

Experience of an assessment of the vertical Francis hydroturbines vibration state at heads from 40 to 300 m

E. Dolmatov, A. Zaharov, S. Ilin, I. Kuznetsov, A. Nikiforov

OJSC Power Machines, 3A, Vatulina str.,
195009 St.Petersburg, Russian Federation
Baranova_ev@spb.power-m.ru

Abstract: The article covers a choice of main vibration parameter at an assessment of a vibration state of vertical Francis hydroturbines. At present time vibration velocity and vibration displacement are adopted as main parameters of non-rotating parts vibration in the international standard ISO 10816-5:2000 «Mechanical vibration — Evaluation of machine vibration by measurements on non-rotating parts — Part 5: Machine sets in hydraulic power generating and pumping plants» (further ISO 10816-5:2000).

The hydraulic turbines refer to the slow-speed machines with rotation speed from 60 to 600 rpm ($\sim 1 - 10$ Hz).

So maximum vibration displacements and dynamic stresses in hydraulic turbines supporting parts are in low-frequency region of vibration spectrum.

In this report comparative data of hydro units supporting parts vibration velocity and vibration displacement measurements are presented. Using these data assessment of hydro units vibration state has been done.

It is shown that the assessment of a hydro unit vibration state using parameter "vibration displacement" corresponds to the fundamental principles of operational reliability and fatigue strength of hydro units supporting parts.

It is noted that when hydro units operate at small and partial loads with high low-frequency unsteady flow ($f < f_{rot}$) we have the smallest vibration velocity and the greatest vibration displacement of hydro units supporting parts.

Specialists of LMZ (Saint-Petersburg) have developed Russian standard RD 24.023.117-88 «Vibration measurement and evaluation vibration state of vertical hydraulic turbines» which was published in 1989. In this document vibration displacement was considered as a main parameter. Evaluation of turbine vibration was performed according to the effective value of turbine supporting parts vibration displacement.

1. Choice of the main vibration parameter

The hydraulic turbines belong to the slow-speed machines with rotation speed from 60 to 600 rpm ($\sim 1 \dots 10$ Hz).

The characteristic feature of such machines is that maximum vibration displacements of supporting parts (and consequently dynamic stresses in them) occur in low-frequency region of 0.5...100 Hz.

In accordance with the working standards, hydro-unit vibration state is evaluated by supporting parts vibration (guide bearing, head cover, etc.) and shaft vibration levels.



The main frequencies of disturbing forces acting on the supporting parts and rotating parts are rotational frequency (f_{rot}), vortex frequency ($0.2...0.6f_{rot}$), vane frequency ($f_G = z_G \times f_{rot}$) and blade frequency ($f_R = z_R \times f_{rot}$).

Supporting parts vibration displacements on these frequencies are the largest in steady-state operating conditions that determine their stress state and service life.

Special attention should be paid to partial load operation conditions of the hydro-unit (Fig.1).

