed balancing method classification is included in [1].

A machine rotor balancing process usually requires production standstill. To reduce the standstill, the use of stationary balancing machines that require rotor dismantling is abandoned in favour of rotor balancing in its own bearings. The process, although less time consuming since the rotor is not dismantled, also can not be carried out in normal machine operation. The process is carried out using manual or automatic balancers.

An active vibration elimination process consists in generating forces acting on the rotor to compensate the dynamic loads due to unbalance. An active vibration eliminator mounted on the rotor shaft should provide automatic control. The sensors on the bearing housings continuously measure relevant vibration parameters.

The signals are transmitted to an analysing unit, which determines vectors of forces compensating the dynamic bearing response. The results are used by the control elements to apply the constraints to the rotor.

Since Van de Vegte has presented a concept of automatic rotor balancing [2-4], several more or less rational designs of the control mechanisms, allowing at least theoretical implementation of the process [5-10] were developed. The tests on automatic balancing were performed by Thearle [11, 12], Alexander [13], Cade [14] and Lee [15].

An automatic rotor balancer and an algorithm of its operation are currently researched at the AGH University of Science and Technology, Faculty of Mechanical Engineering and Robotics. A solution offered by Uhl and Mańka [16] can be an alternative to the balancing systems by Lord - a market leader in balancing solutions.

Although there are many significant differences between the solutions, an active vibration eliminator usually consists of three basic components: vibration transducer, controller and control system. A mechanical section of the control system includes some kind of rotary elements, usually a set of two disks, allowing a change in the resulting unbalance. Other solutions are available [17, 18], although they are facing several design issues.

The costs of the active balancer are high and often exceed the costs of a new fan or pump. If continuous machine operation is not required, and we only want to improve the rotor balancing process, much simpler devices can be used, i.e. manual balancers.

It is incomprehensible why the fan and pump manufacturers do not realize the need to equip their products with components to move the balancing process outside the rotor disk. It eliminates the arduousness of contact with the dust laden and hot gases flowing through the spiral fan chamber when attaching the test and correction weights to the rotor disk.

2. AUTOMATIC BALANCERS

There are various concepts of automatic balancer operation available which are in constant development. A solution based on two identical and asymmetrical disks, whose rotation results in unbalance at controlled extent and orientation are used to compensate the unbalance of large industrial rotors. The number of balancers mounted on the rotor usually equals the number of its correction planes. An online balancing system includes sensors, controller and balancing regulator. The balancers allow automatic compensation of the existing rotor dynamic state in continuous mode without the need to stop the rotor.

The automatic balancing process has the following stages:

- determining machine performance by determining vibration amplitude at frequency equal to the rotor rotation frequency,
- determining position of the correction weights using a predetermined method,
- adjusting disk position from its initial to corrected position.

Fig. 1 shows the balancer mounted on the fan rotor shaft fan by Lord.

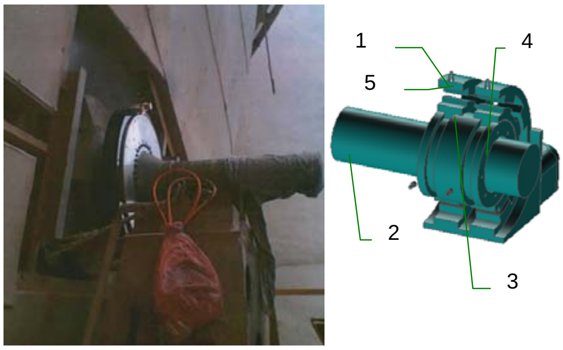

Fig.1. View and diagram of the active balancer: 1. external rings, 2. rotor shaft, 3. balancing disks, 4. clamp rings, 5. vibration sensors and phase markers

The main issue with a balancer design is to transmit the drive to the rotating system and maintain a specific position of the disks. The drive and precision disk positioning system in Lord balancers uses slip-ring motors, whereas the disk position is maintained with the electromagnetic clamps and springs pressing the disks to the rings mounted on the shaft.

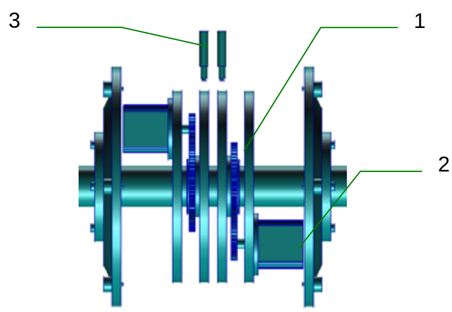

Fig.2. Example balancing disk drive design. 1. mechanisms for changing disk position in motion, 2. stepper motor, 3. transducer for absolute vibration and phase angle measurements

Fig. 2 shows an alternative method to change a relative balancer disk position. The design of a disk drive mechanism is based on [19]. In this solution, the expensive slip-ring motors were replaced with the stepper motors supplied via a rotary joint. The balancing disks are maintained in a specific position by a self-locking gear and a stepper motor's holding torque.

