APPLICATION OF COST EFFECTIVE CONDITION BASED MAINTENANCE & OVERHAULING PRACTICES IN NTPC

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ABSTRACT

NTPC has a fleet of turbine sets of 12 x 200 MW & 10X500 MW capacity of KWU design. The overhauling practices generally being followed, are in line with manufacturer's recommendations. The decision, for deviations in overhaul schedules/work scope, whenever taken is based on the health assessment of turbines through ***Predictive Maintenance techniques*** *such as Vibration monitoring, Noise monitoring and operational history etc.*

These machines are experiencing high shaft vibration problem at HPT front during commissioning and even after overhauling. In order to ensure the satisfactory operation of these machines, it is necessary to maintain the maintenance protocol within their design tolerances. Inspite of maintaining all these maintenance norms close to the design values, there had been instances wherein high HP shaft vibration problem have been experienced. The problem was critically analyzed using above techniques and the root causes such High swing of rotor, Unbalance in far away planes such as LP-GEN, Catenary disturbance and inadequate gland clearances were identified.

Inferences drawn through proven field experiences have been brought out in this paper alongwith current techniques and practices for monitoring, limiting values and acceptable criteria and possible remedial measures for improvement in shaft vibration which could help in reduction of the overhaul cost through reliable and trouble free operations.

1.0 INTRODUCTION

NTPC is the largest power public utility in India, having in operation 7 Super Thermal Power stations and 5 Gas based Power operating stations. The critical nature of power supply situation in the country, has necessitated the requirement of high equipments availability. In order to sustain and maintain the high availability, the scheduled overhauling practices, as prescribed by the manufacturers are normally adhered to but in some cases, after introducing condition based maintenance philosophy, it became necessary to deviate from the manufacturer's recommendations without sacrificing the performance and intended care. Thus, we have been able to achieve higher availability by diagnosis of specific problems of turbine generators and rotating machines by resorting to short stoppage, as required rather than the adhering to standard interval for the overhauling originally prescribed. Thus, this technique has helped to restore the desired normal health and process parameters of the plants, same as what were originally intended by judicious postponement of major overhauls. In fact it has proved to be most cost effective means of preventative maintenance.

The application of Condition Based Maintenance was adopted in high speed turbines as a tool for extending the availability as mentioned above. The vibration spectrum analysis has been used to resolve chronic and generic problem of KWU turbines related to HIgh pressure turbine shaft vibration. These machines are experiencing high HPT shaft vibration during commissioning and even after overhauling. In order to identify the probable causes, an exhaustive monitoring survey both within and outside NTPC was conducted, which enable database for coming to conclusions and arriving at solutions.

The efforts have been made to assimilate the probable causes, based on the field experience. To substantiate, the various actions suggested to arrest HPT front vibration, experience gained in NTPC stations, is included in this paper. This may serve as a guidelines in tackling similar types of problems

Which may arise in other stations. The shaft vibration monitoring strategies for Turbine generator and their interpretations are also included alogwith the acceptable standards being followed internationally. The correlation between the bearing and shaft vibration has also been explained in brief.

2.0 HIGH SHAFT VIBRATION PROBLEMS EXPERIENCED IN NTPC STATIONS & ELSWHERE

High shaft vibration problems were experienced at SSTPS, KSTPS RSTPS, FSTPS Kota T.P.S. and FGUTPS. In SSTPS the shaft vibration at HPT front had rising tendency with increase in load which was caused by rubbing in glands as gland clearances in LP- Turbine were not adequate. Subsequently the same was corrected and problem resolved. At KSTPS, the shaft vibration had similar tendency but having different vibration characteristics, the predominant frequency was observed at 1/2 x rpm which was suspected to be due to self excitation i.e. steam whirl or oil whirl. The problem was temporarily resolved by reducing unit load and subsequently resolved by adjusting the axis of rotation close to the mass principal axis. The high shaft vibration problem of the same unit had been resolved earlier by replacement of journal bearing No. 1 It concludes that system damping can be improved by changing the bearing resulting in reduction of shaft vibration. At RSTPS problem was resolved by reducing the swing check value by correcting the coupling facial runout. The problems at other stations were also resolved which had resulted from disturbance in catenary or misalignment/unbalance in the rotors. Based on the authors experience the probable causes are assimilated to provide assistance to site engineers to deal with similar types of problems quickly and effectively in future.
The high shaft vibration problems resolved/being resolved in NTPC are presented here.

CASE NO. 1 STATION:- RSTPS 500MW, UNIT IV.

HPT front bearing had high shaft vibration maximum of the order of 103 microns 0->peak since commissioning of the unit and also observed during PG test.

After Commissioning in June 1988.

DATE	MW	FREQUENCY SPEED	SHAFT VIBRATION
09.11.88 Gradually increase to	300	3004	65 MICRONS
02.01.90 During P.G. TEST	500	2940	95 MICRONS
02.03.90 After overhauling	500	2940	100 MICRONS
09.02.91	394	2894	120 MICRONS

a. **ACTION CARRIED OUT IN FIRST SHORT STOPPAGE**

Again, unit was hand tripped in June '90 and carried out the following activities.

i. Checking of shaft runout
ii. Inspection of MOP bearings
iii. Adjustment of bearing clearances
iv. Lifting the front bearing pedestal to load the HP front by adding 0.3 mm shims.
v. Checking of front pedestal sliding clearance and
vii. Realignment of MOP and HP shaft.

After rolling the machine no improvement was noticed in HPT front shaft vibration instead it increased to 125 microns 0->peak.

b. **ACTION PLANNED IN SECOND SHORT STOPPAGE**

During the overhauling, MOP bearings were replaced as abnormal sound was noticed from bearing no 1. Vibration problem continued to exist.

