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- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

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Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100$^{\text{th}}$ Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery.* At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

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JÓZEF RYBCZYŃSKI*

Influence of journal bearing properties on critical speeds of complex turbocompressor train

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Abstract

Critical speeds of rotating machine depend grossly on the properties of rotor supporting structure. The properties of rotor support result mainly from the oil film stiffness of slide bearings. In the paper presented are the results of analysis of the influence of slide bearings parameters along with the results of calculations of critical speeds and shapes of vibrations of rotors supported in such bearings. Obtained results can prove useful during the design stage of rotors with respect to modification of dynamic properties of machine through relevant modification of bearings. On the other hand, the results enable assessment of the extent in which the insufficient knowledge of slide bearings characteristics can influence the results of calculations of critical speeds of rotating machine. A sample analysis has been conducted for a typical average size turbocompressor. Investigated has been the influence of relative changes of bearings length, relative bearings clearance and mean oil temperature on critical speeds of turbocompressor.

Keywords: Rotating machine; Slide Bearing; Critical speed

1 Introduction

Critical speeds of a real rotating machine depend not only on the geometric and material features of the rotor itself, but also depend to a large extent on the mechanical properties of the structure supporting the rotor. Rotor support properties depend on mechanical characteristics of the oil film of slide bearings as well as the characteristics of bearings support mechanisms consisting of the bearings stands and foundation. The set of values of critical speeds of the rotating machine forms its important characteristic feature due to the possibility

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of occurrence of resonance with periodical forces inducing vibrations, which are generated by the machine [1]. Industrial rotating machinery such as power engineering turbines should have the certificates confirming their expected dynamical properties. Such properties should be verified during the machine tests. Usually the client requires from the manufacturer certificates confirming that none from at least two first critical frequencies of each mechanically independent part of the machine is not lying in the immediate vicinity of operational rotational speed and even the doubled value of such speed.

Specific requirements with respect to dynamic properties of various type of rotating machinery are controlled by relevant standards. In the case of large rotating machinery most often these must conform to the API standards (American Petroleum Institute). API standards require, amongst the others, calculated values of natural bending and torsional vibrations of machine.

Presently such calculations are conducted on the basis of finite elements, which enables precise modeling of inertial and elastic properties of rotors. The condition for the correctness of the results is however precise data about the machine structure and the properties of the rotor support. Of significance is also the selection of adequate calculation model [2, 3]. Discrepancies between the results of calculations and the subsequent results of measurements on the assembled machine result mainly from the impossibility of account in calculations of the presence and flow of the working fluid as well as unprecise data characterizing the properties of rotor support [4, 5].

2 Objective of the work

The topic of the present work is analysis of the influence of changes of some construction and operation parameters of slide bearings on the critical speeds and corresponding forms of vibrations of rotors supported in such bearings. Knowledge of general characteristics, depicting the influence of changes of such parameters on rotor critical speeds, can prove adequate in forecasting and modifications of dynamical properties of rotating machinery. Information about in which direction and which degree can possibly to influence the machine dynamic properties through the modification of structure and parameters of bearings operation enable planned changes of dynamical properties of machine by modification of bearings. Such knowledge on one hand can have application in amendment of critical frequencies of rotors in order to detune them from the possible resonances. On the other hand possible is the analysis of errors, which are borne by rotor speeds and which can result from inadequate bearings calculations such as insufficient knowledge of a real bearings clearance, wrong assumptions with respect to oil properties, inadequate bearings model [6].

3 Object of investigations

Calculations have been performed for a typical, average size rotating machine consisting of three rotors. Assumed in the analysis machine is not any existing technical object, however its major features are taken from the real object. Despite the fact that calculations have been performed for the machine as a whole the results of calculation pertain in practice to its three particular rotors respectively. From the point of view of lateral vibrations, such machine can be decoupled and each rotor can be analysed independently with respect to critical speeds and forms of vibrations. Three rotors forming the machine are connected by means of flexible toothed couplings, which do not convey transverse forces and only the torque.