An algorithm to determine the unbalance vector is based on the influence coefficient matrix, which is determined immediately prior to balancing. If the rotor rotates with identical speed and its weight and support rigidity does not change (as in power turbines), the rotor can be balanced using the influence coefficient matrix, determined immediately after installing the balancers, to significantly reduce the balancing time.

The influence coefficient method can be used for balancing with automatic balancers to enable change in the correction plane from the unbalanced rotor disk to the balancer's disk plane. The method used in balancing both rigid and flexible rotors is currently being improved with the optimized algorithms [20]. Other algorithms, including an algorithm based on a dynamic stiffness of the rotor [21, 22] are also used in automatic rotor balancing, and are of particular interest.

3. PRINCIPLE OF MANUAL ROTOR BALANCER OPERATION

The design of a manual rotor balancer is very simple. Unlike active balancers which do not require an operator, a manual balancer requires manual adjustment of the disk position. An assumption that the system rigidity is constant in time and the influence coefficient matrix depends on rotor speed only, in most cases introduces a significant error when determining the correction vectors. Thus it is recommended to determine the influence coefficient matrix each time prior to balancing.

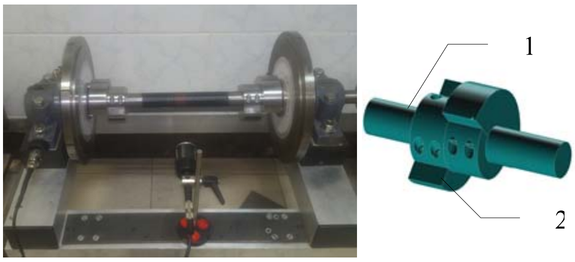

Fig.3. Manual balancer method of mounting on the shaft and diagram: 1. shaft, 2. balancing disks

The balancer mounting on the rotor shaft provides easy access to disks and easy adjustment of their relative position. Choosing the rotor disk as a correction plane is not convenient due to its significant distance from the hatch plane in large fans. The access hatche's dimensions in small fans make it difficult to attach the correction weights to the rotor disk.

4. DETERMINING INFLUENCE COEFFICIENT MATRIX

A following relation between an unbalance vector $\mathbf{F}_{nk}$ and a system response vector $\mathbf{N}_{nk}$ can be observed

$$\mathbf{N}_{nk} = \mathbf{A}\mathbf{F}_{nk} \tag{1}$$

where: **A** is an influence coefficient matrix. An unbalance is the effect of all forces acting on the rotor in relation to a measurement plane (Fig. 4), i.e. both the effects of forces due to asymmetrical rotor weight distribution in relation to its axis of rotation and the constraints due to instantaneous balancer disk position.

$$\mathbf{F}_{nk} = \mathbf{F}_n + \mathbf{F}_k \tag{2}$$

Błąd! Nie można odnaleźć źródła odsyłacza.

Fig.4. Initial position of the balancer disks

Here

$$\mathbf{F}_k = \mathbf{F}_k^1 + \mathbf{F}_k^2 \tag{3}$$

For two balancers mounted on the rotor shaft

$$\mathbf{F}_{nk} = \begin{bmatrix} \mathbf{F}_{nk}^1 \\ \mathbf{F}_{nk}^2 \end{bmatrix}, \quad \mathbf{N}_{nk} = \begin{bmatrix} \mathbf{N}_{nk}^1 \\ \mathbf{N}_{nk}^2 \end{bmatrix} \tag{4}$$

The equation (1) does not allow to determine the correction vector, since the influence coefficient matrix **A** is not known.

If one or two disks are rotated in a way that the unbalance vector position of one of the balancers changes, the vector is $\mathbf{F}'_k$. The system response vector, determined baseing on vibration parameter measurements for both bearings will also change. See Fig. 5.

Błąd! Nie można odnaleźć źródła odsyłacza.