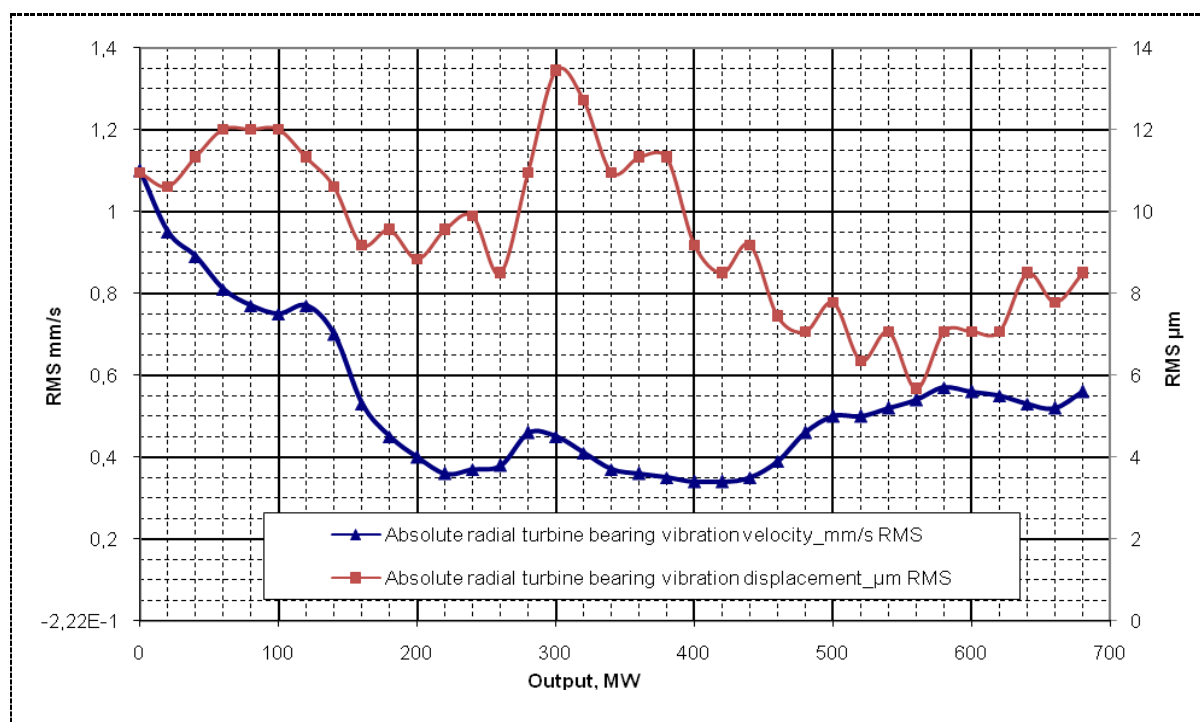


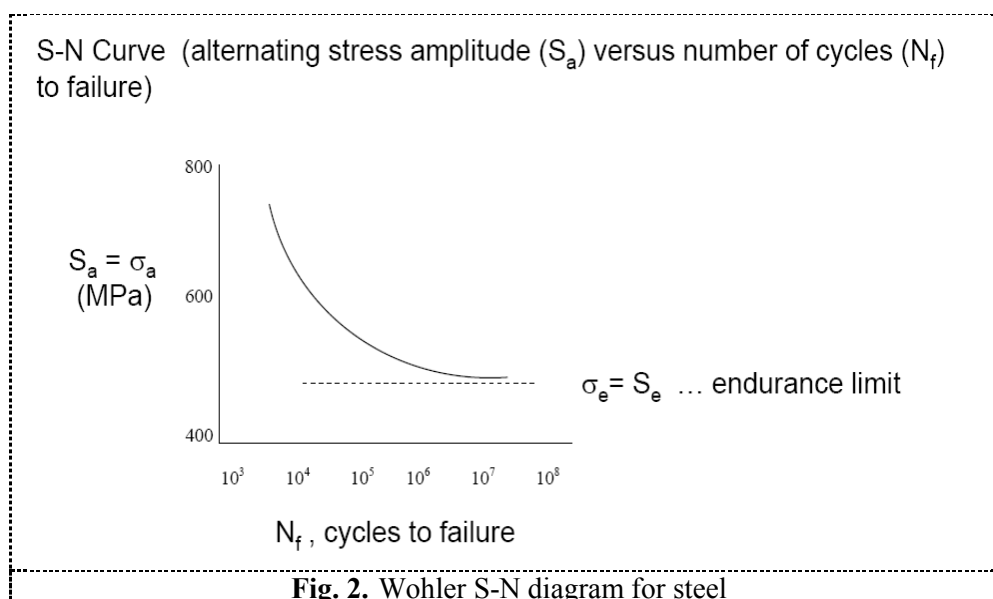
Fig.1. Absolute radial turbine bearing vibration

(Prated= 640 MW, Hrated = 194 m, nominal speed: 142.8 rpm)

It is evident that evaluation of supporting parts vibration state based on vibration velocity parameter can lead to incorrect results.

That is why evaluation of turbine vibration is performed according to the effective value of vibration displacement of turbine supporting parts.

