PINSTECH-158

VIBRATION ANALYSIS OF PRIMARY INLET PIPE LINE DURING STEADY STATE AND TRANSIENT CONDITIONS OF PAKISTAN RESEARCH REACTOR-1

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ABSTRACT

The Primary Water Inlet Pipeline (PW-IPL) is of stainless steel conveying demineralized water from hold-up tank to the reactor pool of Pakistan Research Reactor-1 (PARR-1). The section of the pipeline from heat exchangers to the valve pit is hanger supported in the pump room and the rest of the section from valve pit to the reactor pool is embedded. The PW-IPL is subjected to steady state and transient vibrations. The reactor pumps, which drive the coolant through various circuits mainly contribute the steady state vibrations, while transient vibrations arise due to instant closure of the check valve (water hammer). The ASME Boiler and Pressure Vessel code provides data about the acceptable limits of stresses related to the primary static stress due to steady state vibrations. However, due to complexity in the pipe structure, stresses related to the transient vibrations are neglected in the code.

In this report attempt has been made to analyze both steady state and transient vibrations of PW-IPL of PARR-1. Since, both the steady state and transient vibrations affect the hanger-supported section of the PW-IPL, therefore, it was selected for vibration test measurements. In the analysis vibration data was compared with the allowable limits and estimations of maximum pressure build-up, deflection, natural frequency, tensile and shear load on hanger support, and the ratio of maximum combine stress to the allowable load were made.

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1. INTRODUCTION

The Pakistan Research Reactor-1 (PARR-1) is a swimming pool type, originally designed to operate at a full power level of 5 MW with highly enriched uranium (HEU) fuel. Due to non-proliferation resistance policy adopted by the fuel exporting countries, the availability of HEU fuel virtually became impossible. It was decided to convert it to commercially available low enriched uranium (LEU) fuel. Changing experimental needs and requirement for isotope production demand higher neutron flux levels. Therefore, conversion to LEU was availed as an opportunity to upgrade the reactor power to 9 MW. During conversion and upgradation, most of the reactor systems were modified and several additional facilities were provided. Major modification and additions were carried out in the reactor cooling system. These involved installation of new primary pumps, a set of heat exchangers, a cooling tower and new primary pipelines. To check the systems integrity it is essential to perform vibration tests. Results of the tests performed on the primary pumps and the core support structure comply with the requirements for pre-operational and initial startup vibration testing of nuclear power plant (NPP) systems **[1,2]**. Attempt is now being made to investigate steady state and transient vibrations of the cooling system.

The Primary Cooling System (PCS) of PARR-1 is a closed loop, circulating demineralized water through the core and the heat exchangers, to remove the heat generated in the core. The cooling system has been designed in accordance with the American Society of Mechanical Engineers (ASME) and American Standards Association (ASA) codes **[3,4]**, considering the effects of various loads such as pressure, weight, thermal and seismic loads. The PCS is subjected to various dynamic loads due to the flow-induced vibrations. The steady state vibrations are mainly contributed by the reactor coolant pumps, which drive the coolant through various circuits. These pumps generate periodic and random disturbances in the fluid, which in turn vibrate the system. The transient vibrations in the pipeline arise due to an instant closure of the check valve(s), initiating a water hammer, while switching to new operating condition. Other sources of vibrations are located at the singular points such as cross-sectional enlargements, bends, valves and T-junctions. The presence of these singular points increases the possibility with which such disturbances can excite in resonance beam type flexural modes of the pipe lengths, and may induce vibrations of lower order frequency **[5]**.

Water hammer initiating from startup or shutdown of pump causes built-up of pressure/velocity waves traveling upstream from the point of origin and reflected from the subsequent boundary. It may result in very high amplitude vibration levels. These vibrations can cause the degradation of the piping structures and supports and hence lead to fatigue failure. Therefore, for the integrity of the piping system, it is important to estimate the dynamic stress associated with such vibration modes.