ACTION CARRIED OUT IN THIRD SHORT STOPPAGE

Turbine was tripped and vibration Signature was taken which indicated 1x rpm frequency was only predominant. Finally attempted to balance the rotor as unbalance was suspected, one of the causes for vibration.

Though during the initial rolling shaft vibration came down to 48 microns but subsequently increased to (140 microns 0->peak). It was decided to remove the weight (40 grams) fixed in HP front groove.

BRG	Before adding weight No load	Before adding weight Load	After Adding weight No Load	After Adding weight Load
B1	80	120	50	140
B2	45	75	40	90
B3	45	15	55	15
B4	50	30	25	30

d. ACTION CARRIED OUT IN FOURTH SHORT STOPPAGE

The following works were carried out during overhauling.

i Bearing inspection.
ii. Roll check, swing check of HP rotor,
iii. Alignment checking of HP-IP coupling.
iv. MOP inspection and oil guard setting

The J1 runout was high and bearings conditions were checked by UT and DPT and found to be o.k. Thrust float checked and found to be .33. Swing check value was measured and found to be on higher side R=-.33; L=+.33.

MOP_HP and HP_IP couplings alignment was carried out. The roll check was done and final readings were recorded as follows

v. Hangers also adjusted and horn drop test carried out. Vibration values increased by 5 microns 0->Peak

e. ACTION CARRIED OUT IN AUGUST '93

In order to reduce the swing check value the followings jobs were carried out.

i. Facial runout of HP & IP coupling were found to be 0.035 and 0.03 respectively. These high points were coupled together causing the swing of .39 on No. 1 Bearing.
ii. Brown burnt marks were noticed in the inner faces
iii. The faces were scrapped and swing check reading observed to .13 mm with loose coupling bolts. But swing checked again with tight coupling-bolts and found to be .17mm.

RESULTS : VIBRATION LEVELS REDUCED SIGNIFICANTLY AFTER CORRECTION OF SWING.

CCR READINGS LAOD 375 MW

Shaft	Before correction Pedestal	Before correction Shaft	After correction Pedestal	
BRG NO.1	115	15	75	12
BRG NO.2	75	10	55	8.5
BRG NO.3	25	5	28	4
BRG NO.4	15	7	20	8

CONCLUSION :- COUPLING OF ROTORS WITH HIGH POINTS ON COUPLING IN THE SAME PLANE CAN CAUSE HIGH SWING AT BRG NO.1 WHICH CAN RESULT IN HIGH SHAFT VIBRATION. CORRECTION OF SWING CAN REDUCE SHAFT VIBRATION.

CASE NO. 2 STATION-KSTPS 500MW, UNIT IV

Maximum shaft vibration of the order of 90-100 microns 0->pk was observed at HPT front since recommissioning.

ACTION CARRIED OUT
As done in case of RSTPS case cited above before August '93 overhaul and in addition to this Bearing no. 1 was replaced.

RESULTS :- SHAFT VIBRATION REDUCED TO 50 MICRONS 0->PEAK FROM 100 MICRONS 0->PEAK

•• CONCLUSION :-ADEQUATE BEDDING AREA AND BEARING CLEARANCES, PROPER SEATING OF BEARING, CAN IMPROVE SYSTEM DAMPING.

CASE-3 FSTPS 500 MW UNIT IV

Max shaft vibration of the order of 140 microns 0->peak observed since commissioning of the unit in Sep '92. Vibration signatures indicated that high bearing/shaft vibration was due to unbalance or improper bearing seating. The machine was tripped and re-synchronized on 12.12.92 but shaft vibration was found to be still higher (max 160 0->Peak) side on the bearing no. 1. The machine was hand tripped and further line of action planned.

a. ACTION CARRIED OUT IN MARCH '93

i) HPT front/exhaust gland seal inspected.

ii) MOP adjusted and aligned.

iii) Bearing oil and yoke clearances checked.

iv) Balancing was carried out as follows On 01.03.93m, a weight of 160 grams was put on plane i.e. on HPT front (groove), and machine was rolled to 3000 rpm on 27.02.93. The shaft vibration level were recorded at all the bearings

Balancing of HP rotor

27/2 BEFORE BALAN.	01/03 160gm HPTF.	02/03 160gm HPTR 400gm HPTR	03/03 160gm HPTF	04/03 wt moved by 30deg gm.HPTR	05/03 wt 160gm HPTF & 400
B1	130	131	112	121	139
B2	40	50	43	52	53
B3	70	70	68	72	67
B4	38	41	33	46	43

No change in the vibration level was noticed even after putting the above weights.

b. ACTION CARRIED OUT IN April '93

i) Bearing no 1 was replaced with new bearing along with spherical seat.

ii) Roll check, swing check HP/IP was carried out.

iii) Balancing was carried out through LP planes
Machine was rolled to 3000 rpm and the following combination of wright & locations were tried in LPT Front and LPT Rear.

BALACING THROUGH LP PLANES

	TRIAL I	II	III	IV	V	VI
	LPR-360gm at 0°	LPR-360gm at 180°	LPR-720gm at 0°	LPR-720gm at 160°	LPR-360 at 0° &720gm at 240°	LPR-360 at 0° 720gm at 240°
	LPF-360gm at 180°	LPF-360gm at 0°	LPF-360gm at 180°	LPF-360gm at 30° &720gm 180°	LPF-360gm at 60°' & 720gm at 180°	LPF-36° at 60°
	15/4/93	16/4/93	16/4/93	16/4/93	17/4/93	18/4/93
Speed	3009	2997	3000	3000	3000	3000
B1	179	138	142	122	114	88
B2	67	46	46	40	37	17
B3	62	45	42	85	34	58
B4	34	20	19	46	20	27

After doing above jobs the vibration levels significantly reduced to 90 microns 0->peak from 140 microns 0->peak.