Fig.5. Position of the rotated balancer disks

$$\mathbf{F}'_k = \begin{bmatrix} {}^1\mathbf{F}'_k \\ {}^2\mathbf{F}_k \end{bmatrix}, \quad \mathbf{N}'_{nk} = \begin{bmatrix} {}^1\mathbf{N}'_{nk} \\ {}^2\mathbf{N}'_{nk} \end{bmatrix} \tag{5}$$

The same procedure applies to second balancer disks

$$\mathbf{F}''_k = \begin{bmatrix} {}^1\mathbf{F}_k \\ {}^2\mathbf{F}'_k \end{bmatrix}, \quad \mathbf{N}''_{nk} = \begin{bmatrix} {}^1\mathbf{N}''_{nk} \\ {}^2\mathbf{N}''_{nk} \end{bmatrix}$$

Considering the fact that the asymmetry of the rotor support rigidity may affects its response to the constraints due to unbalance, it is recommended to determine the disk position from the condition of vibration amplitude optimization in horizontal and vertical direction.

The matrix equation relating the system response with specific unbalance and balancer's unbalance is

$$\begin{bmatrix} N_{11}^{0} \\ N_{12}^{0} \\ N_{21}^{0} \\ N_{22}^{0} \\ \ldots\ldots \\ N_{11}^{1} \\ N_{12}^{1} \\ N_{21}^{1} \\ N_{22}^{1} \\ \ldots\ldots \\ N_{11}^{2} \\ N_{12}^{2} \\ N_{21}^{2} \\ N_{22}^{2} \end{bmatrix} = \begin{bmatrix} a_{111} & a_{112} \\ a_{121} & a_{122} \\ a_{211} & a_{212} \\ a_{221} & a_{222} \\ \ldots\ldots & \ldots\ldots \\ a_{111} & a_{112} \\ a_{121} & a_{122} \\ a_{211} & a_{212} \\ a_{221} & a_{222} \\ \ldots\ldots & \ldots\ldots \\ a_{111} & a_{112} \\ a_{121} & a_{122} \\ a_{211} & a_{212} \\ a_{221} & a_{222} \end{bmatrix} \begin{bmatrix} F_{1}^{0} \\ F_{2}^{0} \\ \ldots\ldots \\ F_{1}^{1} \\ F_{2}^{1} \\ \ldots\ldots \\ F_{1}^{2} \\ F_{2}^{2} \end{bmatrix} \tag{5}$$

The influence coefficient matrix indices can be determined from the equation (5).

5. NUMERICAL SIMULATION

High performance industrial fans are usually supported by the spring vibration dampers. The disk diameters are up to three metres. Although the speed of large rotors rarely exceeds 1000 rpm, the disk unbalance is associated with the bearing vibration rates up to several mm·s^{-1}.

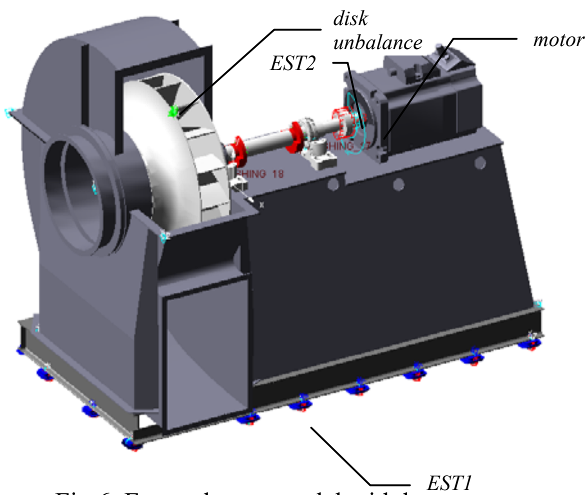

Fig.6. Fan and rotor model with balancers: EST1,EST2 - stiffness-damping elements

The disk must be balanced immediately after the unbalance is detected by the system monitoring the rotor vibration parameters.

Fig. 6 shows the method of manual balancer's mounting on the centrifugal fan rotor shaft. Two separate disk pairs are mounted near the rotor bearings. The location is determined by the intent to limit the vibration level of the bearings as the rotor components most prone to damage.

The simulations of balancer operation were performed on the rotor model shown in Fig. 6. It is a supported rotor with bearings on the same side of a relatively narrow disk. The disk weight at *2230 mm* diameters is *2400 kg*. It was assumed that the two pairs of the balancer disk are in the distance of 800 mm from each other. The distance of the first balancer from the rotor disk face is *1130 mm*.

