N94- 34190

A REVIEW OF VIBRATION PROBLEMS IN POWER STATION BOILER FEED PUMPS

David France
Weir Pumps Limited
Glasgow, Scotland

S14-37
128558
p. 28

SYNOPSIS

Boiler Feed Pump reliability and availability is recognised as important to the overall efficiency of power generation. Vibration monitoring is often used as part of planned maintenance. This paper reviews a number of different types of boiler feed pump vibration problems describing some methods of solution in the process. It is hoped that this review may assist both designers and users faced with similar problems.

1.0 INTRODUCTION

The reliability of the boiler feed pump in modern power stations is seen as essential to ensure the overall availability of the power generating plant and is often included in the rotating machinery vibration monitoring programme practised by many power station maintenance departments. The vibration characteristics of a boiler feed pump can be complex and difficult to understand and it is the purpose of this paper to describe typical vibration mechanisms at work to assist those engaged in the diagnosis of problems. Case studies are presented which demonstrate the following types of vibration responses:-

1. Forced
2. Resonant
3. Self excited
4. Transient
5. Cavitation

Some solutions to these are described in the hope that these may assist both designers and users faced with similar problems.

2.0 FORCED VIBRATIONS

Forced vibrations - vibrations which are produced by the application of an external set of oscillatory forces.

Centrifugal pump vibrations are often described as being either synchronous or non-synchronous with respect to its operating speed. Usually synchronous is taken to mean the vibration component at the pump's operating speed although it is often taken to include any harmonic of this. Nonsynchronous covers all other frequency components. Synchronous forcing influences may arise from either mechanical or hydraulic sources of unbalance. Mechanical sources are fairly well understood because of the general similarity in other types of rotating equipment. Hydraulic sources may result from either casting asymmetries or partial blockage of impeller passageways by, for example, ingress of foreign matter. Generally speaking the magnitude of hydraulic unbalance generated by an impeller is greater than the residual mass unbalance component. A Japanese investigator, ref.1, measured the hydraulic force as a function of flowrate, fig.1. The maximum value of unbalance qualities of G2.5 - G6.3. Practical experience suggests that precision castings for the impellers of high speed feed pumps could offer a hydraulic reduction of possibly an order of magnitude. Nonsynchronous forced responses usually arise at part load operations where hydraulic forces, generated by recirculating flows within the impeller or diffuser, may give rise to irregular or cyclic low frequency excitations. These have been discussed previously by a number of authors, ref.2, 3. A typical example of this can be seen in fig.2, where the shaft displacement increased threefold between the design flowrate and the recirculation flow quantity.

Substantial forces may be generated due to interaction of the impeller blades with guide vanes in either the upstream or downstream flow fields. These forces manifest themselves as vibration at the impeller blade number and blade number harmonics. The main area of concern to any maintenance engineer is undoubtedly the effect that high blade vibration levels may have on the reliability of the pump. Understandably the worry is whether or not this may result in a fatigue failure of impeller or diffuser. General machinery vibration guidelines offer little help in this regard except to suggest that vibration monitoring may help to identify deterioration. Data from the field serve to illustrate the difficulties of interpretation further. For example, blade vibration levels of 15mm/sec rms are not untypical on some of our boiler feed pumps yet these pumps have given many years of trouble free operation. On the other hand, impeller blade or impeller shroud failures have occurred after some 6 months to 2 years into operation yet external levels of blade vibration have been less than 2mm/sec rms. Clearly, therefore, there is not necessarily a one - one relationship between vibration level and the stress levels within the working components. Insight into generation of vibration at blade frequency and blade frequency harmonics results from the studies by Zotov, ref.4. this theory shows that it is possible to select impeller and diffuser blade number combinations to suppress the vibration at the fundamental blade frequency, or if required, at a higher blade frequency harmonic. This is indicated by the values shown in Table I. The table shows the radial vibration force frequency components relative to a unit magnitude radial interaction force between each of the impeller and diffuser blades. It is significant that the vibration of the pump casing can be minimised in this way, but care is needed to ensure that certain blade combinations do not give rise to reinforced pressure fluctuations in the fluid as could be the case with 6 x 9, and 8 x 12 blade combinations. Experimental data obtained from pumps with 7 x 12 and 7 x 13 blade combinations show encouraging correlation with theory, these also are given inTable I. Although blade numbers can have a powerful effect on the distribution of the harmonic content of the resultant blade interaction forces, the stresses induced on the individual blades are not significantly modified by the number of blades between the impeller and diffuser. The magnitude of the interaction forces are influenced by the hydraulic design, the stage power, the impeller to diffuser gap, the operating point relative to the best efficiency point and possibly the inlet guide vane geometry and upstream flow field. It seems fairly well accepted that a minimum tip gap of 3% of impeller radius, (derived empirically by Makay, ref.5.), is sensible to avoid highly stressed components. Some results of research by Weir Pumps Limited into blade combinations and tip gaps give interesting insight into such effects, fig.3a, 3b.

In 1985 an unusual blade vibration phenomenon was identified on a number of boiler feed pumps destined for an East European power station. The pumps are 4 stage boiler feed driven by steam turbines. Pump design duty is $0.36m^3$/sec at 2461m head at 5454 rpm, impeller blade number is 7 and diffuser number is 12. Tip gap is 3%. This should have resulted in low blade rate vibration levels, yet high vibration levels were experienced on some of the pumps on the drive end bearing housing only. The vibration was only experienced over a narrow range of flow. A typical blade frequency vibration characteristic is shown in fig.4. Each pump showed similar characteristics but the vibration level on some of the pumps was only about 2.0 - 3.0mm/sec rms. Despite many hours of tests and subsequent examinations of components from a number of pumps, it was not possible to ascertain any difference which might explain the variation in vibration level. The only modification done was to eliminate the upstream inlet guide vanes to the first stage impeller, this had no effect on the magnitude of the vibration. The problem source has never properly been identified, yet the vibration levels would appear not to be detrimental to reliability as the pumps have been operating satisfactorily for more than six years.

3.0. RESONANT VIBRATIONS

Resonance - a phenomenon evident when the frequency of the disturbing force equals a natural frequency of the system.

In 1982 a resonance problem was experienced on a four turbine driven variable speed boiler feed pump, installed at Huntly Power Station, (4 x 250MW), New Zealand. The area is seismically active and the design of the pump and turbine support structure reflected the seismic requirements. The pump was centre line mounted on a baseplate which was secured to a large steel table onto which the turbine was also mounted. The steel table was supported from the floor level by a number of tall vertical metal columns, ensuring a low frequency mounting system for the equipment. A schematic is shown if fig.5. High vibration levels, in excess of 20mm/sec peak, at pump shaft frequency, were obtained at generator loading of 200-220MW, corresponding to a pump speed in the range 5300 rpm-5500rpm. Examination of the amplitude phase relationship suggested a resonance was responsible. Excitation tests were carried out with the pump stationary using an air driven rotary out of balance exciter to stimulate the structure into vibration and to examine the mode shape. These tests confirmed there was a natural frequency of the pump, but the mode shape was complex showing motion involving the pump and turbine support structure. Surprisingly, there was little or no deformation of the pump casing and pump supporting baseplate, the whole body appeared to be moving on the flexibility of the steel supports beneath the base plate. A comparison of the on load vibration characteristic compared with the vibration obtained from the air driven exciter is shown in fig.6a. Exciter tests were then carried out with one of the pumps and its baseplate secured to a concrete foundation. No evidence of any natural frequencies in the range 3000cmp - 6000cpm were found indicating that the steel support structure was important in terms of the mode of vibration of concern. A comparison of the responses on the steelwork and on the concrete floor are shown in fig.6b. The next stage of this investigation was to consider how detuning the resonant frequency could be achieved on site. Clearly a solid support for the pump bedplate would have solved the problem but it was impractical to stiffen the support steelwork sufficiently. It was decided to introduce compliance between the pump casing and the bedplate at the pump support feet. As designed the support feed slide relative to the pump baseplate on greased brass pads to accommodate thermal expansion. A modification was made as shown in fig.7, this introduced horizontal flexibility but still allowed the pump support feet to slide. On site a series of exciter experiments was carried out to optimise the flexibility of the detuning element. Once an optimum was found the pump was run up to full load to demonstrate the modification. The vibration levels were found to be a maximum of 5mm/sec peak. A comparison before and after modification is shown in fig.8.

4.0 SELF EXCITED VIBRATIONS

Self excitation - vibrations which are caused by disturbing forces which result from feedback from the motion of the vibrating object itself.

A review of self excited vibrations in centrifugal pumps has been presented in ref.6. This paper discusses a wide variety of types including the "near running speed" nonsynchronous vibrations reported by a number of authors. There have been only a few reported examples with boiler feed pumps, and then mainly outside the UK.

In the late sixties a 0.8 x running speed vibration problem was found on starting and standby boiler feed pumps at Longannet Power Station, Scotland. The high vibrations were seen over a range of operating speed from 3500 rpm - 6000 rpm as shown in fig.9. The pump was found to undergo an almost instantaneous progression from low level synchronous vibration to "foundation shaking" nonsynchronous vibration as pump speed was increased; typical of a classical self excited vibration phenomenon. The problem was related to premature wear out of the pumps internal clearances caused by intermittent low flow operation and consequential reduction in rotor stiffness. The problem was eliminated by increasing the minimum recirculating flow to eliminate the wear out problem; no further research was done at this time on what was believed to a "one off" phenomenon.

In 1978 serious problems were experienced during works testing of three pumps. Performance tests were conducted at full power and fixed speed and near running speed subsynchronous vibration was experienced. These pumps were 7 stage pumps absorbing 2230Kw at 5850 rpm. The problem was first noticed as flowrate was increased from the minimum recirculation quantity to the design flow. Shaft probes were fitted as standard and the transition from synchronous to nonsynchronous vibration was abrupt. It was also established that reducing flowrate could once again stabilise the pump. Shaft displacement levels were 75μm pk-pk at a frequency of 80Hz-84Hz giving a whirl frequency ratio of 0.82-0.86. A typical average spectrum is shown in fig.10. A 24 hour endurance test on the test bed confirmed the severity of the problem when balance drum leakage flowrates doubled and bearing housing vibrations increased fourfold. Initially it was assumed that the problem was a classical rotor instability, i.e. unstable eigenvalue. It was believed that negative stiffness and forward driving cross coupling could arise due to an impeller/diffuser interaction of the type described in (11). However, it seemed unlikely that the problem was due to an unstable eigenvalue as rotordynamic analysis predicted well damped eigenvalues even with generous allowances for impeller/diffuser interaction forces. Nevertheless, both aspects of the problem were examined, the former by a specially instrumented single stage test rig and the latter by conducting variable speed tests on one of the pumps with clearances set at design values (0.5mm) and then twice design (1mm). To assist the problem diagnosis a pair of displacement probes were installed to view the shaft motion at the balance drum as well as the bearings of the pump. The results of these tests are summarised in fig.11a and fig.11b. Tracking of the amplitude and phase of the synchronous component of shaft vibration over a speed range from 4000 rpm - 6500 rpm showed no significant phase or amplitude change which would indicate even a near proximity to a natural frequency. A similar result was obtained for both the design and twice design clearances build configurations. Rotordynamic calculations confirmed the shaft motion and phase measurements at the balance drum and bearings and absence of unstable eigenvalues. Of enormous significance was the fact that the subsynchronous component of vibration bore an almost fixed ratio with respect to speed (.85) for both clearance cases tested. The subsynchronous component was discernible down to the minimum speed tested. The test rig comprised a single impeller and diffuser stage and was instrumented with shaft motion probes and dynamic pressure transducers to examine distributions of pressure in the diffuser inlet, impeller shroud to casing gaps and near to the impeller eye. Tests were carried out to approximately 80% pump design speed. Sum and difference measurements of two pressures at 180° at impeller eye showed a nonsynchronous frequency component which was out of phase implying a net resultant force on the impeller. This correlated extremely well with the resultant shaft motion as shown in fig.12a and fig.12b and was very similar to that obtained from the pump, fig.10. Tests subsequently confirmed that this was an impeller inlet stall which appeared to rotate relative to the impeller at a frequency equal to the difference of the synchronous and subsynchronous frequencies. The forces involved were quite small and the feedback element which was evidently present with the 7 stage pump was never reproduced on the test rig. In the final event no design changes were made to the pumps, although an impeller with revised inlet geometry had been tested successfully in the single stage test rig. There were however, significant improvements made to the build concentricity of the pump cartridge and a change to the shaft material. Tests confirmed that subsynchronous vibrations were still present but a factor of ten below synchronous amplitude levels. No change in vibration was obtained over a 48 hour endurance test at full load. The pumps have been in service for over twelve years and have given trouble free operation.