As a rule, high frequency vibration displacement (100...350 Hz) of hydro-unit supporting parts does not exceed about 2...3 μm , which corresponds to dynamic stresses ≈ 2 MPa. This stress value is significantly less than endurance limit of the guide bearing material, which is equal to $0.6 \times \sigma$ (σ - ultimate tensile strength; for steel $\sigma = 420...480$ MPa). It is known that if steel sample has not been destroyed during 10^7 cycles, it will not get destroyed during more than 10^7 loading cycles (Wohler S-N diagram). As it is seen, the supporting parts of hydro-units cannot be destroyed under the action of high-frequency disturbing forces (Fig.2).



In special cases, for example, when carrying out diagnostic works intended to detect hydro-unit elements defects (such as runner blades, guide vanes, stay vanes, runner chamber, etc.) it is reasonable to use vibration velocity or vibration acceleration parameters. So, it is possible to get much wider frequency spectrum of disturbing forces acting on hydro-unit elements.

In addition, it should be noted that while measuring vibration displacements of turbine supporting parts with frequency less than 1 Hz, it is necessary to use special low-frequency contact or non-contact transducers.

So as it is shown above to evaluate hydro-unit vibration state is recommended to use effective value of supporting parts vibration displacement instead of vibration velocity parameter.

2. Method of an assessment of the vertical Francis hydroturbines vibration state

In 1989, the guidelines “Measurement of vibrations and assessment of vibration state of vertical-shaft hydraulic turbines” were published. This document has not been revised since 1989.

The guidelines cover vertical-shaft hydro turbines and pump-turbines. Limit levels for vibrations have been obtained based on the summarized data on vibrations of hydro units of 39 hydro power stations (including Francis, Pump-turbines and Diagonal turbines), the vibration state of which remained stable for a long period without failure.

There were hydro units with different numbers of bearings: for umbrella-type machines (where the weight of the rotating parts is supported by the head cover) – 2; for hanging-type machines (where the weight of the rotating parts is supported by the stator columns) – 3. In all cases upper generator bearings were not braced.

Measuring was carried out at previously specified places and at identical turbine operation conditions regarding their design, dimensions (runner diameters) and rotation speed (or heads).

Radial and vertical vibrations of turbine bearing, radial vibrations of generator bearing and shaft vibration displacement at the turbine bearing were considered as characteristics of turbine vibration state.

Main propositions that were taken in consideration when working out the turbine vibration standard were the following ones:

Method of comparison of turbine vibrations provided for storage of information on prototype tests that were carried out according to the unified procedure.

Vibration displacement was considered as a main parameter. This parameter is the most reasonable for turbines that are low-speed machines with rotation speed less than 10 rps. The relevant feature of such turbines is that the greatest vibrations and their components are located in a low-frequency area.

Evaluation of turbine vibration was performed according to the effective value (U_{eff}) of vibration displacement of turbine supporting parts. The effective value was adopted as a measuring quantity, which when compared to peak to peak is a stable value under the steady-state operating conditions and it directly represents the value relating to the energy of oscillation process and, therefore – to the breaking energy of these vibrations.

Acceptable levels of vibration are introduced on the ground of data generalization on vibration of turbines that distinguish with their reliable operation.

For sine stationary signal the mutual relation of effective value and double amplitude of oscillation is determined by the equation:

$$2A = 2\sqrt{2} U_{\text{eff}} \quad (1)$$

In case of complicated multicomponent vibration (vibration of turbine supporting parts) it is accepted to express these data in form of generalized parameters.

Generalization of turbine vibration values is based on the assumption that for supporting parts of reliably operated one-type turbines the ratio of dynamic component of stress effective value to static stress as well as the ratio of effective value of vibration displacement to static deflection are constants.

$$U^* = U_{\text{eff}} / U_{\text{st.max}} = \text{const} \quad (2)$$

Comparison of supporting parts vibration such as turbine head cover, upper and lower generator spiders and other supporting parts of each compared units are regarded as similar flat plates of constant thickness liable to force action.

Similar plates are the plates which have the similar contours in plan and boundary conditions over the corresponding areas of contour are identical.

For the compared plates $U_{\text{eff}} / U_{\text{st.max}} = \text{idem}$ is equivalent to the term $\bar{\sigma}_{\text{eff}} / \bar{\sigma}_{\text{st}} = \text{idem}$

Thus it becomes clear that parameter U^* defines dynamic stiffness and dynamic strength of the plate.

