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VIBRATION ANALYSIS AND DIAGNOSTIC TECHNIQUES, with reference to the implementation of a vibration condition monitoring programme.

P. P. DUCCI

A dissertation submitted to the Faculty of Engineering, University of the Witwatersrand, Johannesburg in fulfilment of the requirements for the Degree of Master of Science in Engineering.

JOHANNESBURG 1987

Volume 1

DECLARATION

I hereby certify that this dissertation is my own work and has not been submitted for a Masters degree to any other University.

P.P. DUCCI

ABSTRACT

This dissertation deals with vibration analysis and detailed diagnostic techniques used for a vibration condition monitoring programme. Various vibration monitoring and diagnostic techniques were examined so that recommendations for the implementation of a vibration condition monitoring programme in ESCOM could be made.

Various vibration related problems pertinent to turbo-generators, feed water pumps and draught group fans are discussed. These include sub-synchronous oil instability, oil whip, unbalance, misalignment, cracked rotors, induction rotor bar defects, gear defects, rub and fan duct vibration. Relevant vibration diagnostic techniques are examined so that the abovementioned defects can be detected.

The effectivenesses of various statistical and computer predicted methods used to ascertain the condition of roller and ball bearings, and shaft critical speeds respectively were determined.

In order to recommend a method whereby sensible vibration trending can be performed the effect of load variations on the vibration severity was determined for draught group fans and feed water pumps.

Recommendations concerning the requirements of periodic and continuous monitoring equipment, the method of monitoring and the method of trending are made. These recommendations account for all the pertinent failure mechanisms of draught group fans and feed water pumps. Recommendations concerning the future requirements of a vibration condition monitoring programme within ESCOM are made. These recommendations deal specifically with on-line continuous monitoring systems and the formatting of vibration data for diagnostic purposes.

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CONTENTS

		Page
DECLARATION		(i)
ABSTRACT		(ii)
ACKNOWLEDGEMENTS		(iii)
CONTENTS		(iv)
LIST OF NOTATION		
1.	INTRODUCTION	1
1.1	BACKGROUND	1
1.2	LITERATURE REVIEW	6
1.2.1	Vibration Condition Monitoring	6
1.2.2	Vibration Measurement Standards	10
1.2.3	Economic Aspects of a Vibration Condition Monitoring Programme	20
1.3	PURPOSE	23
2.	PRESENT STATUS OF VIBRATION CONDITION MONITORING IN ESCOM	24
2.1	MAINTENANCE CONCEPTS	24
2.1.1	Breakdown Maintenance	24
2.1.2	Preventive Maintenance	24
2.1.3	Condition Based Maintenance	25
2.2	ESCOM's PRESENT MAINTENANCE STRATEGY	27
2.2.1	Vibration Condition Monitoring	27
2.3	BENEFITS OF A VIBRATION CONDITION MONITORING PROGRAMME - SUMMARY	30

3. TRANSDUCER SELECTION 31

3.1 ACCELEROMETERS 31
3.1.1 Advantages of Accelerometers 32
3.1.2 Disadvantages of Accelerometers 32
3.2 VELOCITY TRANDDUCERS 33
3.2.1 Advantages of Velocity Transducers 34
3.2.2 Disadvantages of Velocity Transducers 34
3.3 DISPLACEMENT TRANSDUCERS OR PROXIMITY PROBES 35
3.3.1 Advantages of Proximity Probes 36
3.3.2 Disadvantages of Proximity Probes 36
3.4 SHAFT RIDERS 37
3.4.1 Advantages of Shaft Riders 37
3.4.2 Disadvantages of Shaft Riders 37
3.5 DUAL PROBES 38
3.5.1 Advantages of Dual Probes 39
3.5.2 Disadvantages of Dual Probes 40

4. MODULE 1
DETECTION OF ROLLING ELEMENT BEARING
DAMAGE BY STATISTICAL VIBRATION ANALYSIS 40

4.1 INTRODUCTION AND THEORY 40
4.1.1 Literature Review of Rolling Element
Bearing Vibration*Health Monitoring 44
4.2 OBJECT 47
4.3 APPARATUS 48
4.4 EXPERIMENTAL PROCEDURE 51
4.4.1 Data Acquisition 51
4.4.2 Data Processing 53
4.5 OBSERVATIONS AND RESULTS 54
4.6 DISCUSSION 57
4.7 CONCLUSIONS 59

5. MODULE 2
FAULT DIAGNOSIS OF GEARS USING SPECTRUM ANALYSIS 60

5.1 INTRODUCTION AND THEORY 60
5.1.1 Literature Review of Gearbox Vibration Health Monitoring 66
5.2 OBJECTIVES 68
5.3 APPARATUS 69
5.4 EXPERIMENTAL PROCEDURE 70
5.4.1 Data Acquisition 70
5.4.2 Data Processing 70
5.5 OBSERVATIONS AND RESULTS 71
5.6 DISCUSSION 75
5.7 CONCLUSIONS 76

6. MODULE 3
DISCRETE FREQUENCY ANALYSIS 78

6.1 INTRODUCTION 78
6.2 VIBRATION DUE TO UNBALANCE 78
6.3 VIBRATION DUE TO MISALIGNMENT 78
6.4 VIBRATION DUE TO MECHANICAL LOOSENESS 80
6.5 ROLLING ELEMENT FREQUENCIES 82
6.5.1 Illustrative Example 83
6.6 VANE AND BLADE PASSING FREQUENCIES 84
6.7 THE DETECTION OF ROTOR DEFECTS IN INDUCTION MOTORS 86
6.7.1 Introduction 86
6.7.2 Theory 89
6.7.2.1 Vibration 89
6.7.2.2 Current 89
6.7.2.3 Speed 90
6.7.3 Reasons for Rotor Bar Breakage 91
6.8 VIBRATION MONITORING CRITERIA FOR EARLY DETECTION OF TURBINE ROTOR CRACKS 95

	6.8.1	Introduction	95
	6.8.2	Vibration Monitoring Criteria	95
6.9	SHAFT ORBITAL ANALYSIS		100
	6.9.1	Introduction	100
	6.9.2	Shaft Orbital Analysis Using Steady State Data	102
	6.9.3	Shaft Orbital Analysis Using Transient Data	107
6.10	DIAGNOSIS OF SUB-SYNCHRONOUS ROTOR DYNAMIC INSTABILITIES USING SHAFT ORBITAL ANALYSIS		111
	6.10.1	Introduction	111
	6.10.2	Theory	111
	6.10.3	Symptoms of Oil Whirl and Oil Whip	119
	6.10.4	Diagnosis of Oil Whil and Whip	123
6.11	SHAFT CENTRE LINE POSITION ANALYSIS		126
	6.11.1	Introduction	126
	6.11.2	Measurement of Shaft Centre Line Position	126
6.12	CONCLUSIONS		132