RESULT :- VIBRATIONS REDUCED TO 90 MICRONS 0->PEAK AFTER BALANCING THROUGH LP PLANES.

*****CONCLUSION :- HPT FRONT SHAFT VIBRATION CAN BE CONTROLED BY BALANCING EVEN AT FAR AWAY PLANES. MASS PRINCIPLE AXIS CAN BE BROUGHT CLOSER TO AXIS OF ROTATION BY BALANCING***

CASE NO 4 FSTPS UNIT 1 200 MW

Maximum shaft vibration of the order of 200 microns 0->peak was noticed at HPT front after IP overhauling. The machine was rolled after overhauling to 3000 rpm and synchronized. But it got tripped as gland steam temperature was low. Later it did not come on barring & also seal rubbing observed.

ACTION CARRIED OUT SUBSEQUENT TO BARRING GEAR PROBLEM

i) HP/IP Roll check adjustment.
ii) Swing check at HP front was also done and found to be o.k.
iii) Bearing no 1 & 2 inspected and found to be o.k.
iv) The pedestal no 1 was lifted by .15 mm for adequate loading

RESULT :- MACHINE CAME ON BARRING AFTER ADJUSTMENT IN CONTROL DIMENSION

*****CONSLUSION :- AFTER ADJUSTMENT OF PALM KEYS, PACKERS TO THE RECOMMENDED VALUES AND GLAND STEAM TEMPERATURE AS PER DESIGN. THE SEAL RUBBING CAN BE ELIMINATED AND BARRING ENGAGEMENT PROBLEM CAN BE RESOLVED.***

CASE - 5 STATION - FSTPS 200 MW UNIT 1

Maximum shaft vibration observed after Gen rotor replacement. The generator rotor replaced as machine tripped on generator earth fault protection. Coupling alignment and faces were checked. Machine rerolled to 3000 rpm. The shaft vibration of the order of 66 microns observed at 3000 rpm but machine was continuously kept running to 3000 rpm for dry out. In the mean time shaft vibration increased to 150 microns. Vibrations and their phase were taken which indicated that this high shaft vibration was due to unbalance. The trim balancing was planned. Trial weight and its approximate locations were calculated. Machine was stopped to put the weight on the coupling as no other plane was available. Pedestal cover was opened for putting the trial weight, and found that one half of coupling guard was dislocated from its location and found inside the bearing housing. The location of trial weight as calculated was the same from where the coupling guard was missing.

ACTION CARRIED OUT IN AUGUST '93

i) Coupling guard replaced with new one.
ii) Coupling guard bolts were also replaced.

Vibration measurement readings taken before and after coupling guard replacement

BEFORE		AFTER	
3000RPM	3000RPM	LOAD	LOAD
4.10 HRS	7.30 HRS	159 MW	179 MW
B1 66	125	156	74 MICRONS

RESULTS: - HPT FRONT VIBRATION CAMEDOWN AFTER REPLACEMENT OF COUPLING GUARD.

***CONCLUSION :- HPT FRONT VIBRATION CAN BE AFFECTED BY UNBALANCE EVEN AT LOCATION AS FAR OFF AS LP-GEN COUPLING.*

CASE-6:- STATION-FGUTPS 200 MW UNIT 2

Maximum shaft vibration of the order of 140 microns at HPTF and high vertical and axial vibration in bearing no 6. observed since commissioning. The shaft vibration had a tendency to increase when grid frequency was decreasing, and decreasing when grid frequency increases.

ACTION CARRIED OUT IN JAN '93.

Bearing no 6 was inspected and bedding area was increased which was found to be inadequate. Main oil pump was overhauled. No work was done in any of the bearings except pickup calibration. After putting bearing cover shaft vibration reduced to normal i.e. 70 microns.

SHAFT VIBRATION BEFORE SHUT DOWN & AFTER CALIBRATION

	BEFORE		AFTER	
SPEED	2904	3064	2901	2991
B1	132	92	104	78
B2	12	8	17	8
B3	98	79	94	94
B4	30	21	19	16

RESULTS : THE SHAFT VIBRATION REDUCED AFTER CALIBRATION.

***CONCLUSION : THE PICKUP CALIBRATION IS REQUIED TO BE DONE AS PER MANUFACTURER RECOMMENDATIONS OR ONCE IN SIX MONTH WHICH EVER IS EARLIER OTHERWISE IT CAN LEAD TO UNNECESSARY MACHINE SHUTDOWN.*

CASE-7 STATION KOTA 200MW UNIT IV

The machine was first rolled and synchronized in May 88 and full load was achieved on 25/01/90. The shaft vibration level at bearing no 1 was observed to be normal till May '90 (maximum of the order 50-60 microns 0->peak). After this the vibration behaviour of bearing no 1 started deteriorating till first overhaul due on Feb '91. The axial vibration on the bearing no 6 also deteriorated in this period.

THE VIBRATION BEHAVIOUR OF MACHINE RECORDED PRIOR TO THE OVERHAUL.