Tests were performed to assess the vibrational levels of the Primary Water Inlet Pipeline (PW-IPL) under steady state and transient operating conditions. Preliminary investigation and subsequent vibration measurement followed by data analysis indicates that at most of the monitoring positions the peak vibration levels are within the permissible range. A detailed analysis of water hammering was conducted to demonstrate the structural integrity of the piping system under transient displacement loads. This report gives the details of the vibration analysis of the inlet pipeline of the PCS of PARR-1.

2. DESCRIPTION OF THE PRIMARY COOLING SYSTEM OF PARR-1

The PCS comprises plenum, a hold-up tank, two sets of heat exchanger assemblies, two identical pumps PW-P1 and PW-P2, valves and piping. The cooling water flows downward under gravity through the reactor core, grid plate and plenum into the hold-up. Subsequently, water is drawn from the hold-up tank by the circulating pumps and pumped through the shell side of the heat exchangers and PW-IPL back into the reactor pool. The hold-up tank, heat exchangers and pumps are located in the pump room adjacent to the reactor containment hall. The flow diagram of PCS is shown in Fig. 1.

The reactor pool level is maintained by equalizing both inlet and outlet coolant flow rates at $900m^3/h$ for full power operation (9MW). The stall and open end outlet lines are combined into one after butterfly valves V01 and V02 in the valve pit. Similarly, the inlet lines for the two sections of the pool are combined into one before the butterfly valves V03 and V04. The piping before the valves is embedded in the reactor hall floor while the piping after the valve pit is mounted on wall supports in a pipe tunnel. At the discharge ends of PW-P1 and PW-P2, two independent check valves NV-06 and NV-32 are installed. After these check valves, the piping are combined to a common header, which provides separate coolant entrance to each of the heat exchanger. The PW-IPL carries the coolant from the outlet of the heat exchangers to the reactor pool, is supported from the roof by the hangers. Since, both the steady and transient vibrations affect PW-IPL; therefore, it has been selected for vibration measurements. The vibration monitoring points are shown in isometric of PW-IPL depicted in Fig. 2.

3. VIBRATION ACCEPTANCE CRITERIA FOR A PIPING SYSTEM

A piping system is subjected to variety of stresses, some introduced by initial fabrication and erection and some due to dead weights (such as pipe, fittings, insulation etc.), contents of the pipe line, earthquake and water hammering. Internal/external pressure and the restraint of thermal expansion may introduce further stresses. Usually, the dead load effects are always maintained, while earthquake and hammering effect will be variable and reach maximum design value occasionally. Pressure and temperature changes usually occur simultaneously and may be relatively uniform for entire service periods. The Pressure Vessel and Piping (PVP) codes contain tables of allowable stresses related only to the primary static stresses present owing to the dead loading. Due to the lack of adequate analysis and complexity many transient stresses are neglected in the PVP code.

Using ASME/ANSI code **[6]**, the portion of the primary piping system to be tested has been classified into Vibration Monitoring Group (VMG). The PARR-1 piping is placed in more stringent VMG-2 group, due to its shorter length compared to the NPP and the type of equipment available for vibration measurements. However, the pipe section classified into one group for steady state vibrations may be classified into another group for transient vibrations.

3.1 Steady State Vibrations

The steady state vibration is determined by measuring the vibration level during normal operating conditions of the piping system. If the vibration level exceeds an acceptable limit, it may be evaluated by measuring displacement or velocity. The velocity method was adopted due to the availability of the test instrumentation and the convenience of the vibration measurement. The allowable peak velocity was used to determine the level in steady state vibration.