7. MODULE 4

DETERMINATION OF SHAFT CRITICAL SPEEDS USING COMPUTER PREDICTIONS AND EXPERIMENTAL MEASUREMENTS 134

7.1	INTRODUCTION		134
7.2	THEORY		136
7.3	OBJECTIVES		142
7.4	APPARATUS		143
	7.4.1	Experimental Measurements	143
	7.4.2	Computer Predictions	144
	7.4.3	Details of Fan Impellers	144

7.5 EXPERIMENTAL TECHNIQUES 146
7.5.1 Data Acquisition 146
7.5.2 Data Analysis 147
7.5.3 Technique used for Computer Prediction 151
7.5.4 Data for Idealised PA Rotor System 152
7.5.5 Data for Idealised FD Rotor System 153
7.6 RESULTS 159
7.6.1 Experimental Results 159
7.6.2 Computer Predicted Results 170
7.6.3 Computer Predicted Mode Shapes for the PA and FD Fan Rotors 176
7.7 DISCUSSION 179
7.7.1 Experimental Identification of Critical Speeds 179
7.7.2 Computer Predicted Shaft Critical Speeds 181
7.8 CONCLUSIONS 184

8. MODULE 5
THE EFFECT OF VARIATIONS IN LOAD ON THE VIBRATION SEVERITY FOR DRAUGHT GROUP FANS, ELECTRIC AND STEAM WATER PUMPS 185

8.1 INTRODUCTION 185
8.2 OBJECTIVES 193
8.3 DESCRIPTION OF EQUIPMENT AND APPARATUS 193
8.3.1 Equipment used for the Draught Group Fan Tests 193
8.3.2 Equipment used for the Electric and Steam Feedwater Pump Tests 193
8.3.3 Details of Fans and Feed Pumps 194
8.4 EXPERIMENTAL PROCEDURE 196
8.4.1 Data Acquisition for the Draught Group Fans 196
8.4.2 Data Analysis for the Draught Group Fans 196

	8.4.3	Data Acquisition for the Electric and Steam Feedwater Pumps	198
	8.4.4	Data Analysis for the Electric and Steam Feedwater Pumps	202
8.5	RESULTS		203
	8.5.1	Draught Group Fans	203
	8.5.2	Electric and Steam Feedwater Pumps	203
8.6	DISCUSSION		223
	8.6.1	Operating Conditions of the Draught Group Fans and Feedwater Pumps	
	8.6.2	Trending of Vibration Data for the Draught Group Fans	225
	8.6.3	Trending of Vibration Data for Boiler Feed Pumps	226
	8.6.4	Failure Mechanisms Pertinent to Large Boiler Feedpumps	227
		8.6.4.1 Unbalance	227
		8.6.4.2 Worn Mechanical Seals	227
		8.6.4.3 Misalignment between Motor and Pump	229
		8.6.4.4 Bearing Pedestal Resonances	229
		8.6.4.5 Worn Impellers and Shafts	230
		8.6.4.6 The Electric Motor and Booster Pump	230
		8.6.4.7 Survey of Feed Pump Outages	231
		8.6.4.8 The Voith Gearbox	232
	8.6.5	Failure Mechanisms Pertinent to Draught Group Fans	232
		8.6.5.1 Impeller and Blade Failures	233
		8.6.5.2 Fan Induced Duct Vibration	234
	8.6.6	Load Contours for PA Fans	236
	8.6.7	Load Contours for the ID Fans	239
	8.6.8	Load Contours for the FD Fans	239
	8.6.9	Load Contours for the Steam Feed Pump	239
	8.6.10	Load Contours for the Electric Feed Pumps	240

8.7 GENERAL DISCUSSION 242

8.8 CONCLUSIONS 252

9. MODULE 6

OTHER IMPORTANT ASPECTS OF A VIBRATION CONDITION MONITORING PROGRAMME 254

9.1 TRAINING FOR VIBRATION CONDITIONS BASED MAINTENANCE 254

9.1.1 Selection of Personnel 254

9.1.2 Training 256

9.2 CONTINUOUS VERSUS PERIODIC MONITORING 257

9.2.1 Continuous Monitoring 258

9.2.2 Periodic Monitoring 259

9.2.3 Trending of Vibration Data 264

9.2.4 Overall Level Monitoring 265

9.2.4.1 Advantages of Overall Level Monitoring 266

9.2.4.2 Disadvantes of Overall Level Monitoring 267

9.2.5 Bandwidth RMS Trending 267

9.2.6 Setting Alarm and Trip Levels for Discrete Frequencies 270

9.2.7 Spectrum Cascade Trending 271

9.2.8 Phase Trending 274

9.3 MANUAL AND COMPUTERISED LOGGING 274

9.3.1 Manual Data Logging 274

9.3.2 Computerised Data Logging 276

9.4 CONSIDERATIONS CONCERNING THE PURCHASING OF PERIODIC VIBRATION MONITORING EQUIPMENT FOR RECENTLY COMMISSIONED AND FUTURE ESCOM POWER STATIONS 278

9.4.1 Introduction 278

9.4.2 Software 279

9.4.3 Software Requirements 280

9.4.4 Requirements Concerning the Supplier of the Periodic Vibration Monitoring Equipment 280

9.4.5 System Requirements for Periodic Vibration Monitoring Equipment 281

9.5 CONSIDERATIONS CONCERNING THE PURCHASING OF CONTINUOUS ON-LINE VIBRATION MONITORING EQUIPMENT 284

9.5.1 Strip Chart Recorders 285

9.5.2 Dedicated Indicators 286

9.5.3 Computerised Real-Time Systems 287

9.5.4 Requirements of a Computerised Real-Time System 288

9.5.5 Requirements for all On-Line Continuous Systems 289

9.6 STANDARDISED RULES FOR MEASUREMENT ON ROTATING MACHINERY 290

10. DISCUSSION 301

10.1 WHY IMPLEMENT A VIBRATION CONDITION MONITORING PROGRAMME? 301

10.2 WHO WILL BE RESPONSIBLE FOR THE VCM PROGRAMME? 302

10.2.1 Size of Plant 303

10.2.2 Criticality of the Machine to the Production or Assemble Line 304

10.2.3 Number of Machines to be Included into the Programme 305

10.2.4 Sophistication of the Vibration Monitoring Equipment Purchased 305

10.2.5 Whether Continuous On-Line Monitoring or Periodic Monitoring has been chosen 306

10.2.6 Sophistication of the Actual Machinery to be Monitored 307

10.2.7 Extent to which Diagnostic Analysis is to be Performed 309
10.3 WHAT MACHINES ARE TO BE MONITORED? 315
10.4 HOW ARE THE MACHINES GOING TO BE MONITORED? 316
10.5 WHERE TO LOCATE THE VIBRATION MONITORING POINTS 317
10.6 TIME INTERVAL BETWEEN MEASUREMENTS 318
10.7 GENERAL DISCUSSION 319