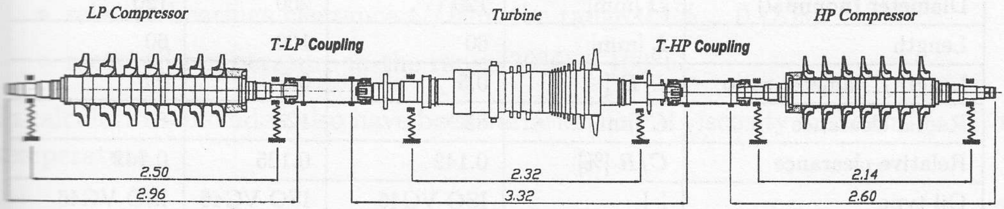

Figure 1. A schematic of the turboset.

A mechanical schematic of the machine has been presented in Fig. 1. It consists of a steam turbine driving the low pressure and high pressure parts of a gas centrifugal compressor. In the figure presented are the rotor lengths and the distance between bearings. Each rotor is supported in two identical slide bearings. All bearings are geometrically similar and are distinct only by the nominal diameter. The compressor bearings have a diameter of 120 mm whereas the turbine bearings – 200 mm. Bearings of both compressors are identical, however their load is different due to a different mass of compressor rotors. These are the bearings with five pivoted pads, each pad of same circumferential length. They are lubricated using ISO VG-46 oil. Assumed has been a simple and uniform structure of bearings, typical to that kind of machinery, in order to reflect to the best extent the influence of isolated selected parameters of bearings neglecting the secondary phenomena connected with a specific bearing structure. In such way eliminated have been from the analysis those variables, which could influence the results of calculations in untypical or unexpected way.

4 Methodology of the analysis

Calculations have been performed using the codes from the suite of computer codes, which have been developed at IFFM PAS for the analysis of the dynamic state of rotating machinery supported in slide bearings. This enables calculations of slide bearings of arbitrary type. Amongst the results of calculations are the coefficients of oil film stiffness. These serve as input data for calculations of dynamics of rotors supported in those bearings [5, 7, 8].

Table 1. Properties of bearings – base case.

Quantity	Nomenclature	Low pressure compressor	Turbine	High pressure compressor
Speed of rotation	n [r/min]	7500	7500	7500
Static load	F [N]	8279	10923	6779
Diameter (nominal)	D [mm]	120	200	120
Length	L [mm]	60	100	60
Length/Diameter ratio	L/D [-]	0.5	0.5	0.5
Radial clearance	C [μm]	85	125	85
Relative clearance	C/R [%]	0.142	0.125	0.142
Oil type	[-]	ISO VG46	ISO VG46	ISO VG46
Oil viscosity at 40°C	η [cP]	39.88	39.88	39.88
Oil density	ρ[kg/m^3]	867	867	867
Eccentricity ratio	e/C [-]	0.437	0.242	0.392
Attitude angle	Φ [deg]	53	56.9	54.2
Minimal film thickness	h_{min} [μm]	50.3	96.4	53.7
Film stiffness coefficient	k_{xx} [N/μm]	244.2	424.4	217.3
Film stiffness coefficient	k_{yy} [N/μm]	286.4	370.5	236.2
First critical frequency	f_{n1} [Hz]	47.62	71.79	57.88
Second critical frequency	f_{n2} [Hz]	138.5	157.6	160.7
Third critical frequency	f_{n3} [Hz]	299.9	228.9	342.1
Fourth critical frequency	f_{n4} [Hz]	427.7	335.2	453.4

Investigations, of which the results have been presented here, were based on calculations of critical speeds of machines, where the major structural parameters of bearings were regarded as variable. Primarily, assumed as optimal bearings features served as input data for the base case. The results of rotor calculations for the base case served as a material for comparisons against the results of calculations of rotor critical speeds performed for the rotor parameters regarded as variable. The characteristic data of bearings for particular rotors for the base

ase have been presented in Tab. 1. The table contains also the results of calcu-ations for the base case and in particular, the coefficients of oil film stiffness.