Consecutive measurements were made along the pipe length to locate the points exhibiting the maximum vibratory velocity. The criterion for acceptability is;

$$V_{max} \leq V_{allowable} \qquad (1)$$

The expression for allowable velocity is **[6]**,

$$V_{allowable} = \frac{C_1 C_4 664 \times 10^{-13} (S_{el})}{C_3 C_5\ \alpha\ C_2 K_2} \quad (2)$$

where;

- C_1 = a correction factor to compensate for the effect of concentrated weights along the characteristic span of the pipe.
- C_2 = secondary stress index as defined in the ASME.
- C_3 = correction factor accounting for pipe contents and insulation
- C_4 = correction factor for end conditions and for configurations different from straight spans of pipe
- C_5 = correction factor to account off-resonance forced vibration,

$$= \frac{1^{st}\ \text{natural frequency of the pipe}}{\text{Measured frequency}}$$

- K_2 = local stress index as defined in the ASME
- α = allowable stress reduction factor = 1.0 for stainless steel
- S_{el} = 0.8 S_A
- S_A = the alternating stress at 10 cycles from Fig I.9.2.2 of the ASME **[6]**

If the conservative values of the correction factors are combined, a screening velocity criterion can be derived which indicates safe levels of vibration for any type of piping configuration. Using this criterion, piping system can be checked and those with vibration velocity levels lower than the screening value would require no further analysis. Piping system that has vibratory velocity levels higher than the screening value needs further analysis to establish their acceptability.

The piping material of primary cooling system of PARR-1 is stainless steel of type 304L, ASTM-A-312. Considering the following conservatives values of the correction factor the screening velocity for most of the piping configuration is calculated as;

C_1 = *0.15*

C_2K_2 = *4*

C_3 = *1.5*

C_4 = *0.7*

C_5 = *1.0*

S_{el} / α = *7,690 psi*

$V_{allowable}$ = *0.5 in/s or 13 mm/s* (Screening Vibration Velocity Value).

3.2 *Transient Vibrations*

The criteria defined for the steady state vibration (explained in section 3.1) can be used for transient vibrations, if the piping system is in the same VMG-2. If the maximum transient velocity is less than the screening vibration velocity, the piping system is considered as a sound structure which do not requires further analysis. However, if the transient velocity exceeds the maximum allowable limit specified for steady state vibration, the stress analysis should be performed to prove the structural integrity subjected to maximum transient loads.

The transient vibrations occurred instantaneously can be assumed as occasional loads such as, thrust from relief or safety valves, loads from pressure and flow transients, and earthquake. If an OBE earthquake occurs during normal reactor operation, the reactor shutdowns immediately and the cooling pumps continuously operate to remove the decay heat from the reactor core. Therefore, there is no possibility of occurrence of transient vibrations due to check valve slamming with the earthquake.

The maximum combined stress on the piping system can be evaluated by applying the resultant moment due to transient vibration instead of the moment due to earthquake load in the equation given in the ASME [6]. Using the initial design values for the pressure and resultant moment on the pipe cross section due to occasional weight, the maximum combined stress (S_{OL}) can be calculated and compared with the allowable stress.

$$S_{OL} = B_1 \frac{P_{max} D_o}{2t_n} + B_2 \frac{(M_A + M_B)}{Z} \leq 1.8\, S_k \qquad (3)$$

where;

B_1, B_2	=	primary stress indices for the product
P_{max}	=	peak pressure
S_k	=	material allowable stress
D_o	=	outside diameter of pipe
t_n	=	nominal wall thickness
M_A	=	resultant moment loading due to dead weight
M_B	=	resultant moment due to occasional load
Z	=	section modulus of pipe

The maximum pressure buildup due to water hammer is directly proportional to the coolant velocity and magnitude of the surge wave velocity and is independent of the pipe length.

When the water is flowing in an elastic pipe, both the pressure wave velocity and maximum rise in pressure can be expressed as;

$$\Delta P = \rho S_v \Delta V \quad \text{and} \quad S_v = \frac{EE'}{[\rho(E' + Ed/T)]} \qquad (4)$$

where, ΔP is the rise in pressure, S_v is the surge wave velocity, E is the bulk modulus of the liquid, E' is the modulus of elasticity of the pipe material, ρ is the density of liquid and ΔV is the change in velocity of the liquid.