11. CONCLUSIONS AND RECOMMENDATIONS 320
11.1 CONCLUSIONS 320
11.2 RECOMMENDATIONS 326
11.3 FUTURE NEEDS OF ESCOM REGARDING VIBRATION CONDITION MONITORING 329

12. REFERENCES 331

13. APPENDICES 337

LIST OF APPENDICES

1. Computor listing of Kurtosis Programme.

2. Derivation of the Computer Simulated Shaft Orbits.

3. Calculation of the Shaft Attitude and Shaft Movement Vector.

4. Interpretation of Bodé plots using a simple mathematical model.

5. The Prohl method for calculating critical speeds of flexible rotors.

6. The transfer matrix method for calculating critical speeds of flexible rotors.

7. Shaft critical sample calculations for FD rotor.

LIST OF NOTATION

SYMBOL	DEFINITION
RMS Velocity	Root Mean Square Velocity
F	Force
m	mass
$\ddot{x}$	acceleration
c	coefficient of damping
$\dot{x}$	velocity
k	stiffness
x	displacement
Hz	Frequency
P(x)	Probability density function
B_1	Skewness
B_2	Kurtosis
R	Gear Ratio
Tg	Number of gear teeth
Tp	Number of pinion teeth
Fp	Pinion rotation frequency
Fg	Gear rotation frequency
F_m	Mesh frequency
Fg	Hunting tooth frequency
F_a	Assembly phase passage frequency
V_i	Number of vanes on impeller
D	Pitch diameter
d	Ball diameter
n	Number of balls in a bearing
θ	angle of contact
R_s	Supply current
p	Number of poles
NDE	Non Drive-End
DE	Drive-End

SYMBOL	DEFINITION
g	Acceleration of gravity
H	Angular Momentum
Θ	Slope
L	Length
E	Youngs Modulus
I	Moment of inertia, moment of area

1. INTRODUCTION

1.1 BACKGROUND

Vibration analysis has been used to determine the mechanical condition of machinery for at least forty to fifty years. In the simplest case the plant engineer relied on his sense of touch and sound to determine the mechanical condition of slow and robust machinery.

World War I and II produced the need for increased reliability of weapons. A profusion of gauges and instruments was introduced during production to make it possible to measure accurately. The art of ergonomics started at that time. At the time of the Korean War the array of instruments became so confusing that comparators were introduced as a simplification. Operators no longer needed to measure absolute dimensions. Instead the terms "on-or-off", "high-or-low" were adequate.

Highly complex aero-space systems, guided missiles and early warning systems introduced during the mid 1950's increased the urgent need for high reliability and surveillance means of checking this reliability, while the number of components, assemblies, sub-systems and systems greatly increased. Simplified data comparators were replaced by high-speed electric multiple-point scanning systems capable of making rapid measurements and comparisons automatically. Results were processed and presented in a format which established the "signature" of the healthy machine for purposes of comparison at a later stage of deterioration.

Modern sophisticated monitoring systems have developed from these beginnings. They have their own self-integrity test systems by which their own fault detection or interrogation signals can be injected to verify the "health" of the monitor itself. It is only during the last decade that the full potential of these facilities for monitoring the "health" or "condition" of a machine has been properly appreciated. With the growing complexity of machinery, the increasing operational stresses, the lowering of safety thresholds and the decline in craft skills, these monitoring techniques have become essential.

A distinction between condition monitoring and vibration condition monitoring has to be made. Condition monitoring embraces all forms of parametric monitoring such as pressure, temperature, flowrate, vibration etc. Vibration condition monitoring deals specifically with the monitoring of a machine's vibration characteristics. This report deals specifically with vibration condition monitoring and the various techniques used to diagnose vibration related problems. It is impossible to monitor the mechanical integrity of a machine conclusively by using only vibration condition monitoring. It is generally believed that vibration monitoring is the most informative as far as the mechanical condition of the machine is concerned.

Vibration condition monitoring has developed from a skill into a science. The days of plant attendants with oily rags and grease guns tending their machines are gone. The intimate relationship between man and machine is no longer economically feasible and not even required because machines are expected to run automatically with only occasional attention from maintenance personnel. Modern machines are

getting bigger and are running at higher speeds then ever before. Vibrations from these machines occur at such high frequencies that sophisticated instruments are needed to detect and measure them.

ESCOM used to pursue a policy of constructing power stations consisting of numerous units of small power capacity. Examples are Vaal Power Station, 18 units of 15 MW each and Vierfontein Power Station, 19 units of 17,6 MW each. Nowadays large power stations like Koeberg and Duvha, consisting of 2 units of 965 MW each and 6 units of 600 MW each respectively have emerged. The advantage of the former policy is that if a unit is unexpectedly lost it would not significantly influence the national supply grid. The disadvantage of such a system is that it is uneconomical due to the construction of a large number of small capacity units. On the other hand the latter policy is economical in terms of construction, but if a large 600 MW or 965 MW unit is unexpectedly lost, it will have a significant influence on the national supply grid. The replacement costs for such a unit are substantial especially if gas-turbine plant and pump storage capacity is used to meet the demand for electricity. It is obvious therefore, that the availability of these large units must be high to keep replacement costs to a minimum. One of the best methods to improve availability is to implement a condition monitoring programme.

Power stations like Tutuka will be used as peak load stations. They will only contribute to the national grid during peak demand for electricity. It is therefore vitally important that when a power station like this is requested to contribute to the national grid it can do so without any delays. It´s availability during peak demand has to be high, and this can be achieved by a condition based maintenance programme.

Sadler (1) uses the example of British Steel Corporation (BSC) to illustrate the economics of increased availability. The cost of maintenance in one of the B.S.C's large integrated steel works in 1979/80 was £90 million and the numbers employed in maintenance, including staff, was 7400. Maintenance accounted for some 15% to 30% of the total conversion cost in a particular process. It has been quoted by Sadler that in general a 4% increase in availability of B.S.C. plants would be worth about £45 000 million per year.