During proving tests on the prototype of a new frame size of barrel casing pump in 1987, the opportunity was taken for an in-depth examination of the effects of annular clearance wear and the influence of inlet swirl at the balance drum upon rotor dynamics and, in particular, subsynchronous vibration. The device used to control inlet swirl was similar to that described in (7). The prototype was a six stage pump with radial diffusers and a balance drum with a design maximum running speed of 5000 rpm. The design parameters of this barrel casing pump are set out in Appendix I. The materials chosen for the renewable wearing rings were chosen for their high wear resistance and freedom from galling. To maintain optimum efficiency the recommended renewal clearances are set at 1.5 times the new design clearances. It was decided to study the effects of wear up to a maximum of twice the renewable limits. The tests were carried out in the production test facilities and the pump was installed on a temporary, but substantial, support structure. The prime mover gave a variable speed capability from 3600-5500 rpm, representing 10% above pump design speed. Instrumentation was fitted to the pump to measure, head, flow, speed, loop temperature, bearing metal and oil temperatures, axial thrust, balance return line flow. Dynamic measurements taken were bearing housing vibration, (seismic), shaft to casing radial displacement and pressure fluctuations in the suction, and discharge and balance return lines. The dynamic signals were taped recorded for subsequent analysis off line. During each test discrete points were taken over the performance envelope of the pump together with data from controlled runs varying one parameter at a time and keeping others constant, e.g. varying speed at constant discharge control valve setting. The tests carried out are summarised in Table II and the incidence of subsynchronous vibration during the tests is summarised in Table III. The pump was stable when clearances were within the recommended limits, but became unstable with 2 x design clearances. In contrast inlet flow swirl control delayed unstable behaviour until clearances approached twice the renewable values. When the subsynchronous vibration occurred it had a well defined character. Figure 13 shows a time history and a spectrum of the shaft motion taken during Test 3. The time history shows the typical beating resulting from the presence of two strong vibration signals with frequencies close together. The spectrum shows not only the running speed vibration and the subsynchronous vibration but also the sidebands associated with these frequencies. This spectrum with its strong difference frequency, (running speed - subsynchronous speed), and the similarly displaced sidebands is typical of this phenomenon.

The vibration locked in and dropped out as the speed was varied. On fig.14 a waterfall plot of the shaft vibration on a run up highlights this characteristic. The waterfall was taken by varying the speed against a constant throttle valve setting, approximately equivalent to the best efficiency point at any speed. At this setting the subsynchronous vibration drops out at 3600 rpm. On fig.15 a waterfall plot of shaft vibration on a run down shows that the subsynchronous vibration dropped out at 3600 rpm. The waterfall plots also show that the subsynchronous vibration did not occur at a fixed frequency but varied as a constant proportion of running speed. In Test 3 the subsynchronous whirl frequency ratio 0.84 irrespective of the actual running speed of the pump. This is particularly well illustrated on the run down characteristic. At speeds, where the subsynchronous vibration was well established, the variation of flow had little effect on the character of the vibration. However, at speeds around the stability boundary of the subsynchronous vibration a change of flow caused the phenomenon to 'lock in' or ';drop out'. In general reducing the flow brought on the onset subsynchronous vibration. The effect of flow on the subsynchronous was noticeably less powerful than the effect of speed. Throttling the balance return line to increase the back pressure on the balance drum also delayed the onset of the subsynchronous vibration. Where the subsynchronous motion was well established the shaft vibration appeared to take up virtually the full bearing clearance. Not unsurprisingly this imposed an upper bound to the amplitude of the vibration. The general character of the bearing housing vibration was the same as the character of the shaft vibration. The vibration levels in the frequency range 10-200Hz, were of order 2mm/sec when running normally and up to 7mm/sec when the subsynchronous vibration was present. The vibration at blade rate and twice blade rate showed some modulation at the difference frequency. The pressure fluctuations in the suction and discharge were largely random in character with some contribution at blade rate and twice blade rate but nothing evident at the subsynchronous frequency. The balance return line pressure pulsed strongly at shaft rate and, when present, at the subsynchronous frequency. Impulse tests on the bearing housings showed that the natural frequencies were well away from running speed. The pump rigid body natural frequencies, (on the temporary supports), were about 2500 cpm whilst the characteristics natural frequencies of the pump body and bearing housings were much higher, above 500Hz. The dynamic characteristics obtained during Test 5, i.e. with swirl control, showed stable operation over the complete range of flow and speed available. With clearance opened out to 3x, Test 6, the pump became unstable at a speed of about 4800 rpm and the whirl ratio remained the same as before at 0.84, fig.16. The frequency of the subsynchronous vibration maintained the same relationship to speed up to the test maximum of 5500rpm as had been obtained before (Test 3).

To summarise, the characteristics of the subsynchronous vibration obtained on this pump were:-

1. the vibration appeared at a constant proportion of running speed, whirl ratios being typically 0.85 and was <u>independent</u> of annular seal clearance or actual pump speed.

2. the vibration 'locked in' and 'dropped out' with speed and to a lesser extent flowrate and there was a very rapid growth in amplitude to a limit cycle; typical of a classical self excited oscillation.

3. the vibration contained not only the subsynchronous frequency but also the difference frequency and sidebands.

4. the shaft vibration at the subsynchronous frequency indicated a forward precessing whirl, again typical of a self excited oscillation.

5. there was no evidence to prove that the subsynchronous vibration response was due to a rotor natural frequency (i.e. not an unstable eigenvalue).

6. control of inlet swirl to the balance drum delayed the onset of subsynchronous vibration.

Since the mid 1980's there have been reported several more instances of the "near running speed" nonsynchronous vibration problem, ref 7, 8, 9. In some cases major redesign of the pump has been necessary to solve the problem. Opinions differ regarding the source of the problem, some believe it is the excitation of the natural frequency of the rotor but others believe that hydraulic destabilising forces in the impeller may be the source. Further research and investigation will no doubt ultimately give more insight into such problems.

5.0 TRANSIENT VIBRATIONS

Transient vibrations - vibrations which result from rapidly changing operating conditions and which decay with increasing time.

Transient vibrations in boiler feed pump installations are fairly rare although most modern power stations have probably experienced pipework vibration resulting from either rapid loading or unloading of the pump or from the rapid opening and closing of flow control valves. Disturbed suction conditions can also lead to pipework vibration and vapour locking within the pump causing the pump to eventually run dry. However, there are few documented examples of pump rotor vibrations which result from transient operations. One such problem arose in 1987 on boiler feed pumps installed in a large modern fossil fired power station in South Africa. The incident occurred each time a pump was restarted following a shut down from hot operation. These pumps were fitted with eddy current shaft monitoring equipment and a vibration trip occasionally occurred due to high level shaft displacement during the restart. Normal steady state levels of shaft vibration were typically 30 μm to 50 μm pk-pk raising to 150μm to 175μm during restart. The problem was caused by a thermally induced bow in the pump shaft induced by a temperature differential in the area of the mechanical seal. A typical shaft displacement record is shown in fig.17, where it can be seen that it took between 40 - 50 seconds before the displacement levels have returned to normal following the restart of the pump. From a comprehensive series of tests on site, it was possible to establish the characteristics of the phenomenon in terms of some of the dependent variables. These were:-

1. length of time the pump remained stationary following hot shut down

2. acceleration rate during run up

3. feed temperature

4. journal bearing loading.

In this particular case the 'solution' was simply to operate a vibration inhibit during start up, it being argued from past experience and detailed analysis of the shaft bending forces involved that the pump would not suffer accelerated wear. It is believed that this phenomenon occurs in most boiler feed pumps fitted with mechanical seals. A review of Weir feed pump installations carried out 1988 showed 127 fully operational pumps with over 1.6 million hours operation. None of these were fitted with shaft displacement monitoring with alarm and trip function. In the absence of shaft monitoring the phenomenon has gone unnoticed with, it would appear, no adverse effect on long term integrity.

6.0 CAVITATION RELATED VIBRATIONS

Cavitation related phenomena are associated with the first stage impeller suction region and may be responsible for many unusual dynamic phenomena in pump and piping systems, ref.10.

The suction stage feed pumps designed by Weir Pumps in the 1970's for CEGB incorporated inducers so that they could cope with the low NPSH transients associated with sudden loss of load. One concern was the susceptibility of these designs to promote surging in the suction pipework in the boiler feed system. Works test are generally unrepresentative of pipework installations on site, particularly as the suction pipework often incorporates a throttle device to control the suction pressure at inlet. This provides considerable damping to the suction pipework, effectively inhibiting the propensity to surge. A series of special tests were carried out in the Weir test facilities to examine the boundary of the surge phenomenon at various flowrates and over a wide range of NPSH values. For these tests a free surface was introduced between the throttle valve and the pump suction. A summary of the cavitation surge band is shown in fig.18. It was comforting to find that although a well defined surge envelope could be defined it occurred at NPSH values below that available during loss of load transient.

In 1988 a cavitation related problem was seen during works testing of pressure stage feed pumps for a Nuclear Power Station. These tests were carried out at rated speed of 5610 rpm and over the full range of flow. Tests however, were carried out without the booster stage pump and the NPSH was less than a half of the site NPSH at rated duty falling to less than a third of the site NPSH at 40% rated flow. Although this was considered acceptable for the hydraulic performance measurements it was found to give unsatisfactory vibration levels as measured on the pump bearing housings compared with the vibration targets. On site the pumps were to meet a vibration level of 2.8mm/sec rms (10Hz-1KHz) at duty flow, 4.5mm/sec rms down to 40% flow and 7.1mm/sec rms at 30%, the recirculation flow quantity. It was desirable to demonstrate these levels during the works test. The measured vibrations were higher than target as characterised in terms of broad band excitation, impeller blade frequency excitation and general unsteady/erratic readings. Figure 19 shows the highest vibration recorded during test at 80m NPSH in terms of the overall level, once per revolution and blade rate frequency components compared with the target. It was suspected that cavitation was responsible for the problem. A review of some visual cavitation tests done on a ½ scale model impeller in our Research Laboratory indicated that sizeable bubble formation could exist 80m NPSH despite the fact that no loss in head had occurred. Typical bubble size data are shown in fig.20 where it can be seen that bubble size in excess of 30mm would be expected at the NPSH used during the test. The NPSH was increased to 145m, still only 50% of the site NPSH at duty flow and 50% at 40% rated flow. The vibration levels were found to be very close to the vibration targets as can been seen in fig.21. Clearly this underlines the need to consider carefully before agreeing to meet tight vibration limits during tests designed primarily to demonstrate hydraulic performance.

7.0 CONCLUDING REMARKS

The paper has provided examples of different types of vibration experienced in boiler feed pumps for Power Station duties. It is likely that other manufacturers can cite similar problems but different solutions may have been adopted. The task of the methods of the pump designer is to try and eliminate the problems in any new designs and, preferably, before site installation. This process is helped if manufacturers and users are encouraged to report on problem areas.

Clearly there are generic problem areas. The subsynchronous vibration problem has generated more interest in the US than in Europe and there are many research projects directed toward investigation of hydraulic destabilising forces and fine clearance seal dynamics. Cavitation and NPSH related dynamic effects offer considerable scope for investigation relating to system and pump rotor stability. Impeller blade frequency vibrations are not completely understood and further research into areas of concern would seem desirable.

8.0 REFERENCES

1. UCHIDA, N. Radial Force on the Impeller of a Centrifugal Pump. Bulletin of the ISME, Vol 14, No.76, 1971.

2. HERGT, P., KREIGER, P. Radial Forces in Centrifugal Pumps with Guide Vanes. Conference on Advanced Class Pumps. I.Mech.E., 1970.

3. BROWN, R.D., FRANCE, D. Vibration Response of Large boiler Feed Pumps. Worthington European Technical Award 1974, Vol III, Hoepli, Milan, 1974.