Generalized parameters were the following ones:

$$U^* = U_{\text{eff}} \times H^{0.5} \times D_1^{-1} \quad (3)$$

and

$$S^* = S \times H^{0.5} \times D_1^{-1} \quad (4)$$

where

- U^* and U_{eff} – generalized and measured parameters of vibration displacement of turbine supporting parts;
- S^* and S - generalized and measured parameters of peak-to-peak of shaft vibration displacement;
- H - maximum head;
- D_1 – runner diameter.

It is known that even for hydraulic turbines of the same series the level of measuring vibrations is not the same that evidently can be explained by geometric deviations while manufacturing or by quality of erection. That is why it is reasonable to consider unit values of similar hydraulic turbines designed for different conditions to be random quantities that have to be processed adequately. When determining the maximum (ultimate) values (U_{\max}^*) and (S_{\max}^*) it was assumed that the generalized parameters of vibration displacement are distributed according to the normal-logarithmic law and the maximum values of generalized parameters of vibration displacement U_{\max}^* and S_{\max}^* correspond to the 99 percent of probability

Parameters calculated are ultimate. Therefore it is accepted that values U_i^* which exceed U_{\max}^* are not admissible. The same is for S_i and S_{\max} .

Analysis of the field tests results showed that the values of generalized parameters are weakly dependent on the output power, if not to take into account some possible cases of deviations. Therefore generalized parameters calculated only for the maximum output power and maximum head were considered.

The area of the acceptable vibrations split out into «satisfactory» and «good».

Boundary of «good» rates area is 8 dB apart from the maximum vibration boundary of «satisfactory» area. On the basis of experience it is known that 8 dB (2.51 times) difference corresponds to increase or decrease of turbine vibration level that leads to essential change of its vibration state. «Unsatisfactory» operation means if equipment failure or damage caused by vibration occurred after commissioning, or after the last overhaul, operation is not permitted.

Transition from generalized parameters of vibration displacement to effective values of vibration displacement (which can be measured directly) can be performed as per formula (3) in the following way:

$$U_{\text{eff}} = U^* \times D_1 \times H^{-0.5} \quad (5)$$

In logarithmic coordinates:

$$\text{Lg } U_{\text{eff}} = \text{Lg } U^* + \text{Lg } D_1 - 0.5 \text{Lg } H \quad (6)$$

In logarithmic coordinates ratio (6) is expressed by straight lines which can be constructed on two arbitrary points.

Using formula $n'_1 = n \times D_1 \times H^{-0.5}$

where

n – rotational speed, rpm and

n'_1 - unit rotational speed, rpm,

effective values of vibration displacement can be performed as per formula:

$$U_{\text{eff}} = U^* \times n'_1 \times n^{-1} \quad (7)$$

In logarithmic coordinates:

$$\text{Lg } U_{\text{eff}} = \text{Lg } U^* + \text{Lg } n'_1 - \text{Lg } n \quad (8)$$

The analog transition from generalized parameters of shaft vibration displacement to peak to peak values of shaft vibration displacement can be performed as per formula (4) in the following forms:

$$\text{Lg } S = \text{Lg } S^* + \text{Lg } D - 0.5 \text{Lg } H \quad (9)$$

$$\text{Lg } S = \text{Lg } S^* + \text{Lg } n'_1 - \text{Lg } n \quad (10)$$

In logarithmic coordinates ratio (6), (8), (9) and (10) are expressed by straight lines which can be constructed on two arbitrary points, Fig. 3, 4, 5 and 6.

Numerical values of coordinates of straight line points which are the boundaries of turbine vibration areas with rates «good» and «satisfactory» for $D_1 = 10\text{m}$ and heads $H = 100\text{m}$ and 500m are represented in Table 1.