	Shaft vibration	Bearing pedestal vibration		
		V	H	A
B1	120-140	microns	14	8
B2	45	22	20	33
B3	25	5	16	6
B4	55	53	16	78
B5	-	29	20	65
B6	-	71	38	131

ACTION CARRIED OUT IN OVERHAULING

i) Overhaul of LP turbine
ii) Repair of generator rear pedestal at BHEL Hardwar.
iii) Realignment of turbine generator rotor system.
iv) Roll checks, of HP/IP turbine.
v) Reaming, Honing LP-Gen coupling.

IMMEDIATE AFTER OVERHAULING THE FOLLOWING OBSERVATIONS WERE MADE

i) BRG no.1 babbit metal temperature low
ii) Sticking of sealing segment of HP/IP turbine gland.
iii) Low Jacking oil pressure at bearing no. 1 & 2 confirming unloading of bearing no 1 but bearing no 2 babbit metal temperature found to be o.k. indicating good loading.
iv) Improper babbit metal temperature of Thrust pads both before and after overhaul. Fitment/ malfunctioning of thermocouples of higher pad thickness though the blue match was done during overhaul.
v) Radial clearances of IP rotor in IP casing were found to be less.
vi) Coupled runout of IP/LP, HP/IP could be improved without reaming or face correction.

b. **ACTION CARRIED OUT IN JULY '92**

i) Roll check of IP turbine was repeated to optimise radial clearances.
ii) Coupled runout of rotors at coupling checked.
iii) Swing value of HP checked.
iv) Front bearing pedestal lifted by .4 mm.
Machine rerolled to 3000 rpm and loaded to full capacity and vibration level recorded were as follows.

	212 MW, 29/7/92; Shaft vibration	49.83HZ Bearing Pedestal Vibration		
0->PEAK		V	H	A
B1	138	36	18	36
B2	50	19	21	11
B3	75	11	27	22
B4	40	67	31	97
B5	-	23	26	99
B6	-	28	20	57

RESULT :-AXIAL VIBRATION CAME DOWN TO 57 MICRONS PK-PK FROM 131 MICRONS AFTER PEDESTAL BLUE MATCHING AT BEARING NO.6

CONCLUSION : ***PEDESTAL BLUE MATCHING IS MUST TO OBTAIN ADEQUATE BEDDING AREA. HIGH AXIAL VIBRATION CAN BE REDUCED BY ENSURING ADEQUATE PEDESTAL BLUE MATCHING.***

CASE 8 STATION SSTPS, 500MW UNIT VII

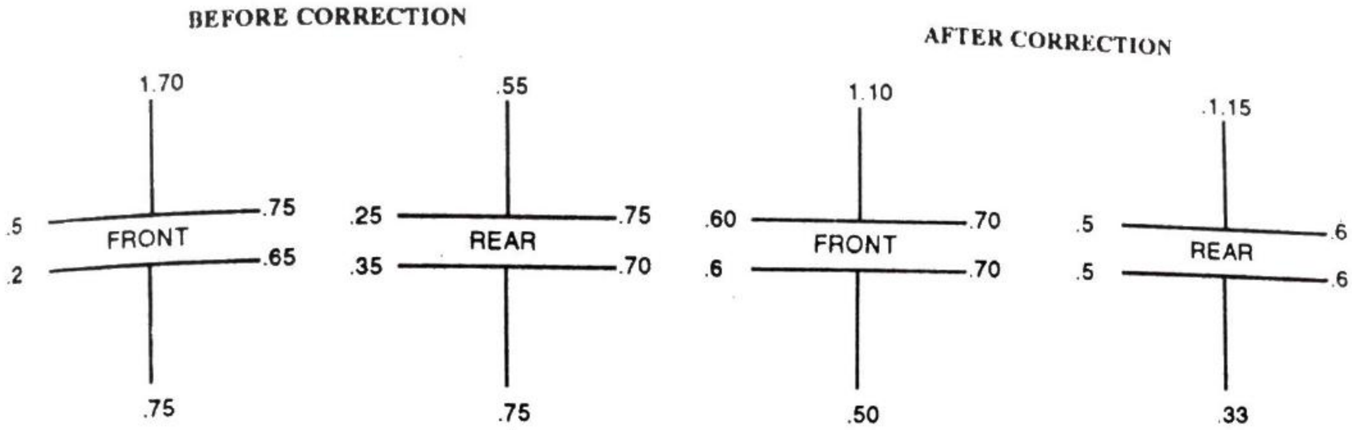
Maximum shaft vibration of the order of 140 microns was observed during commissioning of the unit. In fact shaft vibration was low at (49.35hz) i.e. 50 microns at 50 MW but it had tendency to rise with increase in load. The axial vibration at bearing no 4 also had similar tendency.

LOAD	SHAFT VIBRATION		BEARING VIBRATION (AXIAL)
80MW	B1	50	22
100MW	B2	50	12
200MW	B3	75	42
270MW	B4	140	110

The axial vibration of the order of 120 microns and 18 mm/sec peak was observed at 270 MW load (49.37) HZ) and also sparking due to gland rubbing near LP rear gland noticed.

ACTION CARRIED OUT IN JAN '88

i) LP gland seal clearance increased from .2/.3 to .5/.6 mm as shown below



RESULT AFTER ADJUSTING THE LP FRONT AND REAR GLAND CLEARANCES THE HPT FRONT AND REAR SHAFT VIBRATION REDUCED TO NORMAL I.E. 50 AND 12 MICRONS. THE AXIAL VIBRATION ALSO REDUCED TO NORMAL.

CONCLUSION : HPT FRONT AND REAR VIBRATION CAN BE AFFECTED BY INADEQUATE FRONT AND REAR GLAND CLEARANCES IN LP TURBINES.