The results of calculations of slide bearings using the NLDW code for the earing parameters regarded as variable are presented in Tab. 2. Presented here re the oil film coefficients of stiffness for all analysed cases connected with the hange of bearings distance L/D, the bearings clearance C/R and the oil tem-erature t. The base case has been distinguished with the bold font.

Calculations of rotor critical speeds and forms of vibration have been con-ucted for the support stiffness equal to a mean oil film stiffness in the horizontal nd vertical directions: $k = (k_{xx} + k_{yy})/2$. Calculations for two independent sup-ort stiffness, horizontal and vertical ones, lead to two different critical speeds: orizontal and vertical ones, which overshadows the results of analysis. The ollowing bearings parameters have been varied:

- relative bearings distance L/D in the range 0.35... 0.95,
- relative bearings clearance C/R in the range 0.1%... 0.18%,
- mean oil temperature in the range 30°C... 100°C.

n calculations included also have been variations of oil viscosity and density with emperature.

5 Results

Calculated forms of vibrations for the base case in the light of the machine chematic have been presented in Fig. 2. Underneath each machine part, i.e. w-pressure compressor, turbine and the high pressure compressor presented are heir first four forms of free vibrations in a normalized scale. In each figure esented are also corresponding critical speeds. It can be seen from the figure at all machine parts have similar four subsequent forms of vibrations. Also, e second form of vibrations for the turbine is similar to the second form of rations of compressor, although in the figure scale seems to be different.

Figure 3 contains the critical frequency distributions of particular machine arts in function of analysed bearing parameters:

- relative bearings distance L/D – in the upper raw,
- relative bearings clearance C/R – in the middle raw,
- mean oil film temperature – in the bottom raw.

ach of the nine elementary plots presents the distributions of variation of four st critical frequencies. In Fig. 4 presented are analogically the distributions f relative, expressed in percentages, frequency changes with respect to critical

Table 2. Oil film stiffness of turboset bearings for various values of bearings width, bearing clearance and oil film temperature.

	Oil film stiffness K [·10e9 N/m]								
	LP Compressor			Turbine			HP Compressor		
L/D	L [mm]	K_{xx}	Kyy	L [mm]	K_{xx}	K_{yy}	L [mm]	K_{xx}	K_{yy}
0.5	60	0.2442	0.2864	100	0.4244	0.3705	60	0.2173	0.2362
0.3	36	0.1299	0.4566	60	0.2142	0.2699	36	0.1159	0.3443
0.4	48	0.1987	0.3293	80	0.3044	0.2905	48	0.1773	0.2524
0.6	72	0.2868	0.2866	120	0.5515	0.4877	72	0.2675	0.2437
0.7	84	0.3315	0.3082	140	0.6968	0.6089	84	0.3138	0.2757
0.8	96	0.3874	0.3344	160	0.8644	0.7255	96	0.3639	0.3178
0.9	108	0.4531	0.3629	180	1.0363	0.8435	108	0.4372	0.3433
	LP Compressor			Turbine			HP Compressor		
C/R	C [μm]	K_{xx}	K_{yy}	C [μm]	K_{xx}	K_{yy}	C [μm]	K_{xx}	K_{yy}
1.42/1.25	85	0.2442	0.2864	125	0.4244	0.3705	85	0.2173	0.2362
0.1	60	0.5042	0.4654	100	0.7547	0.6644	60	0.472	0.433
0.11	66	0.4018	0.3955	110	0.5957	0.5052	66	0.3791	0.3483
0.12	72	0.3375	0.342	120	0.4666	0.4196	72	0.3079	0.2993
0.13	78	0.2841	0.3142	130	0.3816	0.3406	78	0.2621	0.2602
0.14	84	0.2498	0.2892	140	0.3133	0.2925	84	0.2227	0.2392
0.15	90	0.2171	0.2789	150	0.2711	0.2487	90	0.1972	0.2198
0.16	96	0.196	0.2664	160	0.2381	0.2161	96	0.1722	0.2114
0.17	102	0.173	0.2639	170	0.2024	0.2017	102	0.1561	0.2013
0.18	108	0.1199	0.2934	180	0.1828	0.182	108	0.1369	0.2002
	LP Compressor			Turbine			HP Compressor		
t[°C]	μ [cP]	K_{xx}	K_{yy}	μ [cP]	K_{xx}	K_{yy}	μ [cP]	K_{xx}	K_{yy}
40.5	39.88	0.2442	0.2864	39.88	0.4244	0.3705	39.88	0.2173	0.2362
30	66.88	0.3125	0.3072	66.88	0.6342	0.553	66.88	0.2949	0.2706
40	40.715	0.2464	0.2859	40.715	0.4234	0.3797	40.715	0.2196	0.2363
50	26.431	0.1534	0.3683	26.431	0.3194	0.2957	26.431	0.1848	0.2437
60	18.098	0.1364	0.4128	18.098	0.2626	0.2622	18.098	0.1195	0.3164
70	12.957	0.1243	0.4617	12.957	0.2304	0.2564	12.957	0.1108	0.3488
80	9.629	0.1115	0.5174	9.629	0.2101	0.2662	9.629	0.0962	0.3932
90	7.385	0.0939	0.5828	7.385	0.1934	0.2863	7.385	0.0867	0.4353
100	5.819	0.0834	0.6436	5.819	0.1392	0.3495	5.819	0.0776	0.4792