ESCOM's new generation power stations have turbo-alternator sets which are among the largest in the world. For example, Matimba rated at 665 MW, Majuba rated at 657 MW already exist. Kendal rated at 665 MW will be the world's largest indirect dry cooled coal-fired power station once completed. It is of paramount importance that adequate vibration monitoring equipment be installed on these large sets to minimise the possibility of catastrophic failure.

In practice all generating units are constrained by compulsory statutory inspections of the boiler and reactor. Even with this constraint it is possible to minimise the outage time. This can be achieved by correcting faults on machines only as indicated by the vibration characteristics and not by routinely dismantling and checking for faults. This unnecessary "opening up to inspect" often does more harm than good as when ladders are left in boilers, spanners are left in sumps, or parts are incorrectly re-assembled. Often this approach is referred to as disturbance maintenance. The life cycle of a machine, follows the well known bath-tub curve, where the x-axis represents time and the y-axis, the probability of failure. This is illustrated in Figure 1.1. When a machine comes off the production line (Time 0) there

is a high probability that the machine will not survive without some defect or eventually die of old age. Some machines will fail very early in life for reasons such as machined components being outside tolerance and not having been rejected at the inspection stages. The initial curving section completes what may be termed the running-in period. When a machine is "run-in", there is a long flat region lasting for many months or even years when it can be expected to continue giving satisfactory service. Eventually, components will begin to wear out and the likelihood of failure increases with time. The "wearing-out" period forms the "foot of the bath". The effect of "opening up to inspect" on a machine which does not need attention is to build teeth into the flat area of the bath-tup curve. These teeth will follow the human error portion at the bottom of the running-in-period.

This report discusses various techniques used to acquire and analyse vibration data and the methods used to diagnose vibration related problems. The effectiveness of these techniques are also discussed. The main body of this report, therefore consists of individual modules most with their own introduction, theory, literature survey, discussion etc. The last chapter dealing with the recommendations concerning the implementation of a vibration condition monitoring programme, incorporates all the relevant conclusions and recommendations from all the modules.

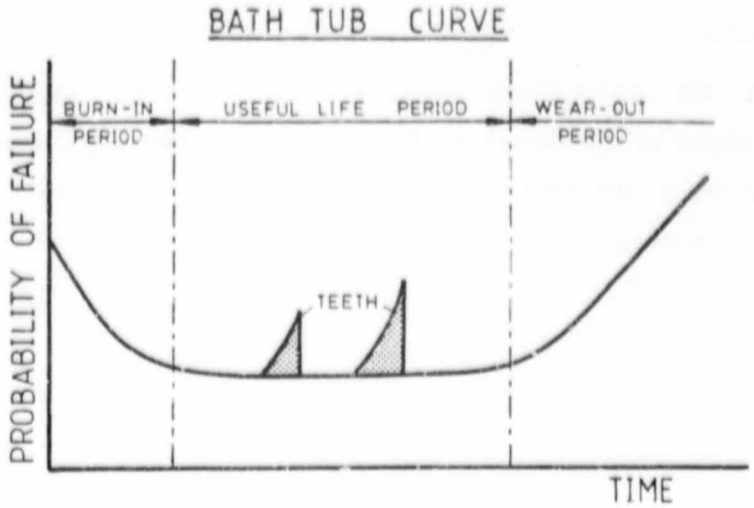


FIGURE 1.1

1.2 LITERATURE REVIEW

1.2.1 Vibration Condition Monitoring

Vibration condition monitoring allows the better-than-average machine to continue giving its satisfactory performance whilst the maintenance effort is concentrated on the worse-than-average machine which merits extra attention.

Low (2) cites three main advantages of condition monitoring in the Central Electricity Generating Board (CEGB) of the U.K., viz:

1) To keep plant running by indicating to the operator the health of the machine and the action required to maintain this.

2) To indicate areas of deterioration and anticipate necessary maintenance activities.

3) To provide knowledge of fault conditions and enable the machine to be operated, where possible, until the most convenient outage.

Low also states that a rational approach to condition monitoring can only be achieved by identifying the most expensive categories of plant failure and their failure modes.

Corben (3) describes vibration monitoring as a maintenance technique which permits a deterioration in machine health to be detected at an early stage without the machine having to be stripped down or even taken out of service.

Kerfoot, R.E. et al (4) suggests a three-tiered approach to a vibration condition monitoring programme:-

LEVEL 1

A complete listing of all possible frequencies at which vibration could exist should be compiled and kept with the machine's characteristics. Such a listing should include information like:-

a) running speeds
b) types of bearings
c) number of elements and size
d) gear mesh frequencies
e) speeds of accessary components
f) information about structural resonances or critical frequencies.

LEVEL 2

The spectrum of a signal from a vibration transducer produced by a running machine can be used as the "signature" or "fingerprint" of that particular machine's operational condition. Gradual changes in the machine condition can be seen as changes in the signature.

LEVEL 3 : The missing link

The major problem in interpreting data, which is taken at different times on the same machine, is that each of the frequency spectra must somehow be normalised to some typical operating condition so that they may be directly compared. In order to give the best insight into how to correct the

vibrations caused by rotating machinery, the transfer functions relating structural response to known inputs must be determined.

Downham and Woods (5) remark that it is important to obtain some knowledge of the response of a machine to particular dynamic forces. This knowledge can be most conveniently obtained, for bearing housing measurements, from a measurement of bearing house impedances at the monitoring points. This would enable the level of dynamic force to be computed directly from a measurement of vibration level under running conditions and, thus, help considerably in qualifying the relative severity of machine vibrations.

According to Battlebury (6) the first step in reducing the length of maintenance outages is for the maintenance and planning engineers to be fully familiar with the current state of the plant. In practice the plant history system is often inadequate. All too often plant is removed from the system without its current state being known, because the relevant maintenance documentation says that machines must be removed from the system for routine maintenance. To build a plant state data bank requires close dialogue between the plant and planning engineer.

In a report published by Michael Neale and Associates (7), the potential of condition based maintenance for certain industries is discussed. Figure 1.2 is obtained by comparing added value output per establishment with capital invested in plant and machinery per employee. A high value for both these factors indicates a favourable application for condition monitoring. Notice how favourable "Electricity" is on the guide.

GUIDE TO THE SELECTION OF SUITABLE INDUSTRIES IN WHICH CONDITION BASED MAINTENACE HAS POTENTIAL.