2.1 PROBABLE CAUSES OF HIGH SHAFT/BEARING VIBRATION.

The high shaft/bearing vibration can be either due to operational mismatch of parameters or improper adjustment of critical clearances, during turbine erection and overhauling. The following are the probable causes of high shaft and bearing vibrations based on conclusions and analysis of cases experienced in NTPC & elsewhere.

PROBABLE CAUSES	REASONS
1. Gland seal rubbing	i) Uneven Seal Gap. ii) Thermal Distortion due to improper draining of gland steam in shut off conditions. iii) Low Gland Steam Temp.
2. Unbalance	i) High run out of rotor ii) Blade breakage. iii) Blade dislocation/droplet erosion of blade/balancing weight/erosion.
3. Improper Coupling Assembly	i) Facial Runout of Coupling ii) Misalignment in coupling iii) Improper Reaming / honing of Coupling Holes. iv) Improper Bolts Weights. v) Improper Coupling Bolt Tightness (differential coupling bolts tightness). vi) High Swing Value resulting from coupling faces.
4.a HP/IP Improper Assembly	i) Rotor to casing clearances ii) Horn drop test of HP-IP iii) Concentricity of rotor w.r.t. to casing iv) Assembly of keys/packers with proper running clearance. v) Improper concentricity between casing & pipe at steam admission & exhaust.
4.b LP improper assembly	i) Improper checks of LP assembly inner casing during erection/overhauls.
5. Improper bearing Adjustment	i) Torus mismatching ii) Improper seating iii) Excessive Lifting of pedestal iv) Oil film disturbance. v) Indequate Bearing loading. vi) Oil viscosity/temperature
6. Catenary disturbance	i) Improper slope ii) Improper reaming/honing iii) Improper coupling matching iv) High eccentricity and unbalance in other planes

3.0 SHAFT AND BEARING VIBRATION :

The vibration measured on a turbine is either shaft vibration or vibration excited by it (e.g. bearing vibration). There are two forms of shaft vibration :-

Forced vibration is caused by residual unbalance of the rotating masses. Additional temperature-sensitive unbalance can occur during non-steady-state operation. When passing through points of resonance it may cause large increases in amplitude. A disturbance to the balance, e.g. caused by a broken blade, will also give rise to increase in amplitude.

Self-excited vibration are the two types namely one caused by oil whirl/oil whip and other steam whirl. Generally self-excited vibration is sometimes called low frequency vibration because the vibration frequency is less than the rotational frequency. Load restrictions due to steam excited rotor vibration (steam whirl) have become a peculiar problem on some type of highly rated machines. ***The vibration usually occurs on the high pressure turbine shaft, with a frequency very close to one of the shaft's own natural resonances. Above some critical output level, known as the instability threshold, the amplitude can increase suddenly so that a rapid load reduction is necessary to protect the machine from damage.***

In some cases, the threshold can be shifted simply by adjustments to the internal clearances between the rotor and stationary parts of the machine, indicating that this is where the steam forces are active. On most existing of the machines, the solution is to raise the instability threshold to well above rated output in an ***ad-hoc fashion-either by redesigning or modifying the glands to reduce the steam forces, or by changing the bearings to increase the system damping.***

3.1 The Relationship Between Shaft and Bearing Vibration

The shaft vibration is transferred to the bearings through the oil film. This film possesses damping and spring properties which affect both shaft vibration and bearing vibration. The vibration is also transmitted further through various intermediate components of the bearings (spherical supports, keys, etc.) to the foundation. Bearing vibration is smaller than shaft vibration. The amplitude ratio of shaft to bearing vibration is influenced by many factors.

The relationship between shaft and vibration of the related bearing housing is dependent on :
* the ratio of oil film stiffness to the foundation spring constants.
* the ratio of the masses of the foundation and shaft.
* the ratio of the oil film damping to the elasticity of the shaft and the oil film.
* the relationship of the operating speed to the critical speed.
* the location of the measuring devices.

The above parameters are also affected by the particular design of the turbine and foundations. Depending on this, the relationship between shaft vibration and bearing vibration can be between **3:1 and 5:1 or in extreme cases 10:1.**

The measurement of shaft vibration normally has the advantage of greater sensitivity over the measurement of bearing vibration. Therefore, it is used to monitor the running characteristics and any change in them. However, shaft vibration values should not be used alone to make an exclusive assessment of the running qualities. Use must also be made of the assessment criteria for bearing vibration which have been well proven in the past.

3.2 Normal and Limit Values of Vibration & Internationally Followed Standards.

The running qualities of turbines can be assessed from the bearing and shaft vibration. The principal basis on which running qualities are assessed are two recommendations published by the VDI.
* VDI 2056 - Classification of Mechanical Vibration In Machines.
* VDI 2059 - Measuring Shaft Vibration For Monitoring Rotating Machines.

Similar recommendations have been prepared by various foreign institutions also such as ISO.10816/1 & 2, ISO 7919.

The recommendations contains definitions of terms which are absolutely essential for a clear understanding between operator and manufacturer.

A) **VDI 2056 Recommendation**

To assess bearing vibration :- VDI recommendation 2056 classifies the maximum root-mean-square value of vibration velocity according to speed, measuring location and measuring direction on the basis of previous measurements and experience in classes ranging from 'good' to 'impermissible'. The use of the classification recommendations covers vibration which can be measured in the spatial coordinates v and h (perpendicular to machine axis in vertical and horizontal direction) and a (along the machine axis). Torsional and bending vibration of the rotating parts is not taken into account.

The classification criteria produced by the VDI committee are based on the well-known Rathbon curves. There are two classification charts according to the type of the turbine generator foundations.