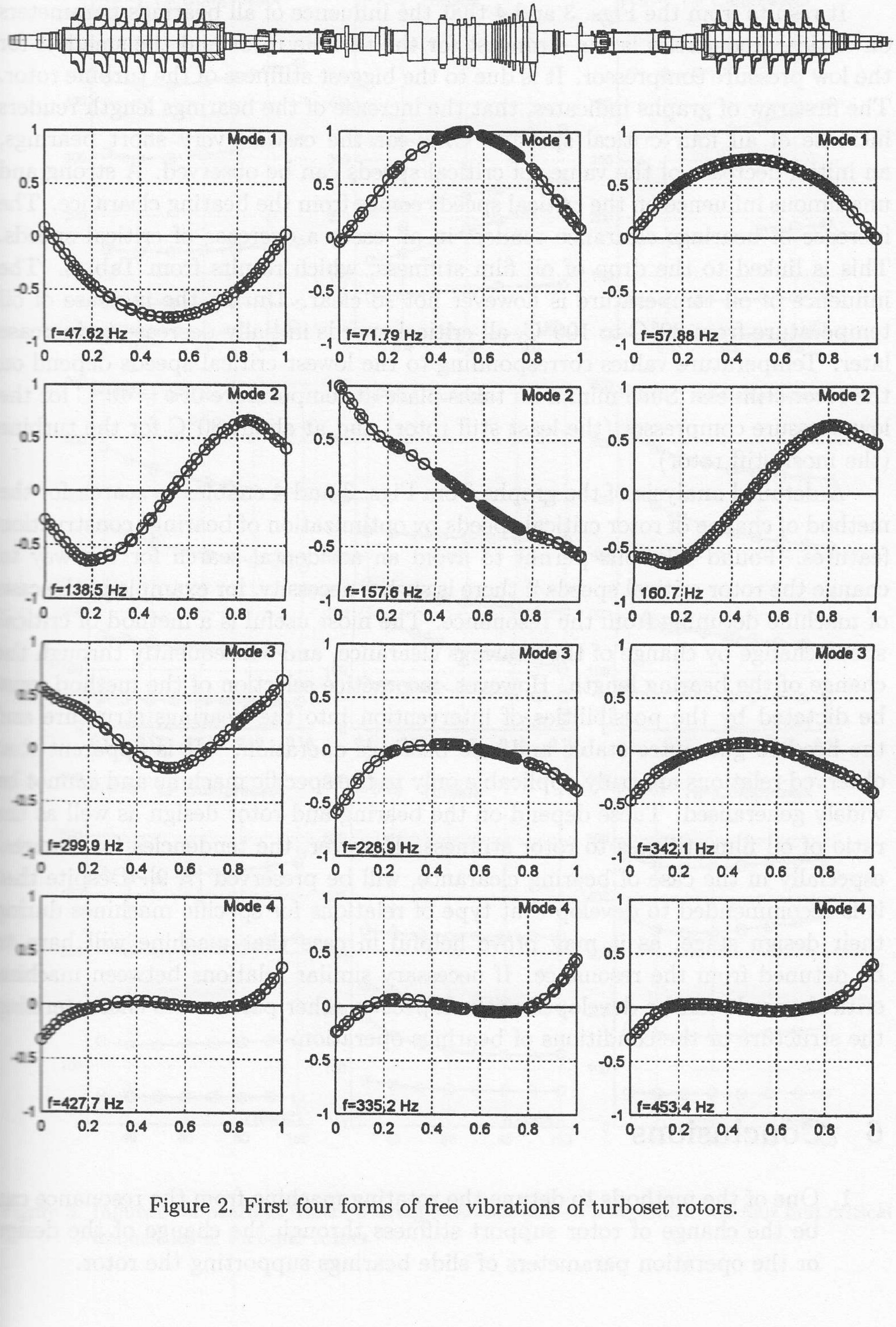

Figure 2. First four forms of free vibrations of turboset rotors.

frequencies of the base case $\Delta f_n / f_{nb}$.

It results from the Figs. 3 and 4 that the influence of all bearings parameters on critical frequencies is the strongest for the turbine rotor and the smallest for the low pressure compressor. It is due to the biggest stiffness of the turbine rotor. The first raw of graphs indicates, that the increase of the bearings length renders increase of all four critical speeds. Only for the case of very short bearings, an initial decrease of the values of critical speeds can be observed. A strong and unanimous influence on the critical speeds comes from the bearing clearance. The increase of bearings clearance renders in all cases a decrease of critical speeds. This is linked to the drop of oil film stiffness, which results from Tab. 2. The influence of oil temperature is however not so clear. During the increase of oil temperature from 30°C to 100°C, all critical speeds initially decrease to increase later. Temperature values corresponding to the lowest critical speeds depend on the rotor stiffness. Such minimum takes place at temperature of $t = 40$°C for the low pressure compressor (the least stiff rotor) and at about 80°C for the turbine (the most stiff rotor).

A detailed analysis of the graphs from Figs. 3 and 4 enables to search for the method of change of rotor critical speeds by optimization of bearings construction features. Found relations permit to avoid an accidental search for the way to change the rotor critical speeds if there is such a necessity, for example in the case of machine detuning from the resonance. The most useful is a method of critical speed change by change of the bearings clearance, and subsequently through the change of the bearing length. However, in practice selection of the method must be dictated by the possibilities of intervention into the bearings structure and the need to guarantee stable and safe bearings operation. It is apparent that observed relations are truly applicable only to the specific machine and cannot be widely generalised. These depend on the bearing and rotor design as well as the ratio of oil film stiffness to rotor stiffness. However, the tendencies for changes, especially in the case of bearing clearance, will be preserved [7, 9]. Despite that it is recommended to develop that type of relations for specific machines during their design stage, as it may prove helpful in case that machine will have to be detuned from the resonance. If necessary similar relations between machine critical speeds can be developed with respect to other parameters characterizing the structure or the conditions of bearings operation.

6 Conclusions

1. One of the methods to detune the rotating machine from the resonance can be the change of rotor support stiffness through the change of the design or the operation parameters of slide bearings supporting the rotor.