Table 1

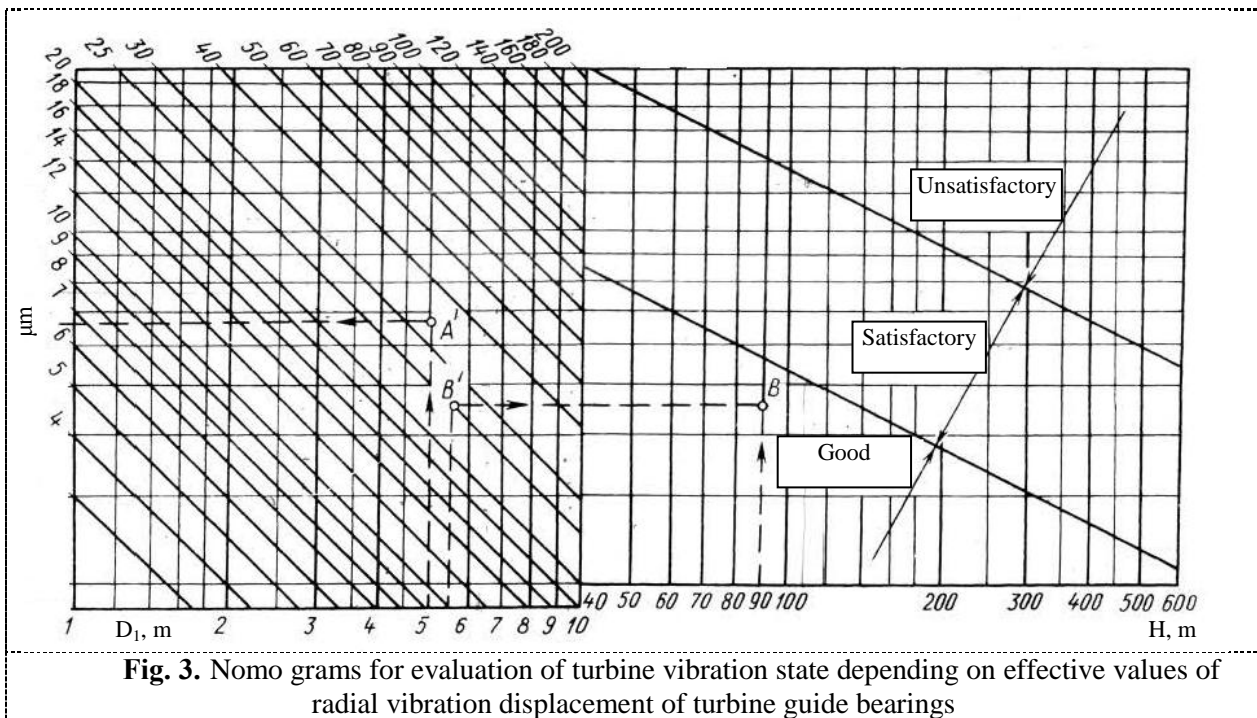
(a) –Vibration of supporting parts

No. of a point	H, m	Francis, Pump-turbines and Diagonal turbines			
		Effective value of radial vibration displacement, μm		Effective value of axial vibration displacement, μm	
		Satisfactory	Good	Satisfactory	Good
1	100	133,0	53,0	164,0	62,5
2	500	59,5	23,7	73,3	29,1