Radial Vibration

* **Turbines on high-tuned rigid and heavy foundations fig. no. 1**
* **Turbines on low tuned light concrete or steel foundations fig. no. 2**

The design of the turbine foundations is such that assessment should be according to chart **fig. no. 2**. The maximum root-mean square velocity of bearing vibration in the turbine during steady-state operation should lie on the line of division between the good and serviceable ranges.

B) **VDI 2059 Recommendations**

VDI 2059 deals recommendations with the use of **shaft vibration measurement** for assessing the running qualities of rotating machines. It also explains the difficulties in laying down exact classification ranges and limits as given in the recommendation 2056.

An Extract from VDI 2059

For large steam turbines it is possible to state, with certain reservations, the following limit values which, if exceeded, will produce a high probability of bearing damage due to dynamic overloading. The following are the limiting values:

Upto 70 microns 0->peak	-	Normal Continuous operation.
Upto 120 microns 0->peak	-	Continuous operation under close watch
Upto 180 microns 0->peak	-	From several hrs. to a few days, i.e. a normal length of time required to investigate and clarify abnormal vibration characteristics.
Upto 360 microns 0->peak	-	from short periods, e.g. when passing through critical speeds in the presence of self excited vibration. 360 microns 0->peak is the maximum value for initiation of automatic trip. It is the manufacturer's responsibility to state this value. In case of KWU Machines it is 200 microns 0->Peak only. The limit value apply to the principal semi - axis of the orbital curve.

3.3 **Assessment of Vibration and Corrective Action When Values are Excessive.**

Apart from the general assessment of the turbine running qualities using the accompanying charts Fig. No. 1 and 2, valuable information about the condition of the turbine can be obtained from a comparison of the actual running characteristics with the normal characteristics observed under identical operating conditions.

Sudden increases in vibration are usually a result of damage or changes which endanger the operational security of the turbine. For example, if the limit values are exceeded there is a danger that the dynamic load on bearing will become excessive or the radial clearances between rotor and casing will be exceeded.

It is therefore not advisable, to operate the turbine at inadmissible vibration values because there is a danger of compounding any such damages.

When the running qualities change within the order of magnitude of the permissible values, the increases in vibration at one or more rotors and bearing housings must be assessed according to whether they are increases above previously noted values or whether they are higher values which occur regularly but temporarily under similar operating conditions.

In the first case it must be assumed to be a change which can endanger the operational security of the turbine. Hence its cause must be determined immediately and rectified if possible.
In the second case it may be increases in vibration occurring during non steady-state operating conditions. If the permissible vibration values are not exceeded, these temporarily increased values usually have no detrimental effects.

When taking a decision on the necessary corrective measures when increased vibration occurs, the first question to ask concerning safety is the extent to which the turbine is endangered.

3.4 a. **Immediate Shutdown of the Turbine is Recommended if:**

* Limit values stated in the above instructions are exceeded, particularly if similar occurrences under comparable operating conditions have not been observed previously and there is a considerable increase in vibration above normal values.

* Loud noises are heard and the limits of other operating parameters such as wall temperature differential as a result of rapid temperature reductions, temperature differential between top and bottom halves of the turbine casings, expansion and bearing temperatures are exceeded.

3.4 b. **Sudden Increase in Vibration Within the Permissible Limits.**

* If it is definitely discovered that the increased vibration is of low frequency (vibration frequency < rotational frequency). it is usually sufficient to reduce load on the machine until the vibration dies away.

* Increase in vibration above normal values, but not exceeding the limit values, can also be assumed to be due to physical changes but these usually allow tests to be carried out to determine the cause of the condition. Such tests involve the determination of the vibration characteristics and intensity by means of measuring equipment independent of the operational measuring devices wherever possible. *It is advisable to measure;- Frequencies, amplitude intensity (H.V.A.) and phase angle and its change. The diagram in Fig no 3 gives general summary from various sources of well known possible causes of rough running of turbines.*