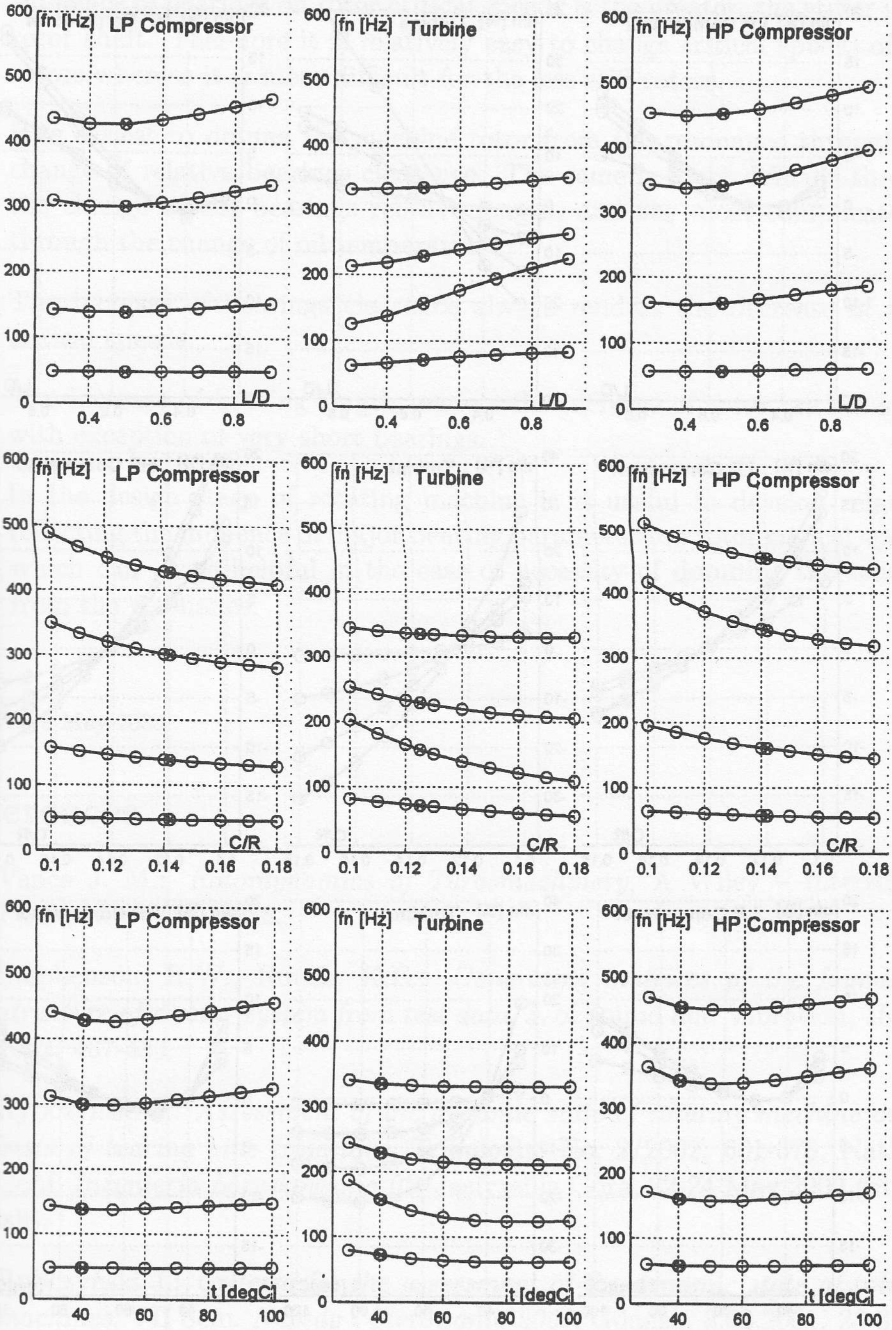

Figure 3. Influence of bearings width, bearing clearance and oil temperature on four first critical frequencies of turboset rotors.