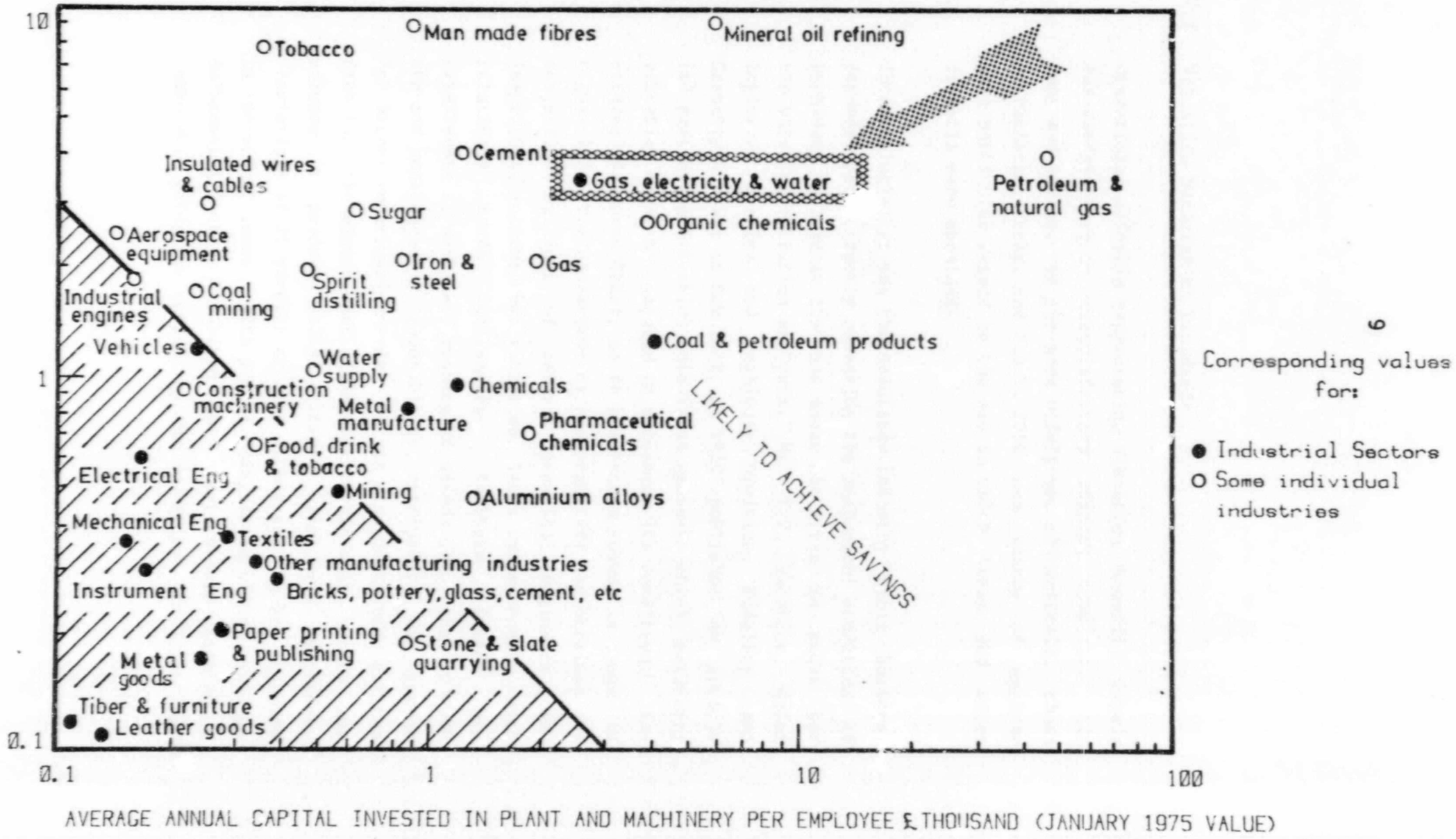


FIGURE 1.2

1.2.2 Vibration Measurement Standards

Operational criteria representing vibration boundary levels for satisfactory or unsatisfactory running conditions are many and varied. Of the more widely quoted criteria, those of Rathbone, Yates, and V.D.I. 2056 are worthy of mention with particular regard to the way in which these and other criteria were obtained.

Interestingly, it was the insurance industry whose business depends upon correctly assessing the mechanical condition of machinery it insures that was among the first to point out the value of vibration analysis. Mr. T.C. Rathbone, Chief Engineer, Turbine and Machinery Division, Fidelity and Casualty Company of New York, in 1939 published an article (8) presenting vibration guidelines against which machinery vibration could be compared to determine its condition. The original Rathbone Chart, as it has become known, is shown in Figure 1.3. The assessment of acceptability was obtained by subjective opinions of several practical engineers and inspectors, followed by vibration level measurements with relatively crude instruments. Rathbone limited his assessments to machinery running at speeds less than 6 000 RPM and considered it reasonable to extrapolate his curves for higher rotational speeds. He was also conscious of the fact that the measurement of bearing levels only, would not necessarily produce valid criteria for all machines. "Variations of 25 percent or more either way can be expected in individual cases. With proper recognition of the various influencing factors described the chart should furnish a reasonable guide for estimating the severity of vibration" (8).