Considering a check valve connected to the pump at one end and a T-junction at the other end (Figure 1). When the valve is suddenly closed due to the shut down of the pump, a compression wave (surge wave) travels upstream with the velocity S_v *to* the end junction and a pressure at the valve is ΔP plus the discharge pressure P_s of the pump that existed at the check valve before it was closed. When the compression wave reaches the junction, it is reflected as an expansion wave. After the expansion wave reflected, the pressure at the valve is ΔP below the static pressure P_s. Compression and expansion waves would travel up and down the pipe alternately and indefinitely if there were no friction. The time taken for the pressure wave to make a round trip is $t = 2L/S_v$, L is the length of the pipe. The pressure ahead of the valve as a function of time is shown in the figure given below. The friction will reduce the amplitude of the pressure wave until equilibrium is attained.

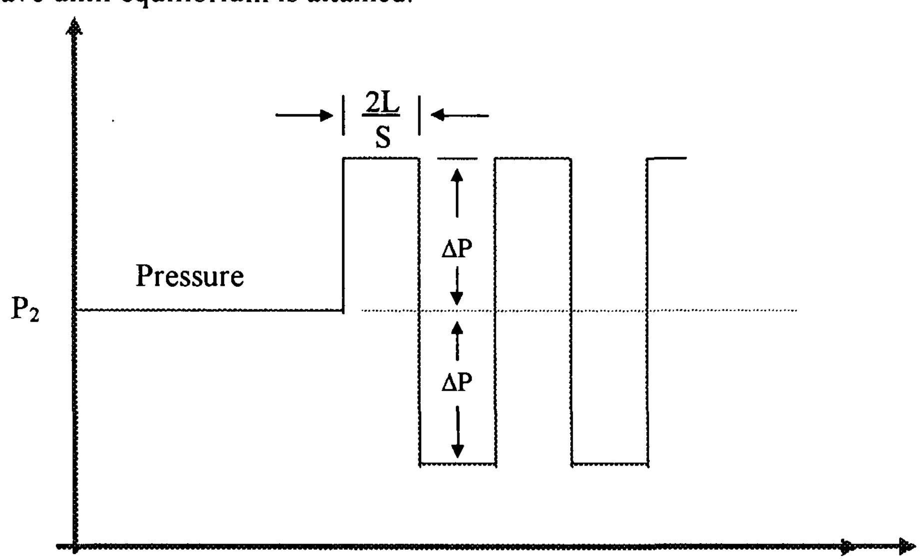

Figure: Pressure Variation Ahead of Closed Valve

To prevent the full rise in pressure, the valve should be closed in time greater than $2L/S_v$. When the closure time is greater than $2L/S_v$, the pressure history at the valve can be estimated by the relation $\Delta P \approx 2\rho VL/\Delta t_c$, where Δt_c is the closure time of the valve.

4. TEST INSTRUMENTATION AND SIGNAL PROCESSING

The vibration monitoring system employed for field measurements comprises single- and triple-axis accelerometers, charge amplifies, high/low pass filters, vibro-meter and dual channel vibration analyzer. The single-axis accelerometer has the maximum input parameter range up to 5 g with the frequency span of 2-6000 Hz and has a sensitivity of 100mv/g, while the triple-axis accelerometer has the frequency span of 1-600 Hz and sensitivity of 1 V/g. The vibro-meter is able to measure peak velocities over the range 10^1-10^3 mm/s, same range for displacement in μm (p-p) and peak acceleration from 0.3-30 g. For accurate measurements over the wide amplitude ranges specified above, the meter is provided with several fixed gain adjustment. Gain normalization of the accelerometer output scale factor is incorporated to read out directly in absolute velocity and displacement units. Maximum frequency range of the dual channel vibration analyzer is 100 kHz. It can analyze vibration, sound, noise and other waveforms by performing time function, auto/cross correlation, coherence, phase impulse response function etc.