4. ZOTOV, B.N. Selection of Number of Runner and Guide Mechanism Blades for Centrifugal Pumps. Russian Engineering Journal, Vol LII, No.11.

5. MAKAY, E., SZAMODY, O. Survey of Feed Pump Outages. EPRI Report - 754, 1978.

6. FRANCE, D. Rotor Instability in Centrifugal Pumps. Shock and Vibration Digest, Vol. 18, No.1, 1986.

7. MASSEY, I. Subsynchronous Vibration Problems in High Speed Multi Stage Pumps. Proc. 14th Turbomachinery Symposium, Texas A&M University, 1985.

8. MARSCHER, W.D. Subsyncrhonous Vibration in Boiler Feed Pumps due to Stable Response to Hydraulic Forces at Part Load. I.Mech.E Conference 'Part Load Pumping, Operation, Control and Behaviour', 1988.

9. FRANCE, D., TAYLOR, P.W. Near Running Speed Subsynchronous Vibration in Centrifugal Pumps. Conference on Vibrations in Centrifugal Pumps, I.Mech.E., London, 1990.

10. NG, S.L. and BRENNEN, C. Experiments on the Dynamic Behaviour of Cavitating Pumps. ASME Fluids Engineering, 1978.

11. BLACK, H.F. Lateral Stability and Vibrations of High Speed Centrifugal Pump Rotors. Symposium on the Dynamic of Rotors, Lyngby, Denmark.

TABLE I

RADIAL VIBRATION FORCES

Blade Number	Blade Rate Harmonic										
Imp/Diff	1	2	3	4	5	6	7	8	9	10	11
6/7	1.114	0	0	0	0	0.186	0	0	0	0	0
6/9	0	0	0	0	0	0	0	0	0	0	0
6/13	0	1.035	0	0	0	0	0	0	0	0	0.188
7/9	0	0	0	0.358	0.286	0	0	0	0	0	0
7/12	0	0	0	0	0.382	0	0.273	0	0	0	0
	0	0.13*	0	0	0.27*	0	0	0	0	0	0
7/13	0	1.035	0	0	0	0	0	0	0	0	0.188
	0	1.0*	0	0.4*	0.13*	0	0	0	0	0	0
8/7	1.114	0	0	0	0	0.186	0	0	0	0	0
8/9	1.432	0	0	0	0	0	0	0.179	0	0	0
8/12	0	0	0	0	0	0	0	0	0	0	0
8/13	0	0	0	0	0.414	0	0	0.259	0	0	0
5/7	0	0	.371	.279							
5/8	0	0	.424	0	.255	0	0	0	0	0	0

N.B. * **Measured Normalised Radial Force Values for 7 x 12 and 7 x 13 blade combinations**

TABLE II

Test Conditions Prototype Pump

Test No.	Build	Clearances
1	Standard design	nominal design
2.	Standard design	1.5 x nominal design
3.	Standard design	2.0 x nominal design
4.	Balance drum swirl control	nominal design
5.	Balance drum swirl control	2.0 x nominal design
6.	Balance drum swirl control	3.0 x nominal design

TABLE III

Incidence of Subsynchronous Vibration

Test		Subsynchronous	Whirl Frequency Ratio	Onset Speed
1.	Design, New	No	--	--
2.	Desing, 1.5 x	No	--	--
3.	Design, 2.0 x	Yes	0.84	≥3900 rpm
4.	Controlled, New	No	--	--
5.	Controlled, 2.0 x	No	--	--
6.	Controlled, 3.0 x	Yes	0.84	≥4800 rpm

Fig.1. Shaft Speed Hydraulic Unbalance as a Function of Flow (After Uchida)

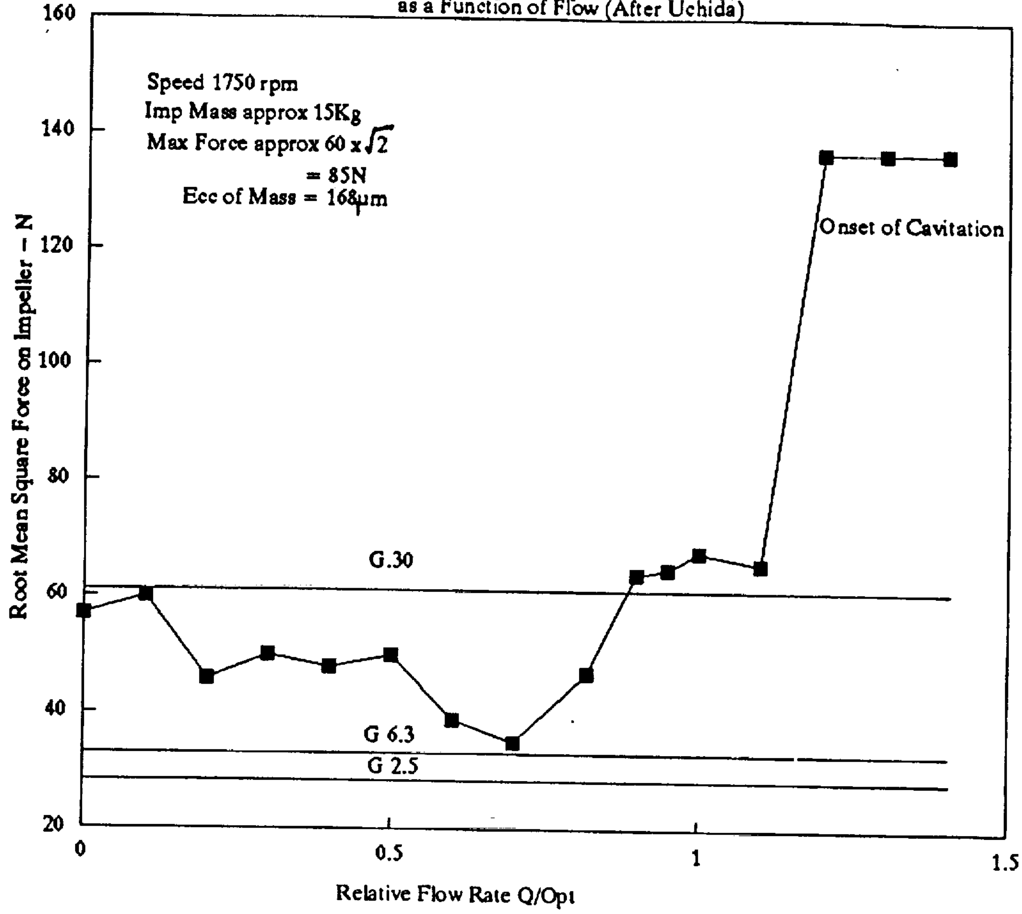


Fig.2 – Pump Shaft Displacement as a Function of Flow

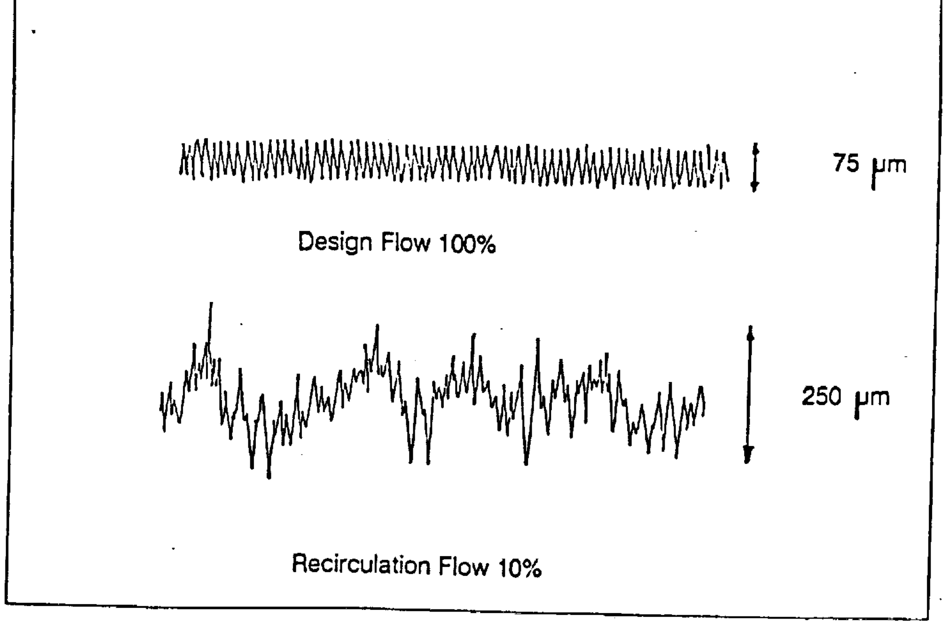


Fig.3a Dynamic Force – Influence of Blade Number and Gap G

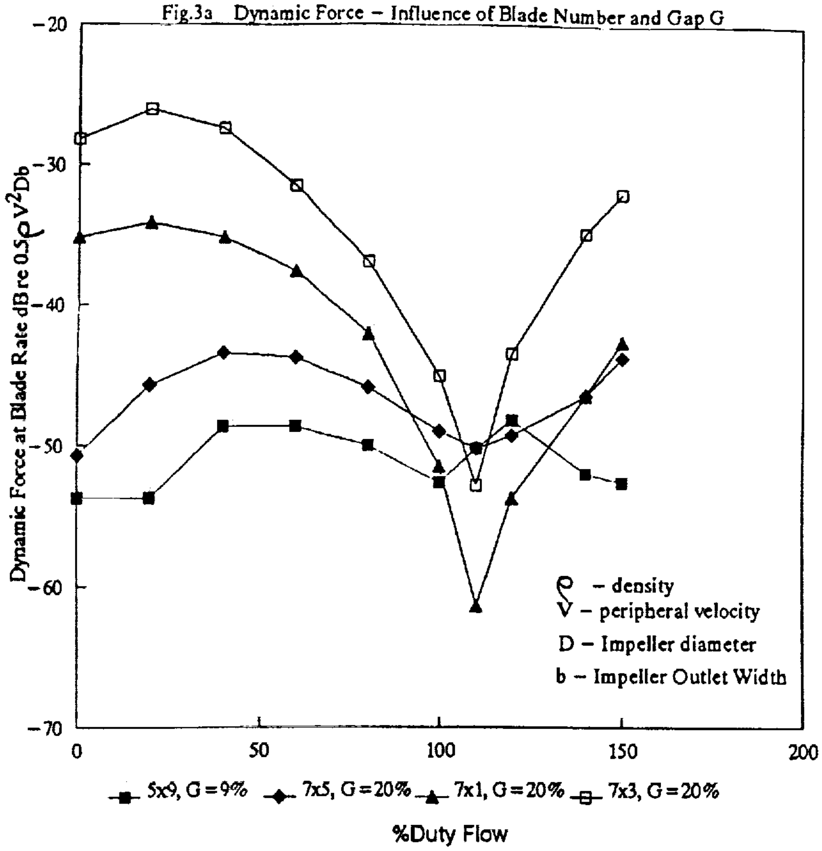
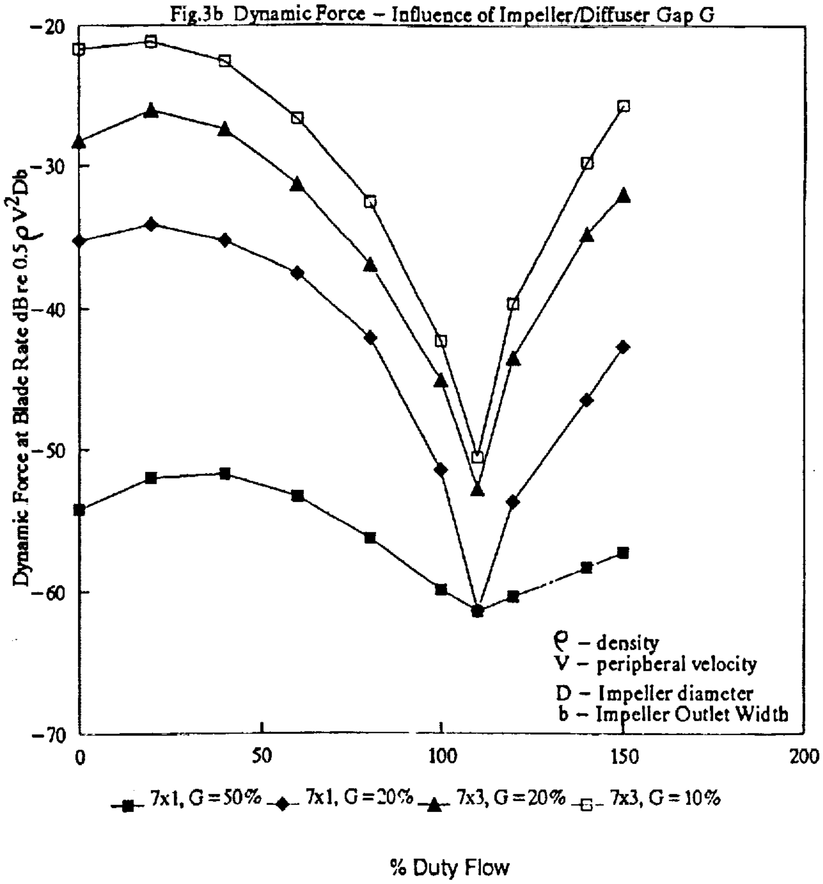
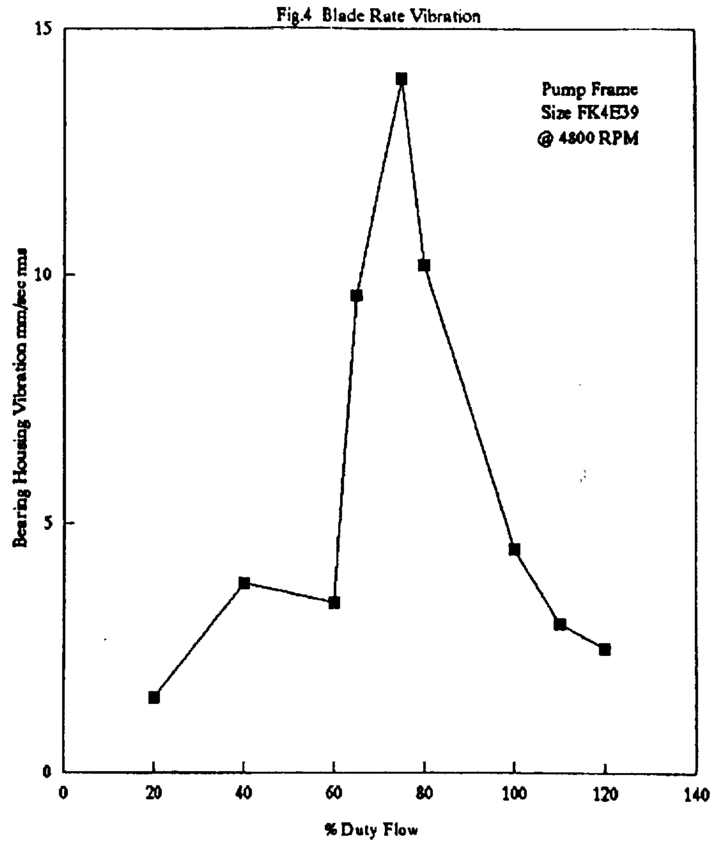