THE ORIGINAL RATHBONE CHART

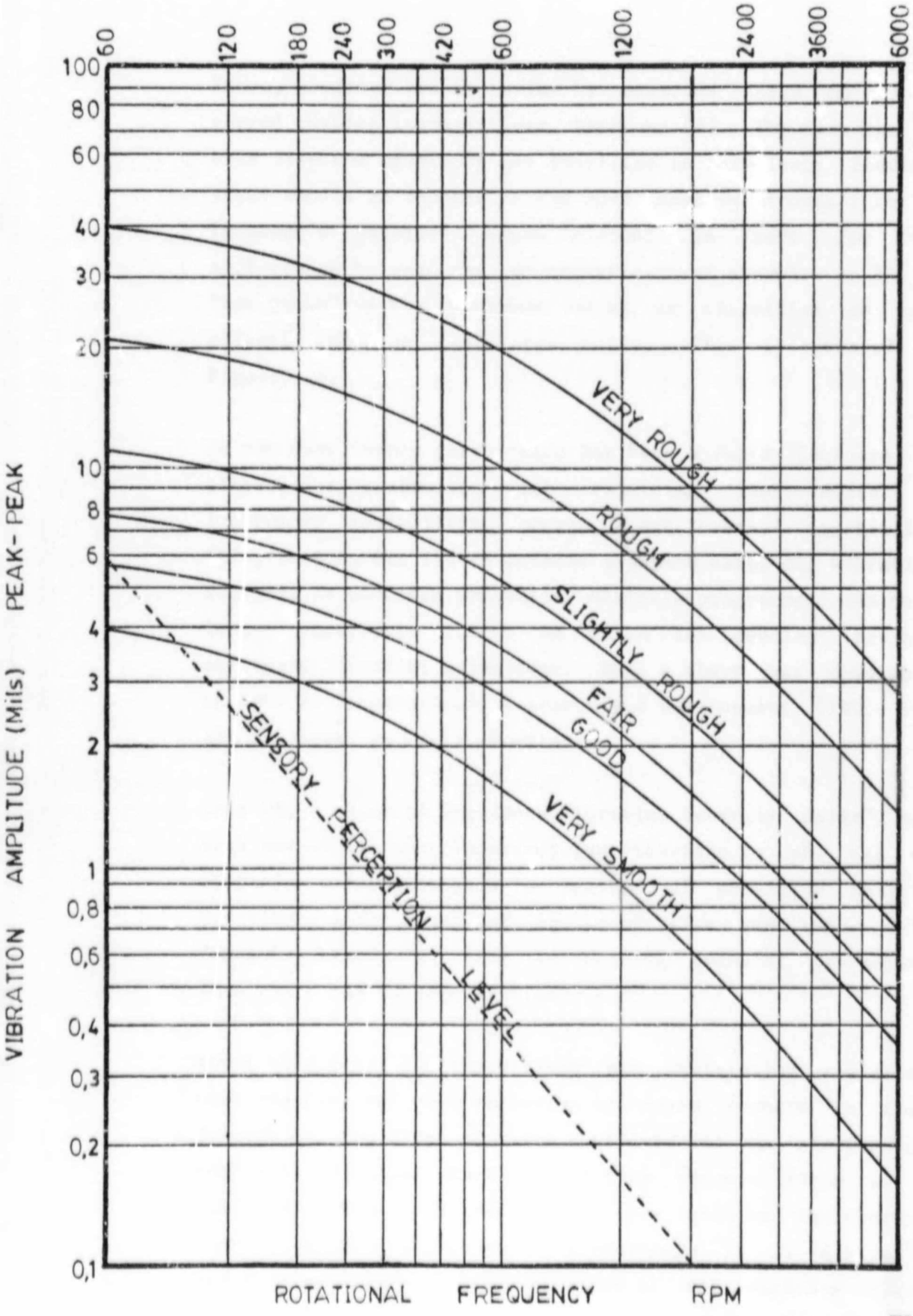


FIGURE 1.3

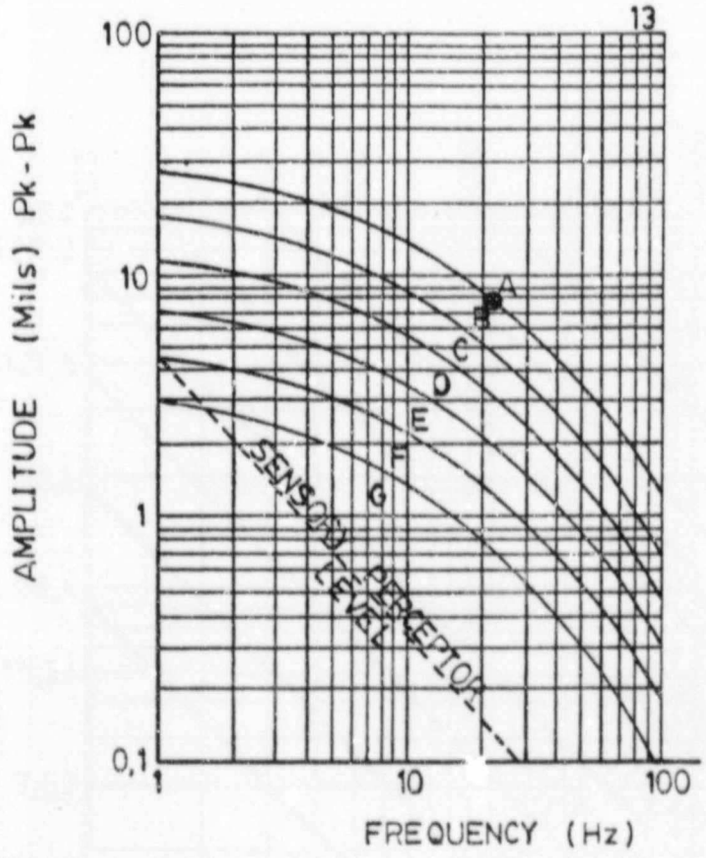
Yates, produced his criterion by numerous tests on marine geared turbine installations, Rathbone (8). The fact that in this case the machinery was installed in relatively flexible steel shells as opposed to the more massive foundations of Rathbone's machines, would account in part for the differences between the two criteria. For example, at 22 Hz "too rough" on the Rathbone curve is classified as only slightly rough on the Yates curve. This illustrated in Figure 1.4.

In the past twenty years there has been a rapid increase in electronics technology with resulting improvements in instrument portability, accuracy and overall capability. This, in turn has led to greatly expanded usage of vibration analysis in industry preventive analysis programmes, and much larger experience factor on which to develop revised machinery vibration guidelines. Such a chart was developed in 1964 by International Research and Development (IRD) (9) and was based on users experience.

This chart "General Machinery Vibration Severity Guide" has been one of the most important contributions to the use of vibration measurements as a monitor of machinery health, covering a wide range of equipment size and type. All foregoing standards refer to bearing pedestal vibration. This chart can be seen in Figure 1.5.