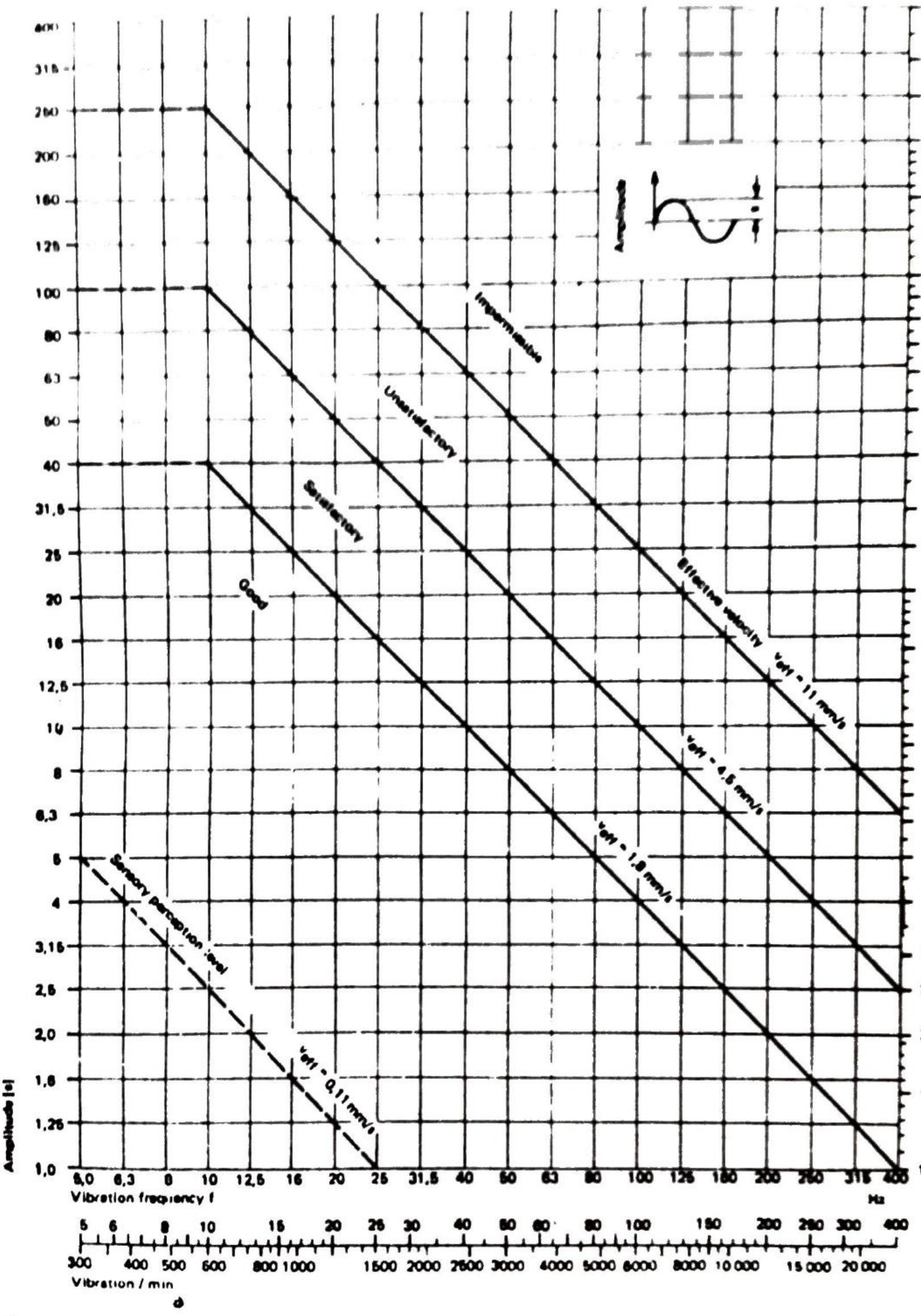


Fig. 1 Limiting vibration characteristics for turbosets mounted on high tuned heavy foundation

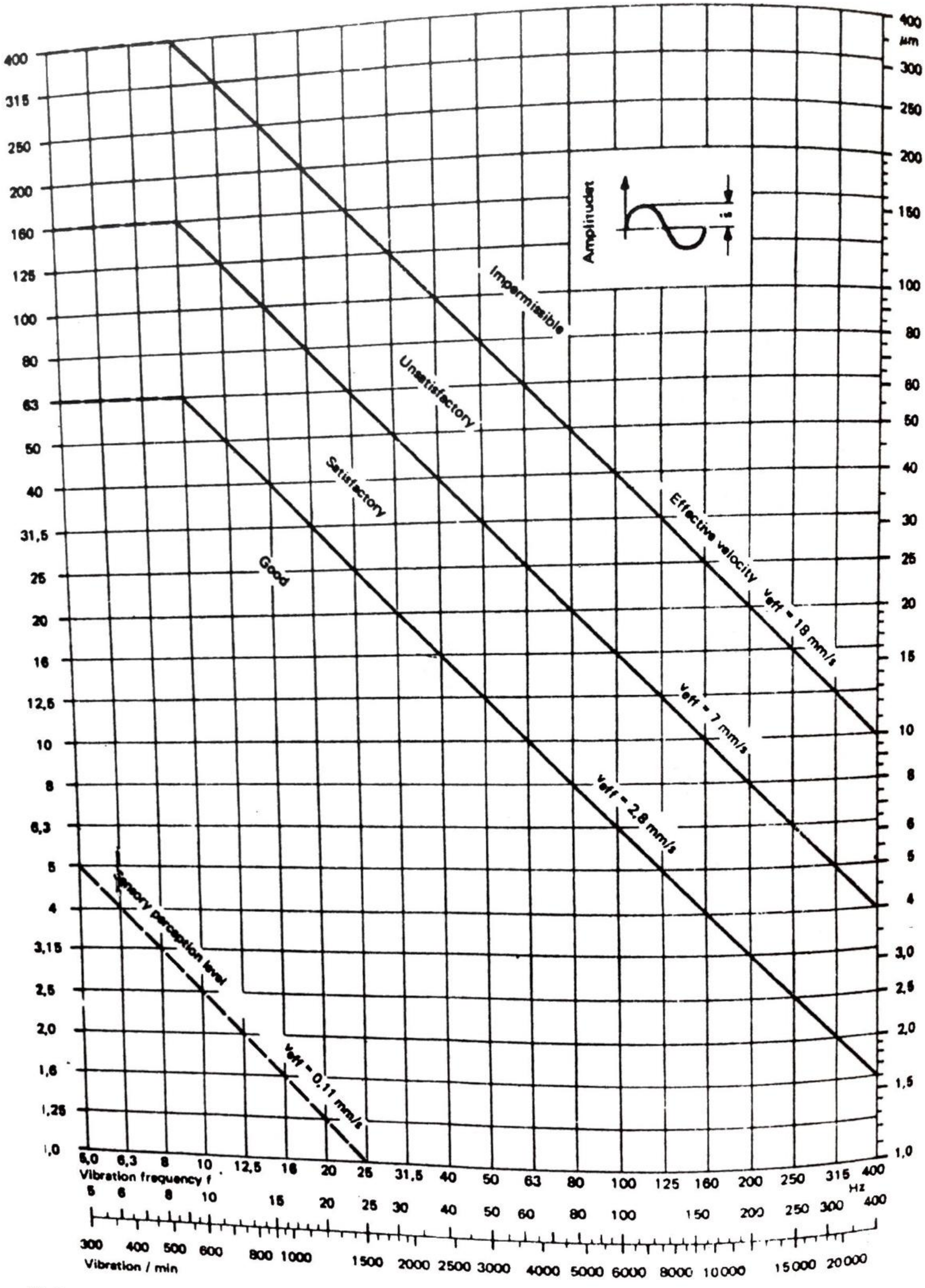


Fig. 2 Limiting vibration characteristics for turbosets mounted on low tuned light foundation