Fig.3b Dynamic Force – Influence of Impeller/Diffuser Gap G





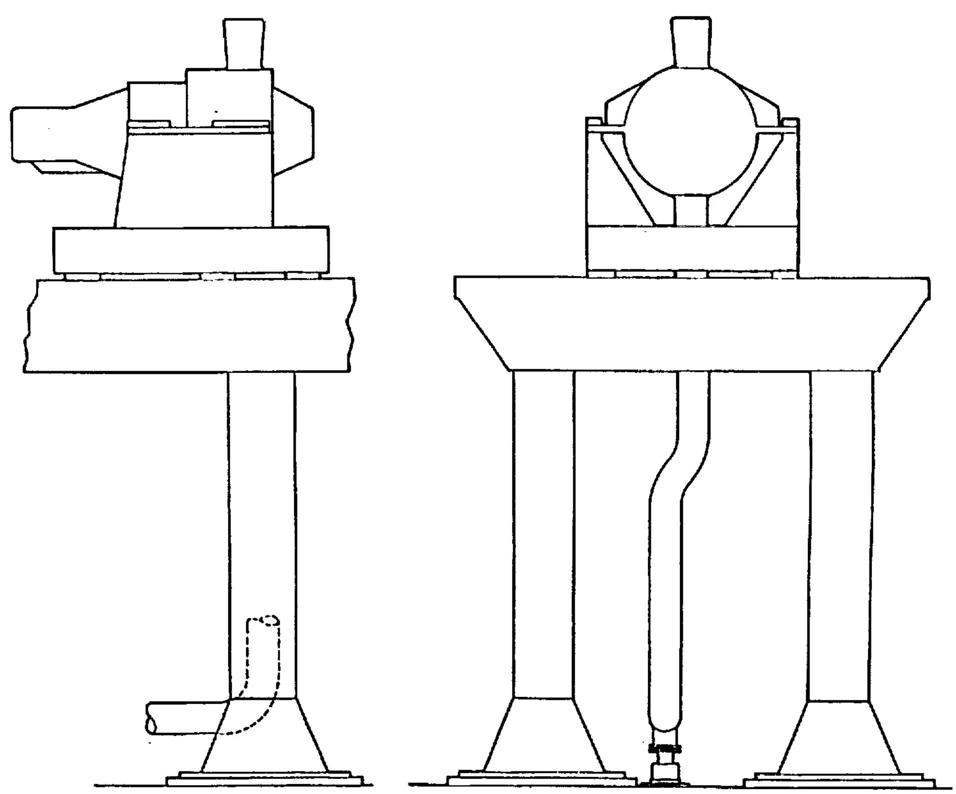


FIG. 5 ARRANGEMENT OF MAIN FEED PUMP
AND SUPPORTING STEELWORK

FIG6a ON LOAD VIBRATION OF PUMP COMPARED WITH VIBRATION OF PUMP EXCITED BY AIR DRIVEN ROTARY OUT OF BALANCE DEVICE

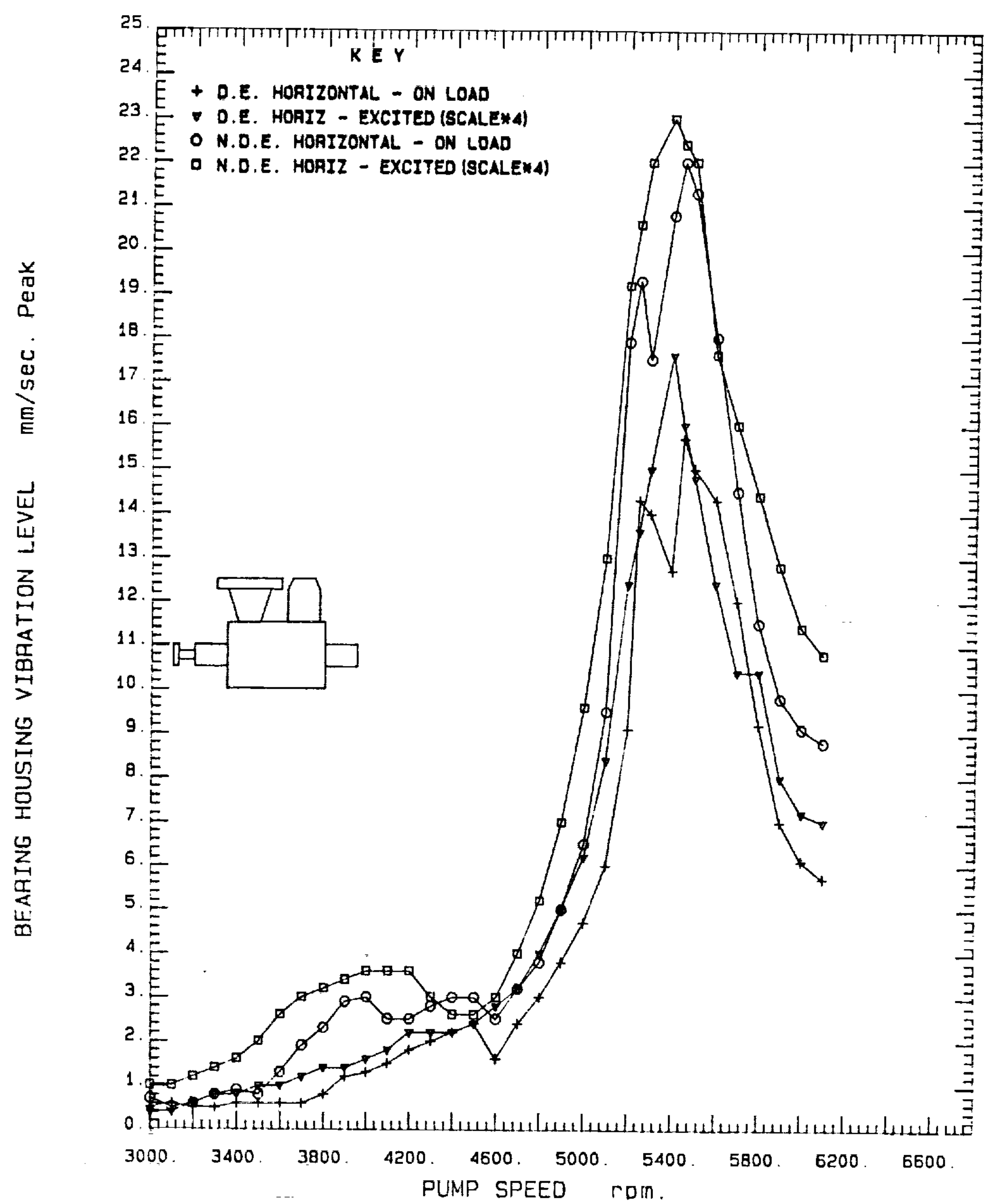
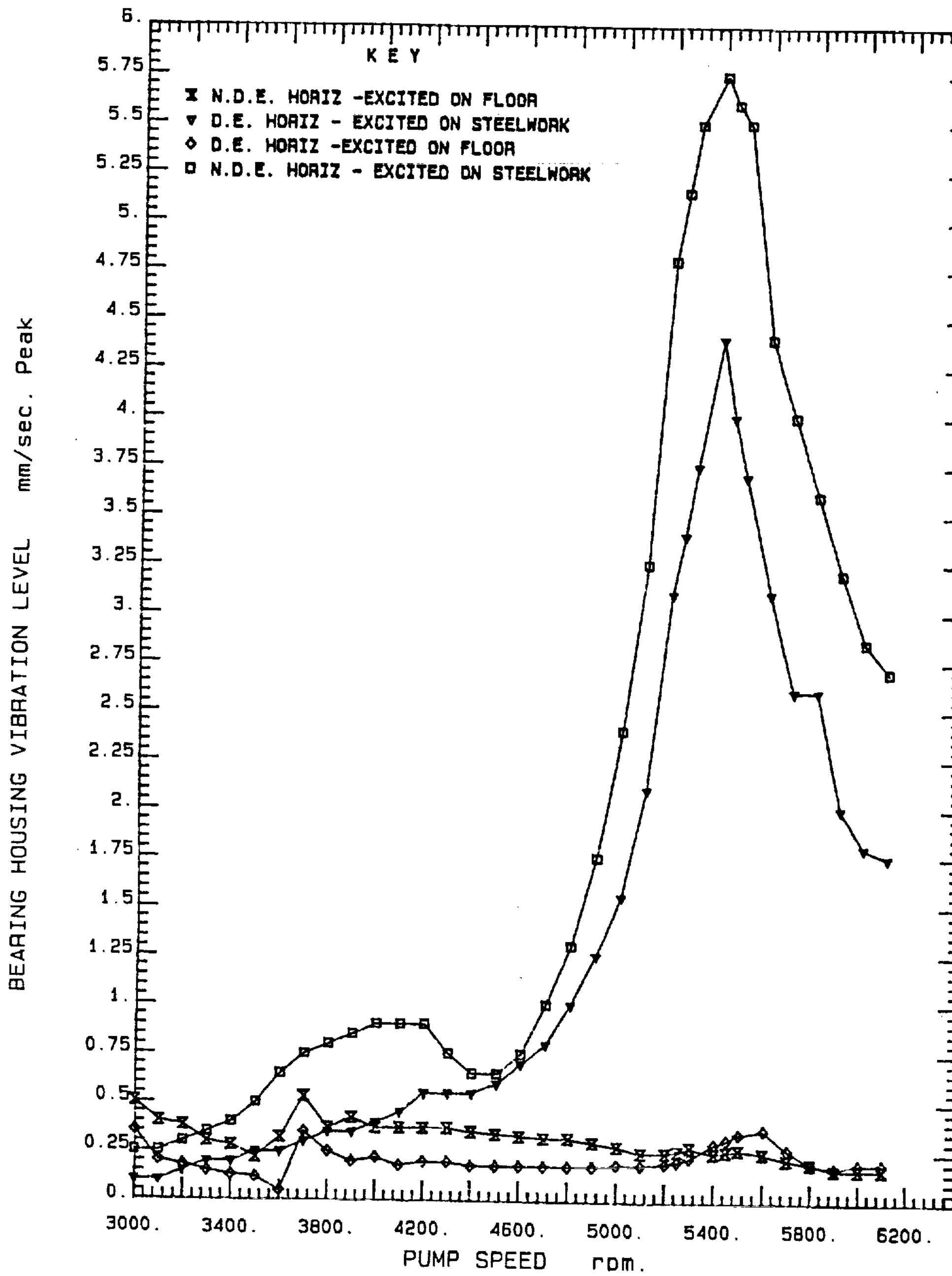