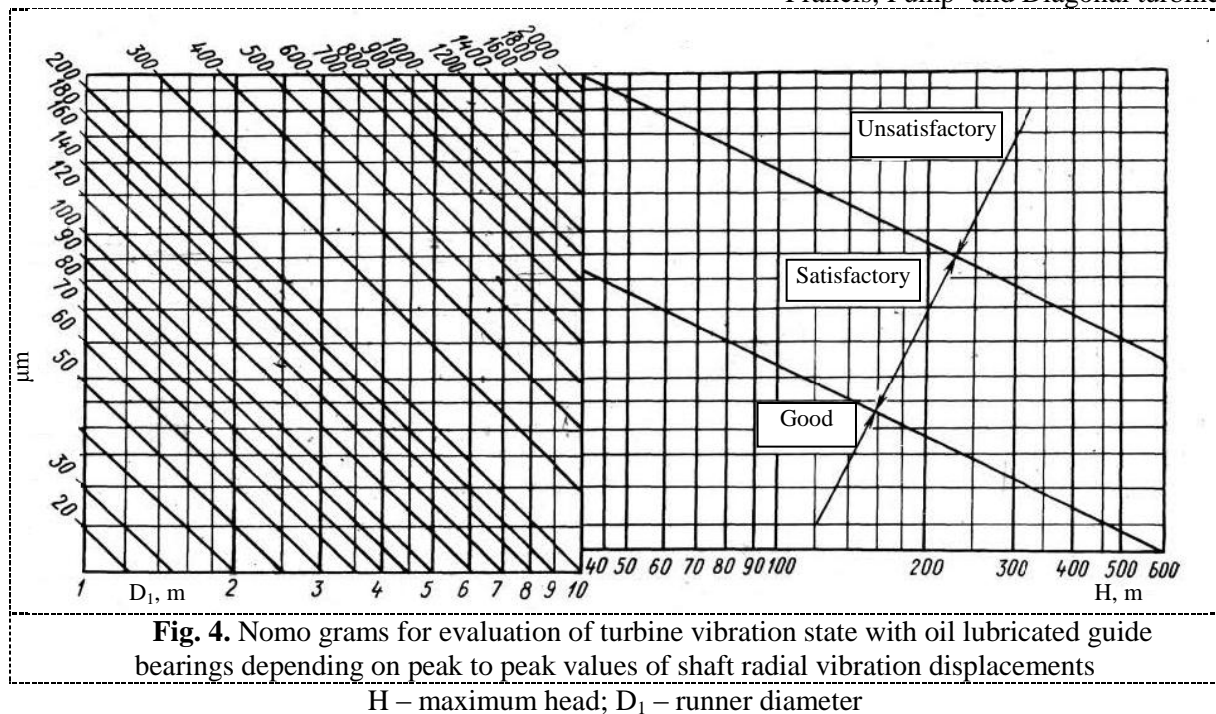
(b) –Vibration displacement of turbine shaft

No. of a point	H, m	Francis, Pump-turbines and Diagonal turbines	
		Peak to peak vibration displacement, μm	
		Satisfactory	Good
1	100	1334	531
2	500	597	237