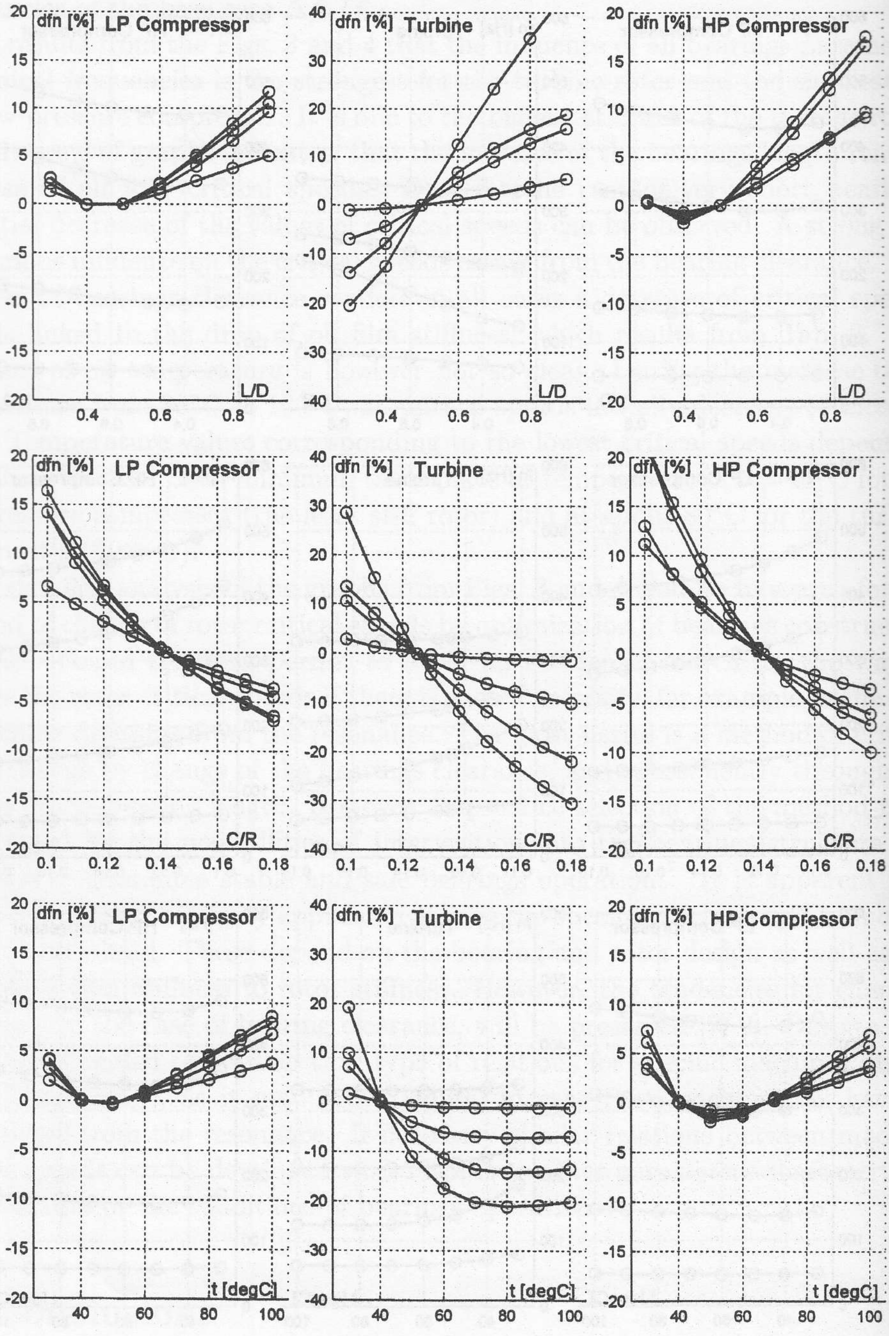

Figure 4. Influence of bearing width, bearing clearance and oil temperature on four first critical frequencies of turboset rotors expressed in percentages with respect to the base case.

2. Influence of bearings on rotor critical speeds is the greater, the stiffer is the rotor shaft. Therefore it is relatively easy to change critical speeds of stiff rotors whereas it is more difficult for the less stiff rotors.
3. It is easiest to detune the machine rotor from the resonance through the change of relative bearings clearance. The same is more difficult through the change of the bearings relative length, and the most complicated is through the change of oil temperature.
4. The increase of bearings clearance always renders the decrease of rotor critical speeds.
5. The increase of bearing length renders the increase of rotor critical speed with exception of very short bearings.
6. In the design stage of rotating machine it is useful to develop relations reflecting the influence of major bearing parameters on rotor critical speeds, which can prove helpful in the case of necessity of detuning the machine from the resonance.

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