FIG 6b RESPONSE OF PUMP ON STEELWORK COMPARED WITH PUMP ON CONCRETE FLOOR - EXCITED BY ROTARY DEVICE



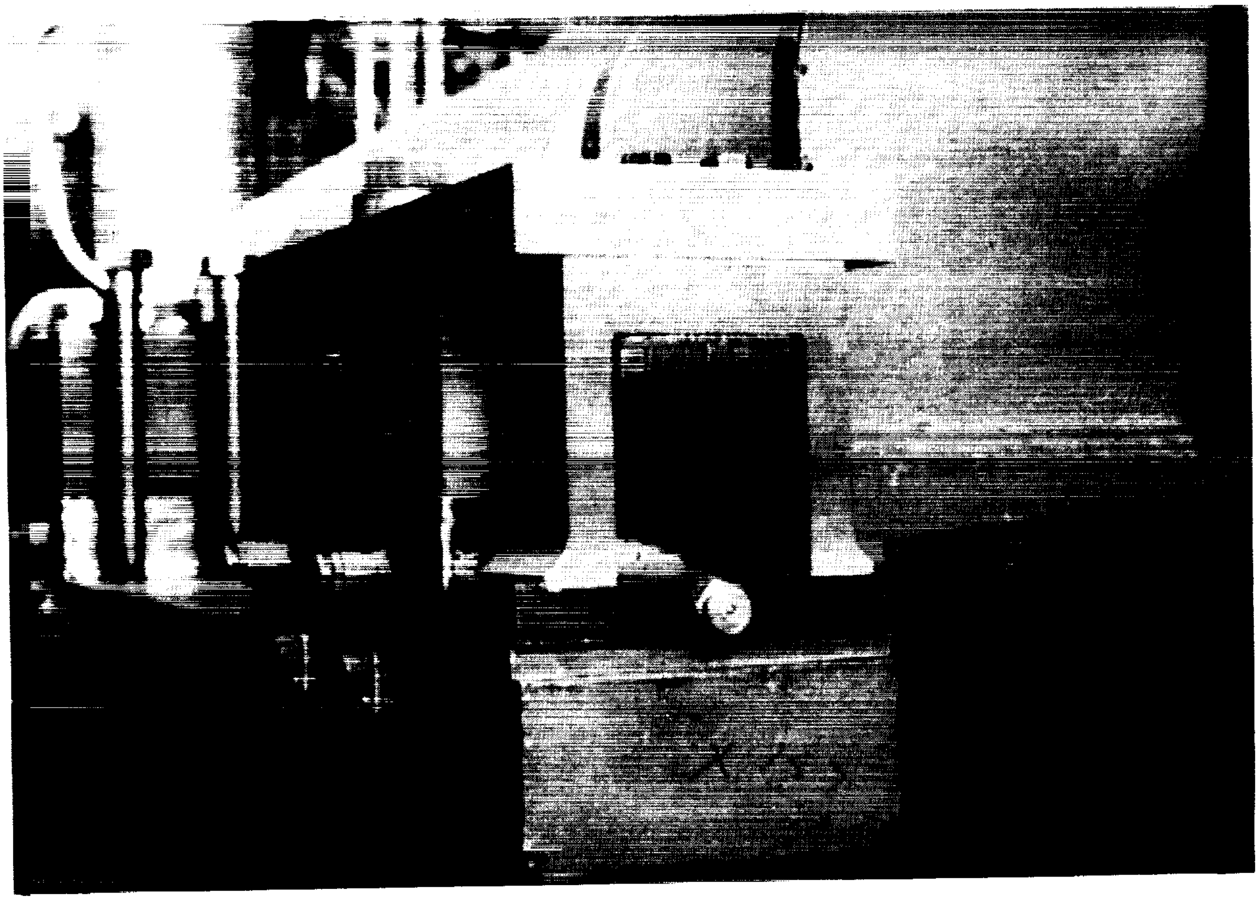
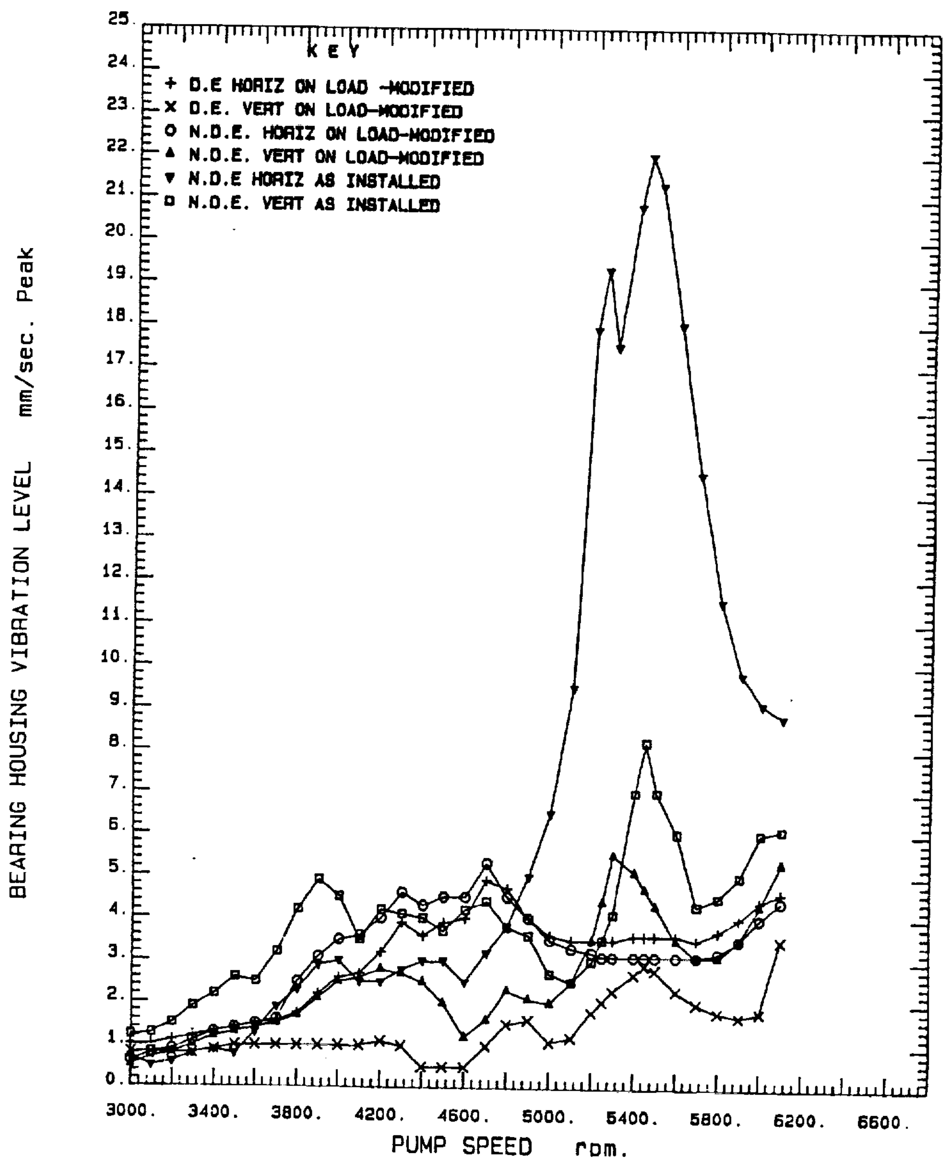


Fig.7 - Detuning Element at Pump Support Feet

FIG 8 ON LOAD VIBRATION RESPONSE OF PUMP FITTED WITH FLEXIBLE DETUNING ELEMENT COMPARED WITH PUMP AS ORIGINALLY INSTALLED