Francis, Pump- and Diagonal turbine



Francis, Pump- and Diagonal turbine



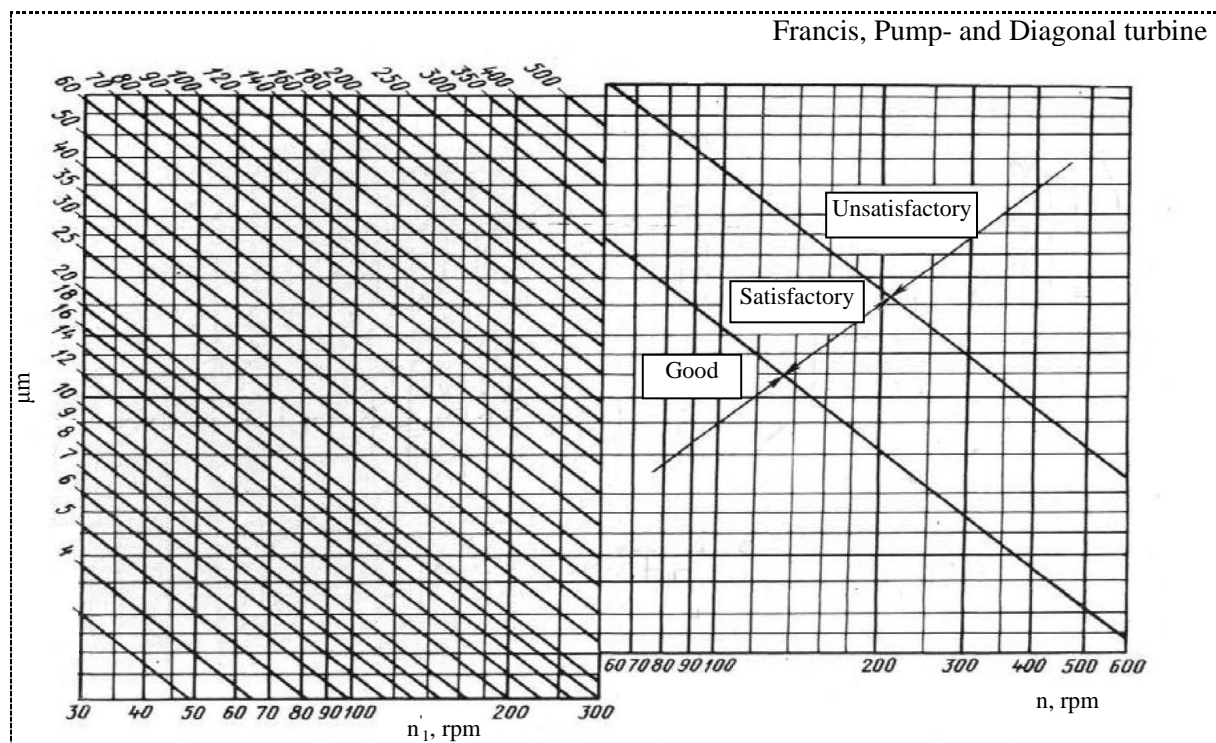


Fig. 5. Nomo grams for evaluation of turbine vibration state depending on effective values of radial vibration displacement of turbine guide bearings
 n – rotational speed, rpm; n_1 – unit rotational speed rpm.

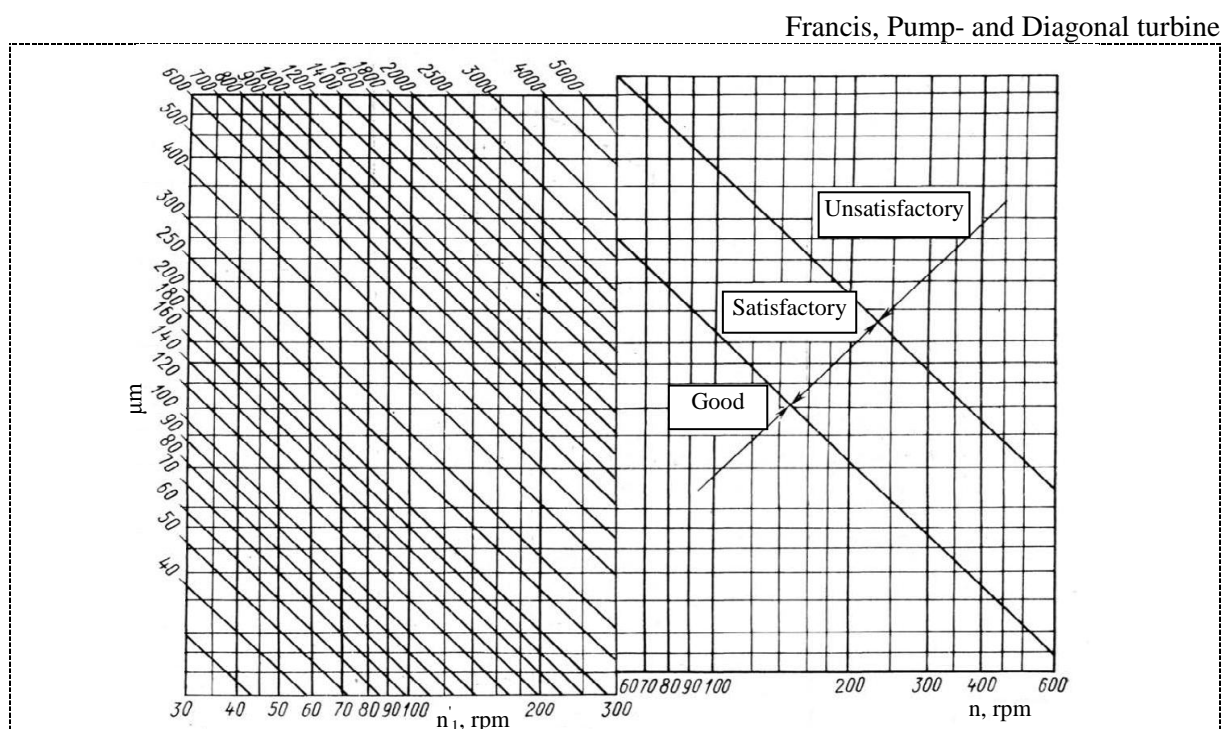


Fig. 6. Nomo grams for evaluation of turbine vibration state with oil lubricated guide bearings depending on peak to peak values of shaft radial vibration displacements
 n – rotational speed, rpm; n_1 – unit rotational speed rpm.

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