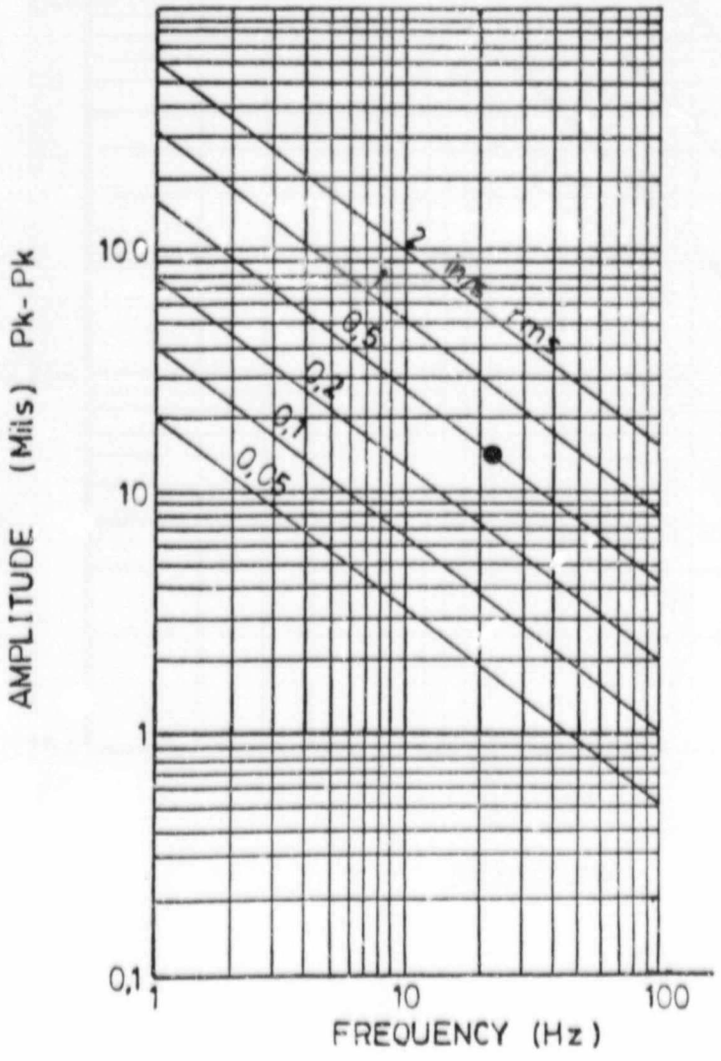
About this time, two troublesome and conflicting realities were emerging and were beginning to cause concern to this discipline. The first of these conflicts was the realisation that velocity alone does not define dynamic force as a function of time. The second was the growing realisation that in many classes of rotating machinery, it is desirable and even necessary to measure motion of the rotating shaft itself. For example, machines with high ratios of case/rotor mass and stiffness.

RATHBONE & YATES VIBRATION SEVERITY CHARTS



RATHBONE

REGION	CONDITION
A	TOO ROUGH TO OPERATE
B	ROUGH CORRECT IMMEDIATELY
C	ROUGH CORRECT
D	SLIGHTLY ROUGH
E	FAIR
F	GOOD
G	VERY SMOOTH



YATES

VELOCITY in/s rms	CONDITION
2	IMMEDIATE INVESTIGATIO REQUIRED
1	UNSATISFACTORY
0,5	SLIGHTLY ROUGH
0,2	NORMAL
0,1	VERY GOOD
0,05	SPINS LIKE A TOP