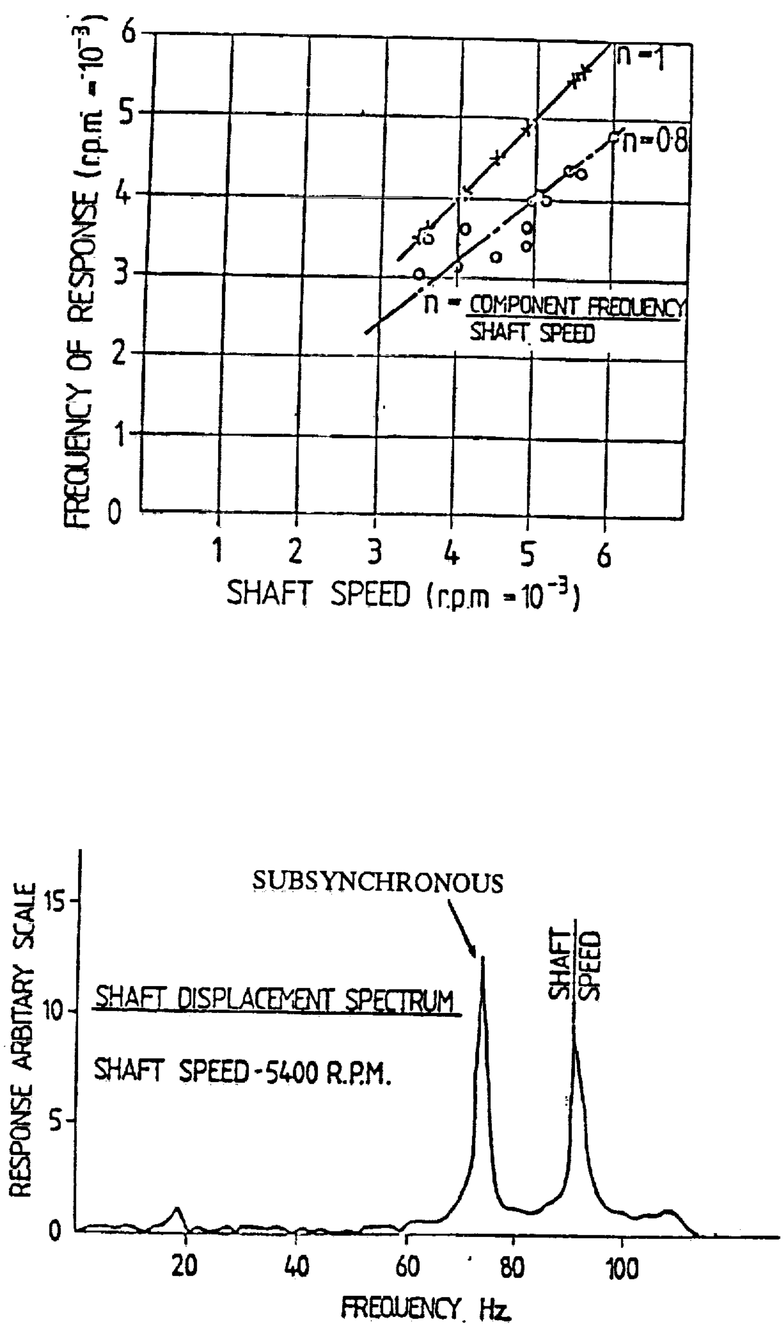
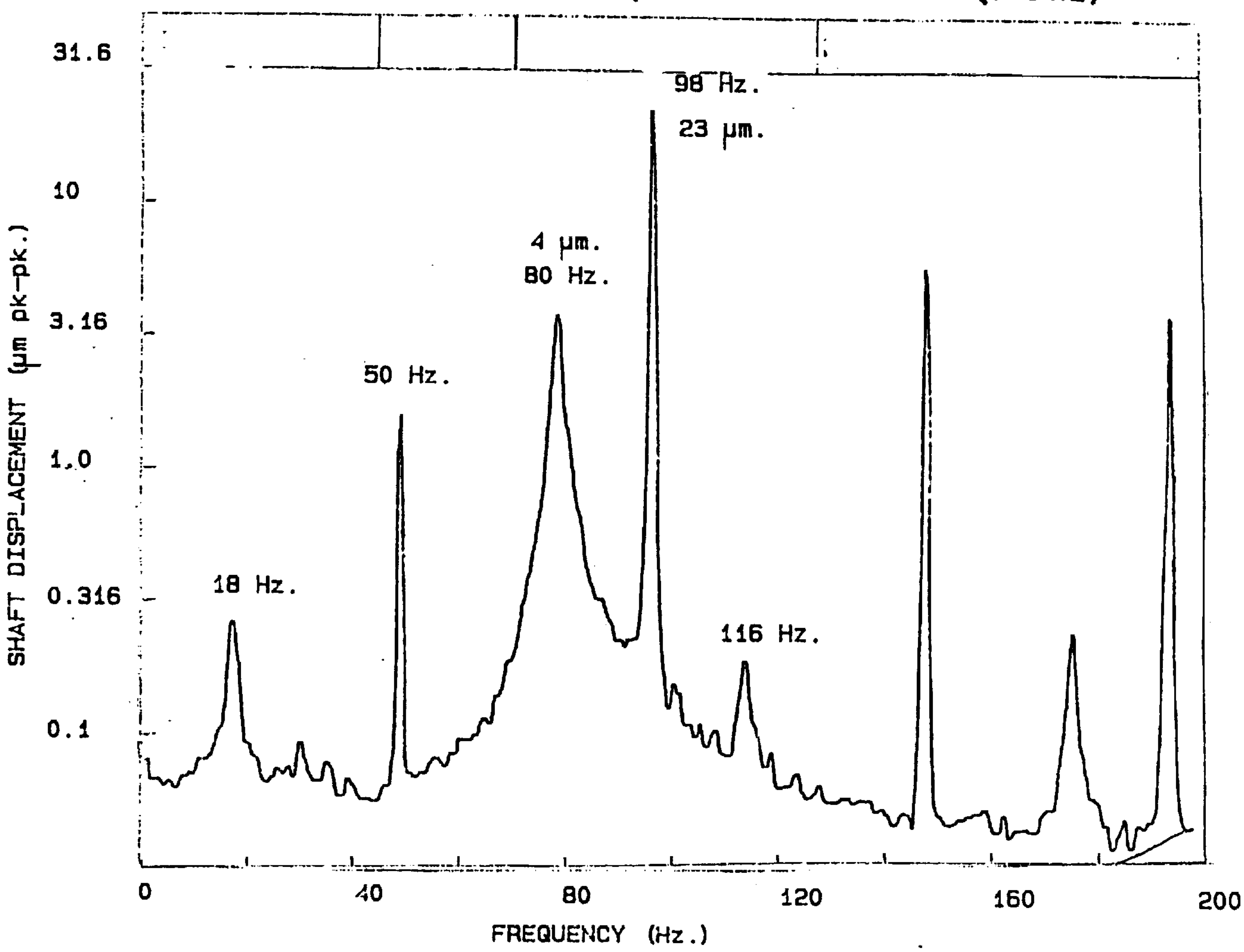


Fig.9. Whirl Frequency Ratio and Frequency Spectrum From 4 Stage Boiler Feed Pump

FIG.10 - Frequency Spectrum taken at Drive End Bearing for 7 Stage Pump with 2 x Design Clearances at 5850 r.p.m.(97.5 Hz)



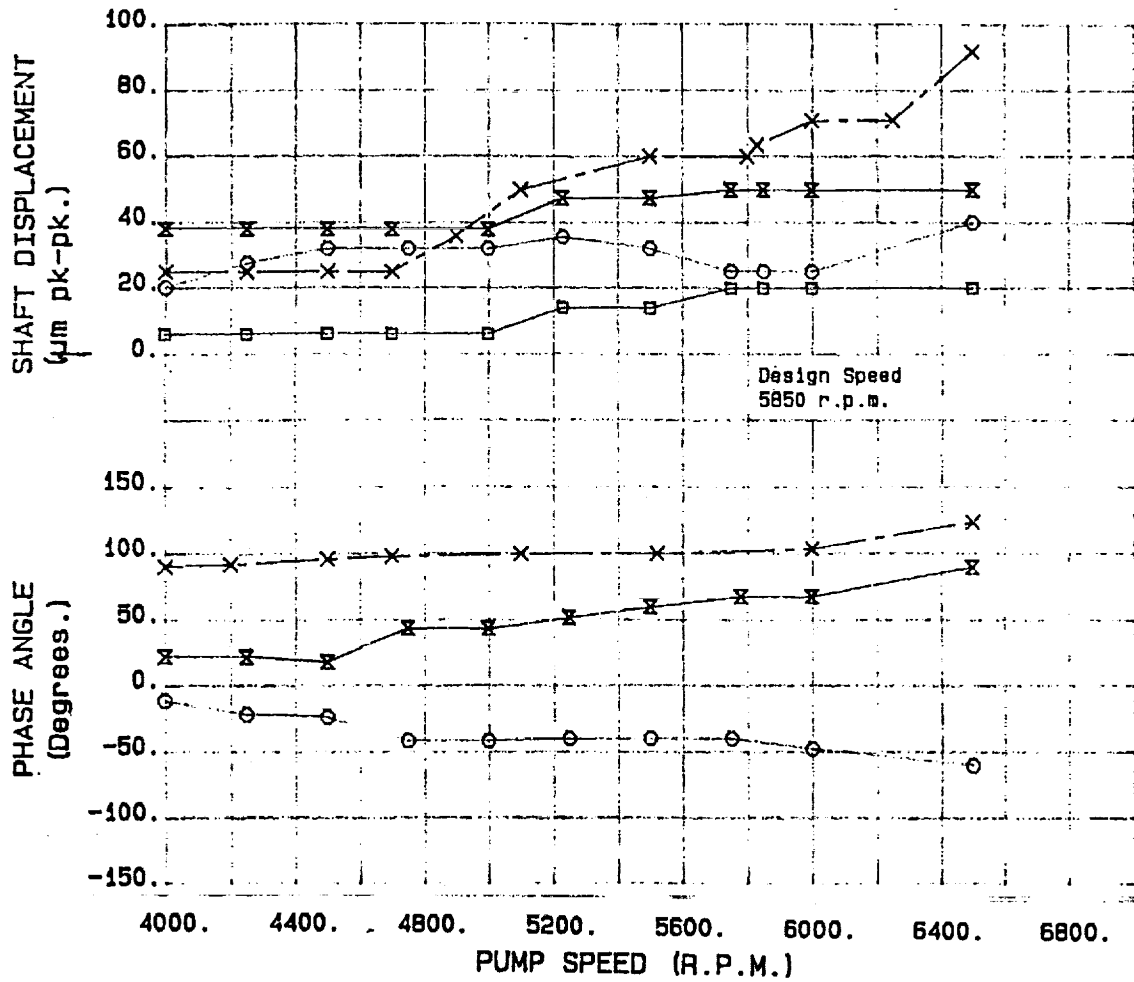


FIG.11a.- Amplitude and Phase for Design and Twice Design Clearances for a 7 Stage Pump

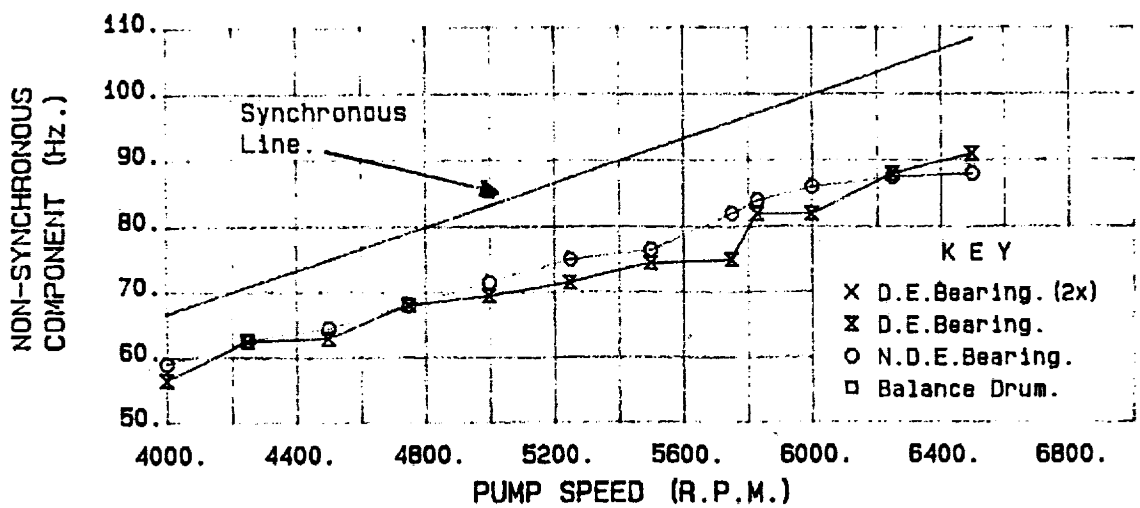


FIG.11b.- Frequency of Non-synchronous Vibration versus Pump Speed for a 7 Stage Pump

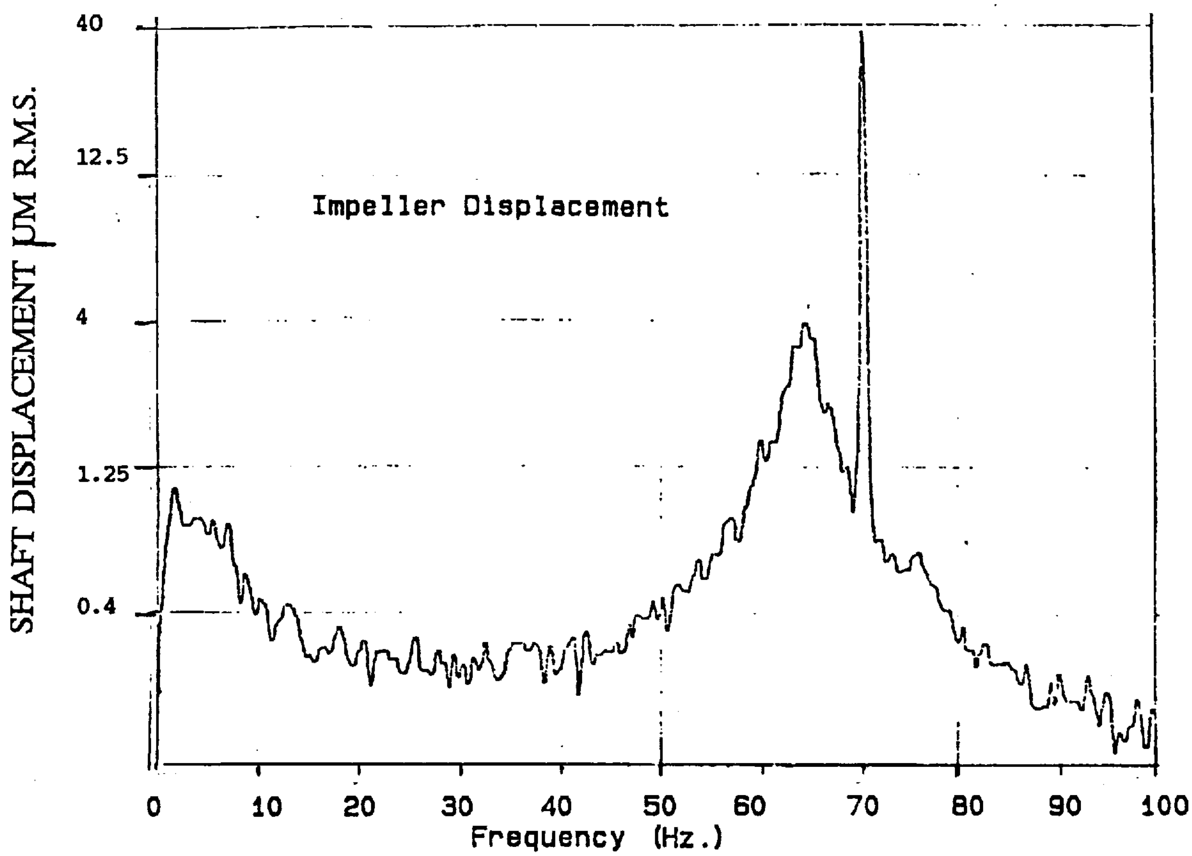


FIG12a - Impeller Displacement From Single Stage Test Rig at 4250 r.p.m. and 65% Best Efficiency Point Flow