CAUSES OF EXCESSIVE VIBRATION

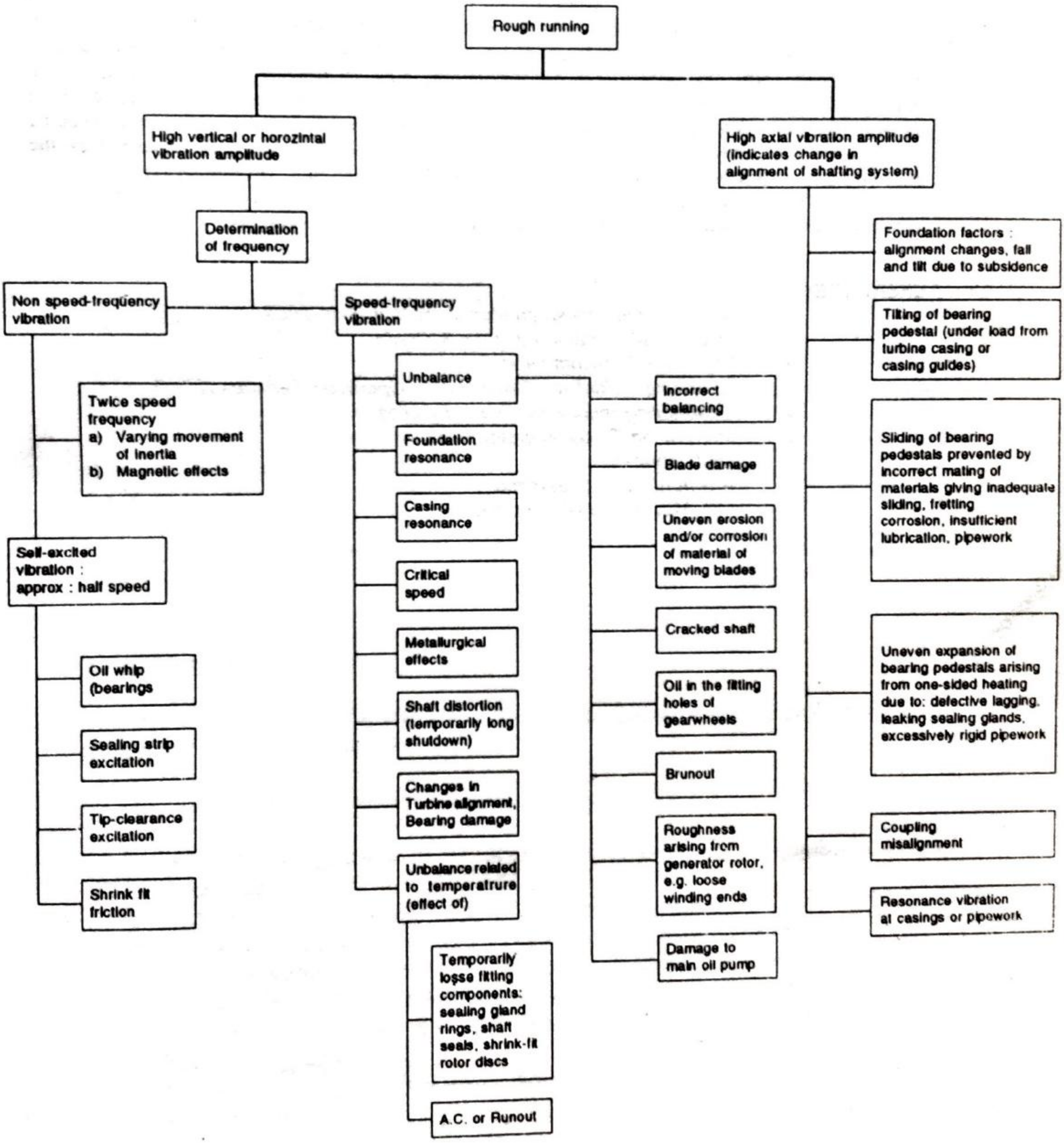


Fig. No. 3

4.0 **ACKNOLEDGEMENT** :-The authors are thankful to ***Shri Nirmal Kumar Jain Executive Director(OS), NTPC*** for encouragement during preparation of the paper and management of NTPC for permission to present this paper.

5.0 CONCLUSIONS:-

The findings of various Case-studies lead authors to conclude that satisfactory operation of turbines requires, strict adherance to maintenance and design protocols. But in some cases, even after maintaining these protocols , HPT shaft vibration is still maintaining high. In order to resolve this, authors had experienced that the unbalance in far away planes such as LP-Gen & inadequate gland clearances in turbines can also affect at High pressure Turbines shaft vibration. The application of condition based maintenance has proven practically that decision for deviation of overhaul as prescribed by the manufactures can be taken without sacrificing the intended duties and care. The turbine overhauling has been deferred beyond manufacturer recommendation in no. of cases by adopting the above technique. The results obtained by application of condition based maintenance are encouraging.

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