FIGURE 1.4

GENERAL MACHINERY VIBRATION SEVERITY CHART

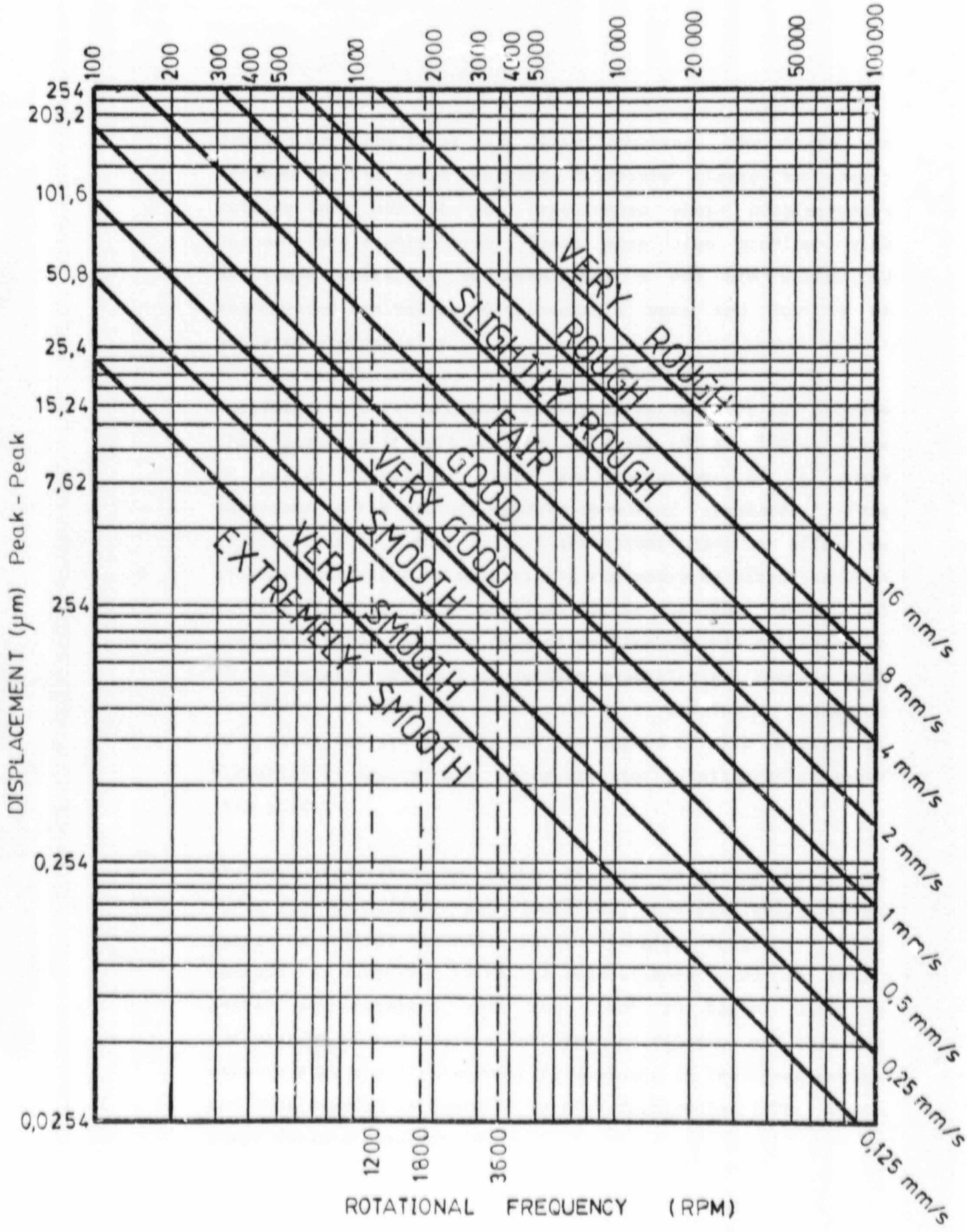


FIGURE 1.5

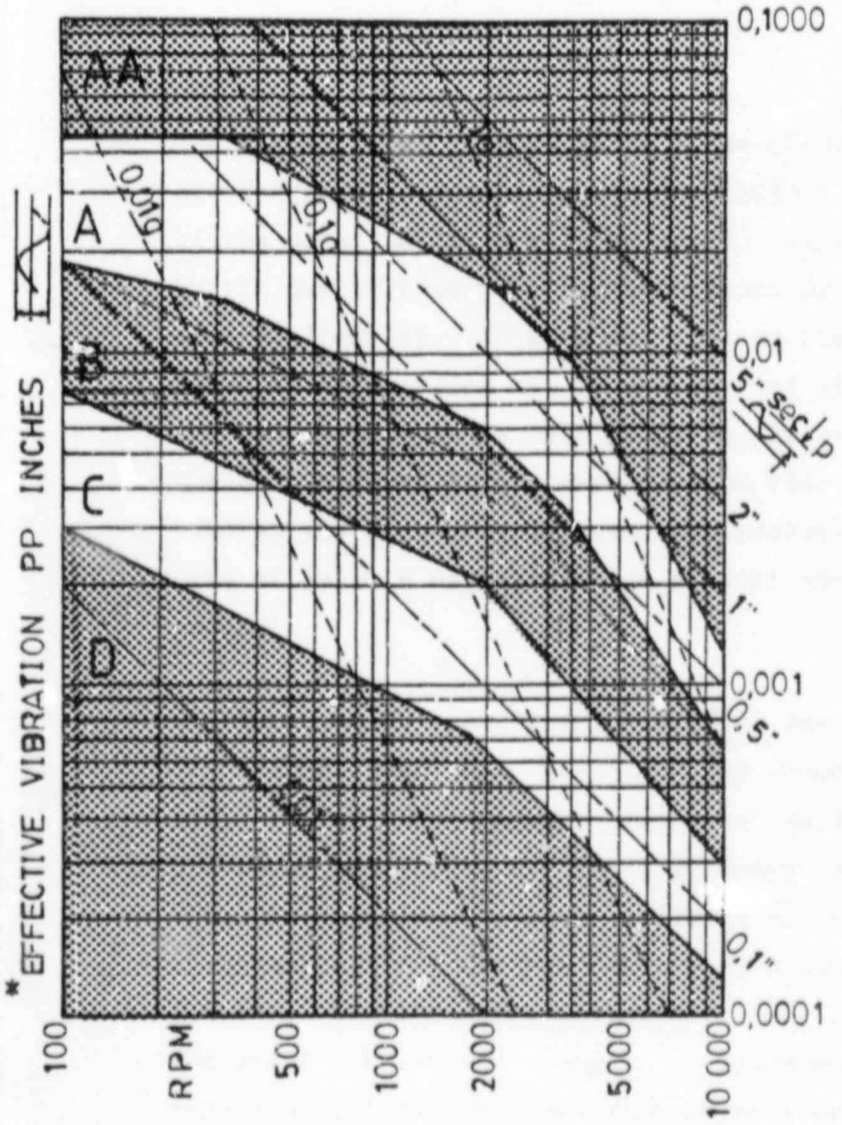
Dealing with the first of these concerns, the issue was avoided by the then current tolerance charts by simply cutting them off at low frequencies where displacements became intolerably high. There were also problems with machinery operating at much over 6 000 RPM where the recommended equivalent displacements were too low to be achieved economically.

Possibly the first and most widely accepted chart to address this situation in part was that of Blake (9) in 1964. This so called "haystack" chart and variations of it were important in the process of development of standards. This is illustred in Figure 1.6. Blake's chart uses an effective vibration obtained by multiplying the measured vibration by a service factor. The service factor is the rating of equipment based upon the type of machine and how critical it is to a plant's overall production capabilities. Factors such as safety, economy, the importance of a machine's operation to a plant and its profit, and the degree of risk involved if vibration is ignored are necessary to intelligently apply this guide.

Although described by Blake himself as being based on operating experience and intuition, nevertheless, these charts do reflect the basic rotor dynamics equation. This concept was discussed by Baxter and Bernhard (10) in 1967. Baxter and Bernhard were the first to begin with a description of the basic rotor dynamics equation and then use this to show why displacement is dominant at low frequencies and acceleration is dominant at high frequencies. The basic rotor dynamic equation is:

$$F = m\ddot{x} + c\dot{x} + kx$$

BLAKE'S EFFECTIVE VIBRATION CHART



AA	DANGEROUS
A	FAILURE IS NEAR
B	FAULTY
C	MINOR FAULTS
D	NO FAULTS

SERVICE FACTORS

SINGLE STAGE CENTRIFUGAL PUMP, ELECTRIC MOTOR, FAN	1
TYPICAL CHEMICAL PROCESSING EQUIPMENT, NONCRITICAL	1
TURBINE, TURBO-GENERATOR, CENTRIFUGAL COMPRESSOR	1,6
CENTRIFUGE STIFF-STAFT, MULTI-STAGE CENTRIFUGAL PUMP	2
MISCELLANEOUS EQUIPMENT, CHARACTERISTICS UNKNOWN	2
CENTRIFUGE, STAFT-SUSPENDED, ON STAFT NEAR BASKET	0,5
CENTRIFUGE, LINK-SUSPENDED, SLUNG	0,3

* EFFECTIVE VIBRATION = MEASURED PEAK TO PEAK VIBRATION, INCHES, MULTIPLIED BY THE SERVICE FACTOR

FIGURE 1.6

The way in which each of these terms dominate its own region of the frequency domain is shown in Figure 1.7 where the similarity of the haystack chart will be noted. However, all this tells us is the qualitative relative importance of displacement, velocity and acceleration. It does not tell us the speed or frequency at which to change from one to another nor should it be implied that any one phenomenon is sufficient in the region of its dominance. These things are highly dependent on the type of machine, its bearing and support stiffness, and probably most importantly, the type of failure or malfunction against which it is desired to protect.

The V.D.I. 2056 specification (11) is one of the most exhaustive in present use and was compiled from previously published criteria for five separate classes of machines. Group G machines for power generation machinery on heavy rigid foundations were taken from Rathbone's work, although his curves were approximated to straight lines and the maximum rotational speed extended to 12 000 RPM. The directive places no restriction on the type of vibration measured (e.g., displacement, velocity, or acceleration) but suggests that where several harmonic components are present an RMS measurement of the complex signal should be used for comparison with the given criteria.

If typical tolerance curves based on shaft measurement are examined, for example those of A.P.I. 670 (12) dating from 1968, it will be seen that these are not constant velocity but are tilted to give lower velocity at low speeds and allow higher velocity at higher speeds, which is also the general pattern for large turbine generators.

As machinery became larger, faster, and more complex with less backup equipment, the need has continued to grow for vibration analysis techniques which can detect incipient defects sooner and

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