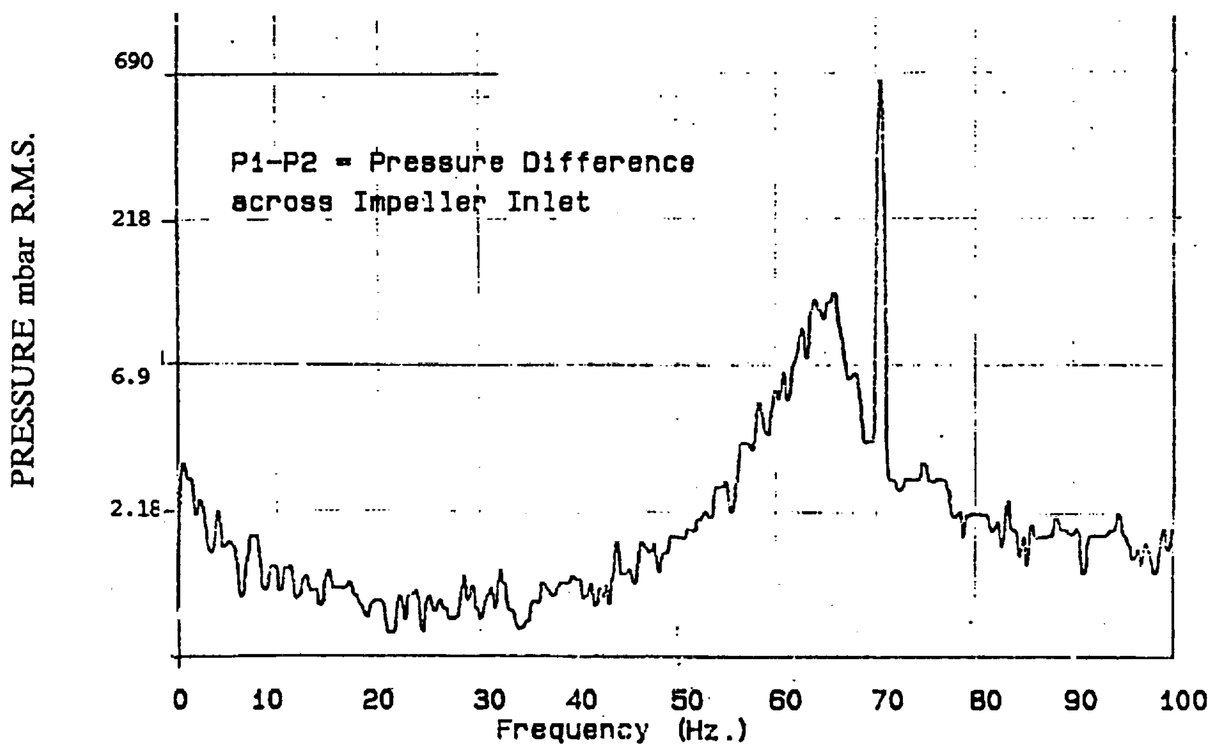


Fig12b - Dynamic Pressure across Impeller Inlet from Single Stage Test Rig at 4250 r.p.m. and 65% Best Efficiency Flow.

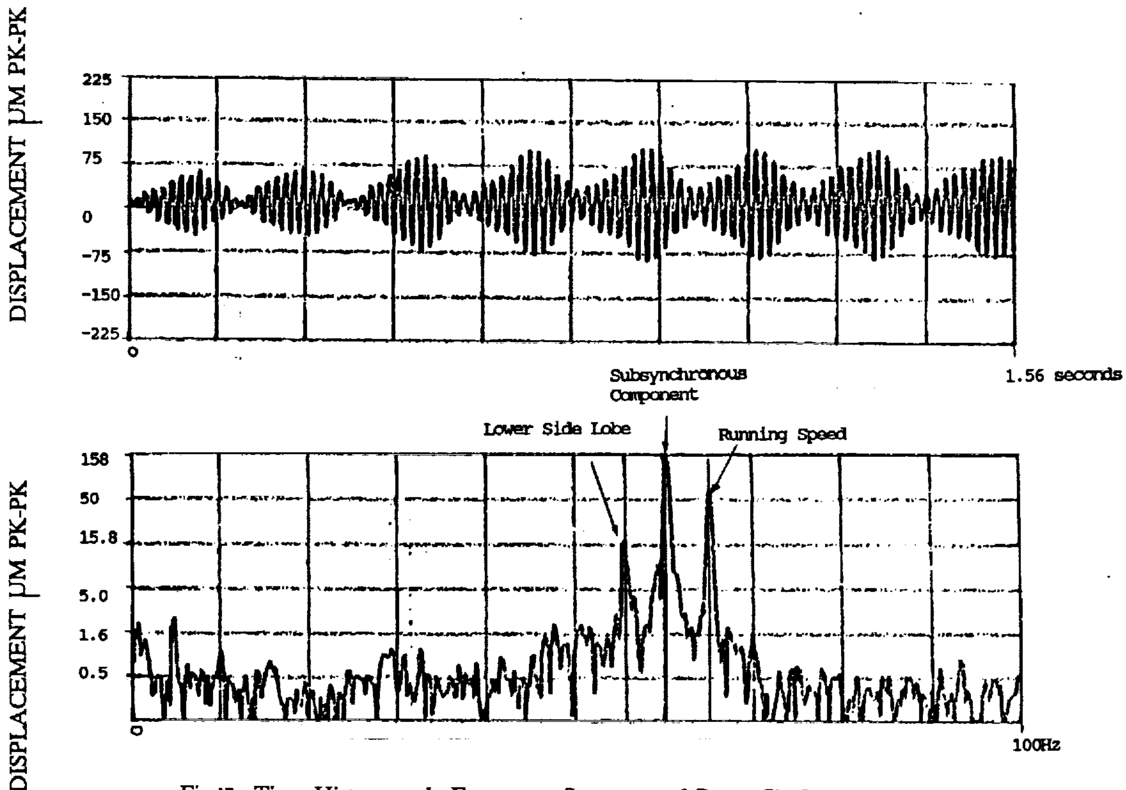


Fig.13 - Time History and Frequency Spectrum of Pump Shaft Displacement [Pump Clearances at Twice Design Values - Refer Test 3]

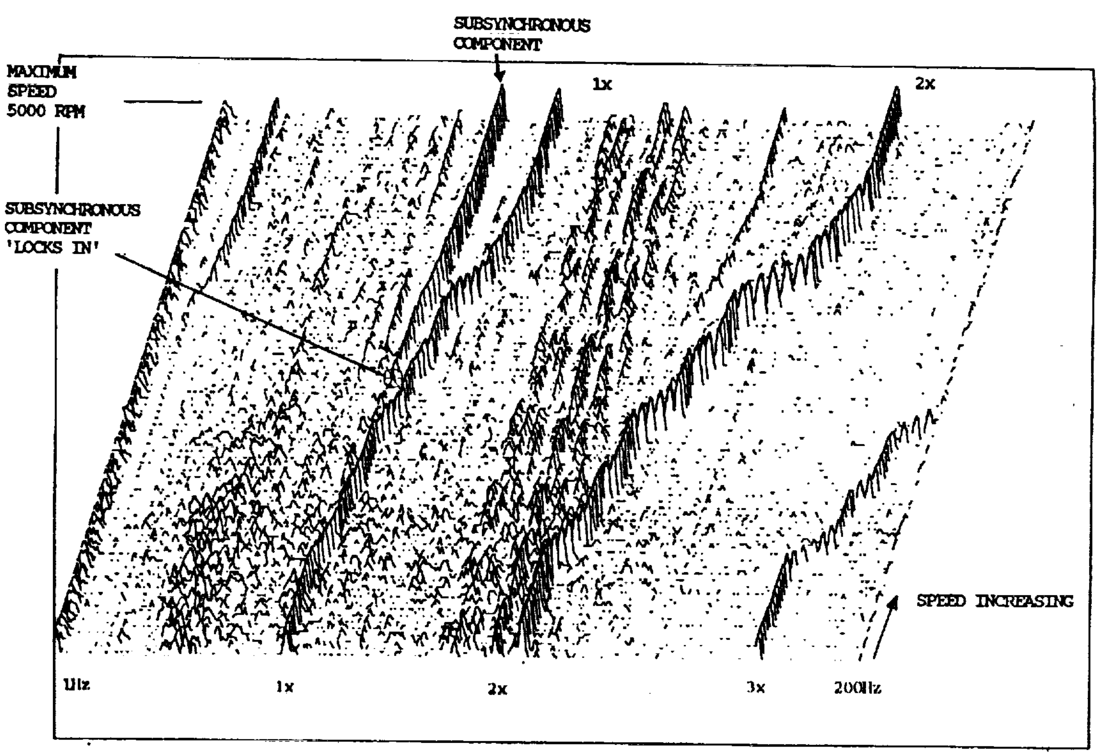


Fig.14 - Waterfall Plot of Shaft Motion during controlled speed increase. (Pump Clearances at Twice Design Values)

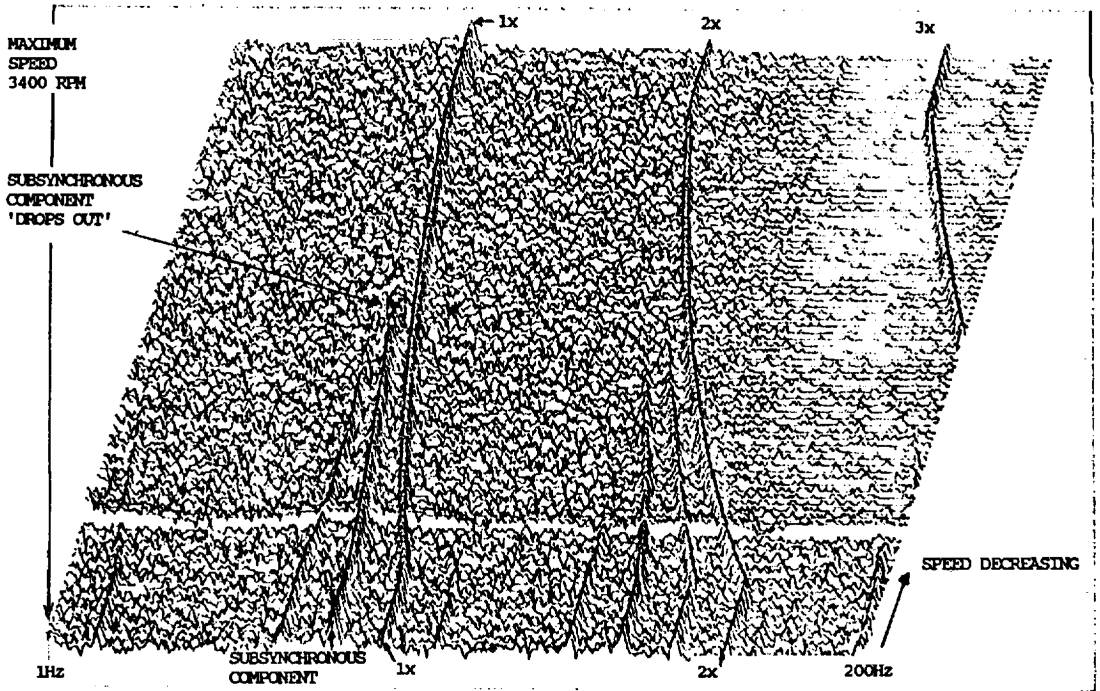


Fig.15 - Waterfall Plot of Shaft Motion during controlled speed reduction. (Pump Clearances at Twice Design Values)