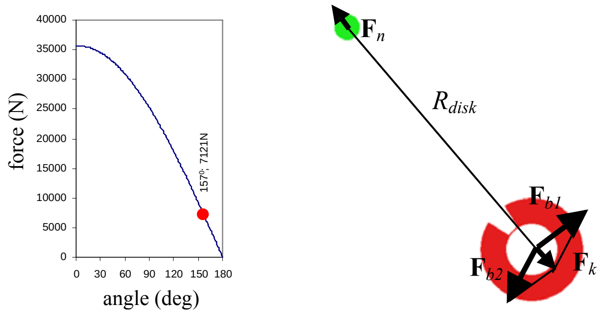

Fig.7. Relation between the force acting on the rotor and the relative angle of balancer disk rotation

The balancer simulations were performed in MSC.ADAMS environment using multi-body dynamics and motion analysis methods. All rotor parts, except for its support and coupling were modelled as rigid bodies, whereas the vibration dampers were modelled as flexible bodies: $2000\ N \cdot mm^{-1}$. Coupling rigidity: flexural-$10^5\ N \cdot mm^{-1}$, torsional $10^8\ N \cdot mm \cdot rad^{-1}$. Rotor unbalance was $6.67\ N \cdot m$.

If the rotor disk is narrow, as in the simulation, the balancing is performed using static method with a single balancer only. The resultant force of the balancer disk unbalance must be equal to the centrifugal force acting on the unbalanced rotor disk. The centrifugal force at rotational frequency $f=16\ Hz$, acting on the weight $m=613\ g$ attached to the disk at the radius of *1150 mm* has a modulus of *7123 N*.

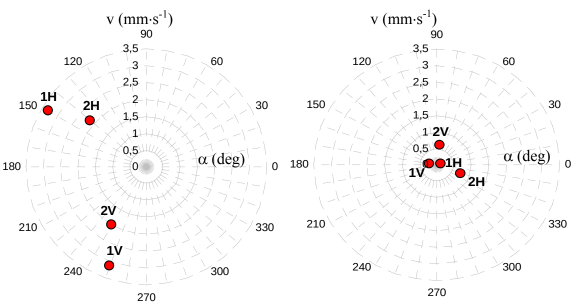

Fig.8. Initial state (A) and end state (B) of the rotor disk unbalance: v- vibration rate, α - phase angle

The force can be compensated by a single passive balancer if the angle of rotation between the disks is *157°* (Fig. 7). The result of the balancer disk setting translates into the system response showed in Fig. 8, with the ends of the unbalance vectors applied in the starting point of the polar reference system.

Use of the manual balancer is especially convenient for balancing the fan rotors with so called thermal unbalance. The thermal unbalance may occur when the rotor disk is surrounded by a high temperature gas stream. The thermal stresses result in disk deformations and change its weight distribution in relation to its rate of rotation, resulting in rotor unbalance, variable in time.

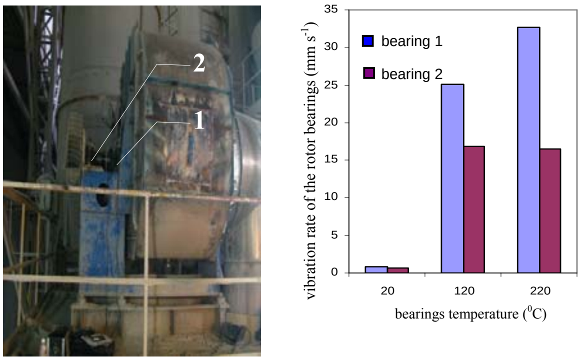

Fig.9. Effects of the temperature of air flowing through the spiral chamber on RMS vibration rate of the rotor bearings

A relation between temperature of air flowing through the spiral chamber of the radial fan and RMS vibration rate of the bearings recorded at the exhaust fan installed in a ceramic sanitary ware furnace is shown in Fig. 9 [23].

The rotor disk should be balanced again in case of the increase in bearing vibration rate due to thermal stresses. High flue gas temperature in the spiral chamber makes it difficult to perform a safe balancing procedure. Use of manual vibration eliminators guarantees a correct dynamic state of the rotor at operating fan temperature.

6. CONCLUSIONS

A manual balancer is a compromise between an automatic balancer, operating in continuous mode and a standard rotor balancing in its own bearings. The personnel involved in turbomachinery rotor balancing is aware of the difficulties associated with attaching the correction weights to the rotor disk. Time to attach the test and correction weights takes approx. 70% of time required for the vibration parameter measurements and correction vector calculations. Balancing the pump rotors, where the access to the rotor disk is almost impossible, since it is immersed in liquid is even more difficult. Using an automatic balancer is not cost effective due to high purchase costs and impossible in smaller rotors due to size of the device itself.

Although manual balancers are not as fast as automatic devices, the simple design makes them virtually fail-safe. The only component that may fail is the signal time course measuring unit, based on which the system response vector and thus the rotor unbalance correction vector are determined.

The positive results of the tests using manual balancers in industrial conditions are a proof of accuracy of the conclusions presented.

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