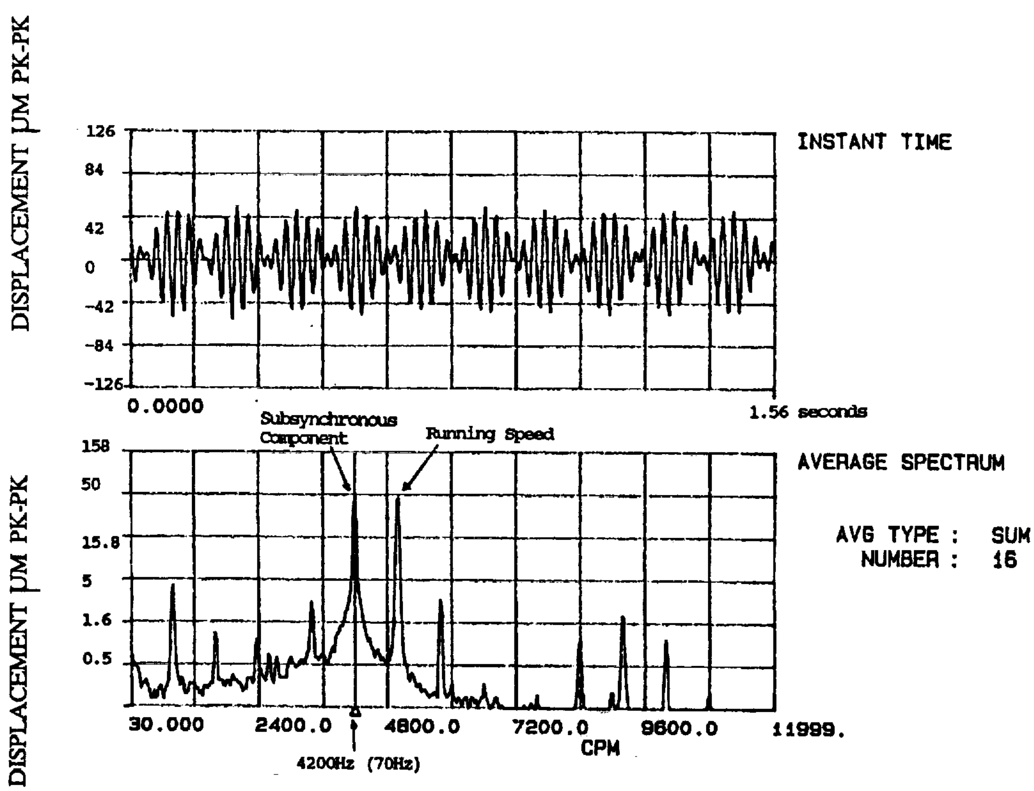


Fig.16 - Time History and Frequency Spectrum of Pump Shaft Displacement [Pump Clearances at 3 x Design, with Swirl Control Test 6]

Fig.17 – Transient Shaft Vibration During Hot Start

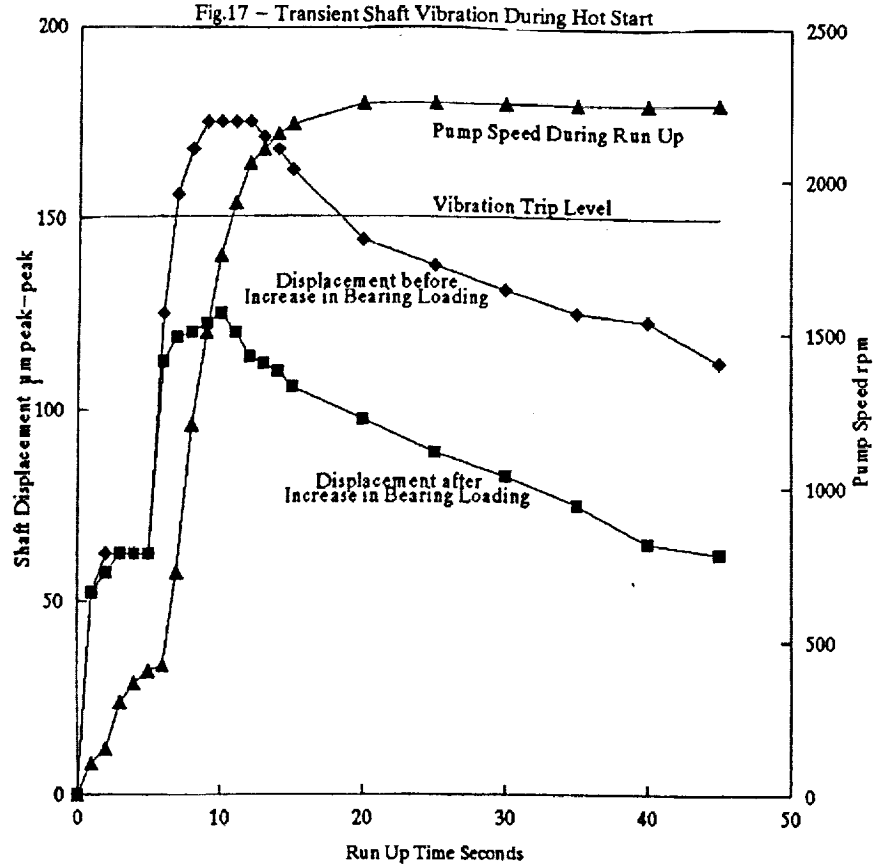


Fig.18 – Cavitation Surge Band on Works Test

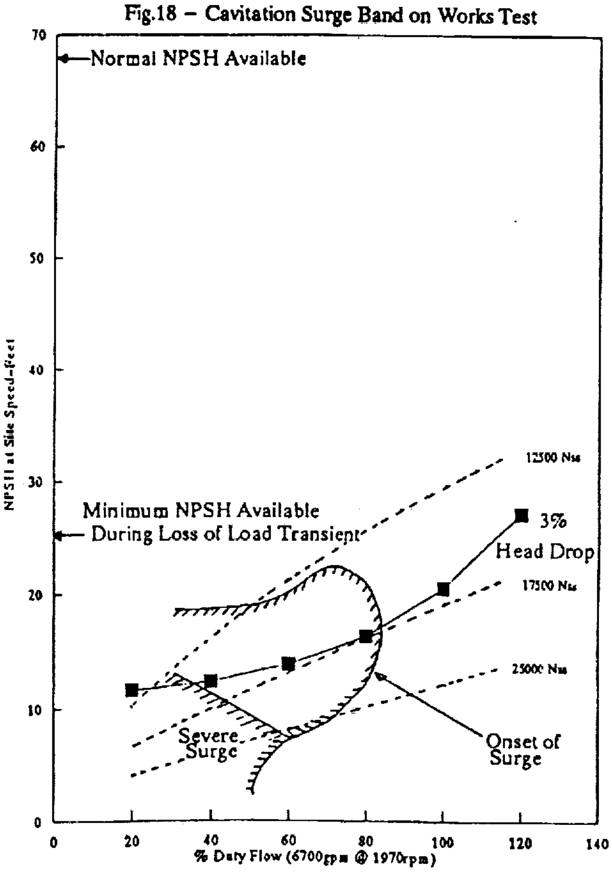


Fig.19 – Pressure Stage Feed Pump Vibration Levels @ 80m NPSH

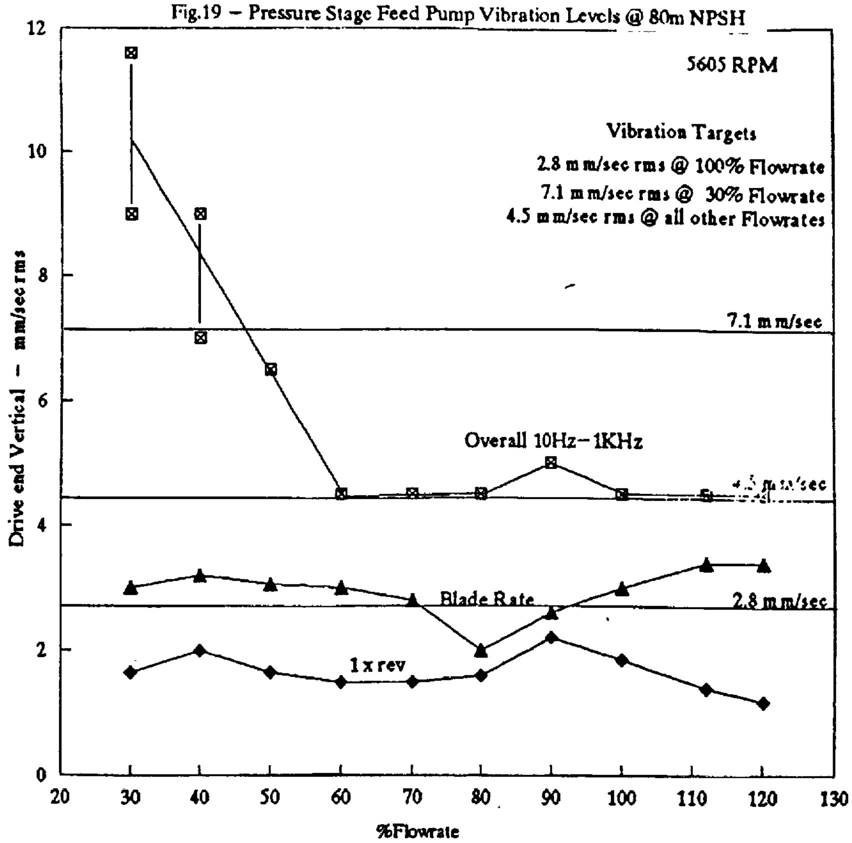


Fig.20 – Scale Model of Cavitation Performance
in Cavitation Visualisation Rig vs Site available NPSH

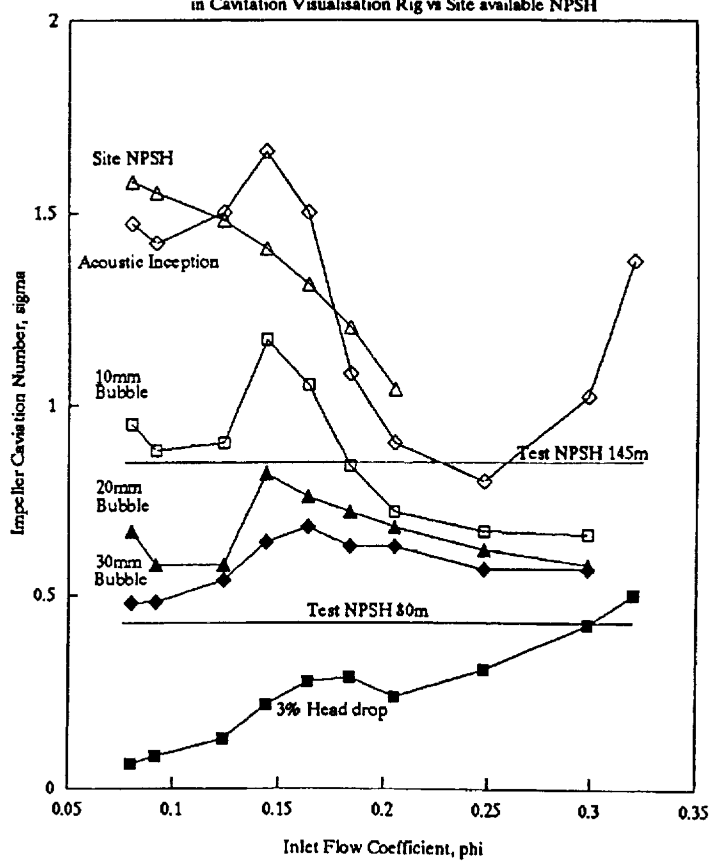


Fig.21 – Pressure Stage Feed Pump Vibration Levels @ 145m NPSH