Low noise cable with a remote charge amplifier was used between transducers and signal conditioning units to avoid cable noise pickup or signal attenuation due to longer length. The DC component of the signal and extremely low frequency components of the vibration were eliminated with switch selected high pass filter. Anti-aliassing low pass filter was also available at the upper end of the vibration band to eliminate unwanted high frequency noise. Therefore, the proper amplitude function (rms, peak, peak-to-peak) consistent with the acceptance criteria for the measured variable were carefully selected.

On-line signal analyses include peak velocity and displacement measurements, real time signal traces, square root of sum of squares (SSRS) of two signals and auto power spectra. During the analysis of the vibration signal the sampling frequency was 4096 Hz with the frequency bandwidth of 200 Hz. For additional off-line studies and processing, the data was recorded on a FM instrumentation tape recorder and analyzed on a PC based signal analyzer. The experimental set-up for the vibration measurement is shown in Fig. 3.

5. VIBRATION MEASUREMENTS AND ANALYSIS

5.1 *Steady State Vibration Analysis*

The vibratory response of VMG-2 piping has been evaluated in accordance with the allowable velocity limits. Vibrations measured at various locations on the PW-IPL of the cooling system of PARR-1 (Fig. 2) using portable vibrometer are presented in Table-I. From the table it is evident that the V_{max} measured at all the monitoring points is much less than the $V_{allowable}$ or the screening velocities (13mm/s).The maximum peak velocity and its Square Root of Sum of Square (SRSS) value integrated from the acceleration signal measured from triple axis accelerometer are tabulated in Table-II. The table reveals the vibratory response of the PW-IPL in three directions in the steady state condition. The vibration in this condition is within the maximum allowable region.

Auto-power spectrum analysis technique is also applied to identify the full frequency content of the vibration signals. Initial wide band analysis has demonstrated that the majority of the response signals are confined to less than 200 Hz in all the three directions i.e. x (axial), y (vertical) and z (radial), and that there existed significant response at characteristics of pump natural frequency and its harmonics. The frequency spectra measured on the hanger-supported section of inlet pipeline from points PW-IPL1 to PW-IPL4 are shown in Fig. 4 and 5. The spectra measured at these points in three directions, depict frequency resonances within almost same bandwidth. The y (vertical) direction spectra exhibit mainly three signal components, 24.5, 49 and 148 Hz. These peaks correspond to the pump rotational frequency (rpm), its second and sixth harmonics. This reveals that the main source of vibration excitation is the coolant pumps, which transmit pressure wave to the associated structure. The pumps are contributing relatively higher vibration levels in the y-direction compared to other two directions, which is demonstrated in the figures and in the tables. The notable peak in the x-direction is the low frequency structural vibration of 6.5 Hz at position PW-IPL2 and PW-IPL3 and 7 Hz at positions PW-IPL1 and PW-IPL4. These resonances contribute to about 40% of the total vibrations. Similarly, the structural vibration in the z-direction is also present around 6.5 Hz with the slip frequency at 10 Hz, dominantly at position PW-IPL2. The vibration level, in general, is minimum in this direction. Obvious reason for low amplitude vibrations in the x- and z-directions is the design and configuration of the inlet pipeline in the PCS of PARR-1, which limit the motion of the pipe structure in these directions. Another interesting aspect of the analysis is the resonance peak around 61 Hz appearing only in the vibration spectra measured vertically at points PW-IPL1, PW-IPL3 and to some extent at PW-IPL4. This peak may be attributed to the deflection of the pipeline in the middle length of the hanger-supported portion.

5.2 *Transient Vibration Analysis*

Vibrations were measured in two different transient conditions due to the closing of the check valves V-06 and V-32. In Condition No.1 both the primary pumps PW-P01 and PW-P02 were simultaneously shutdown resulting in the closure of the two valves and in Condition No. 2 pump PW-P01 was turned-off while PW-P02 was operating. In this case check valve V-06 closed instantaneously and V-32 remained opened. The maximum peak velocities and peak displacement measured in both the conditions are shown in Table-III and Table-IV. At all the measuring points except at the support hangers positions (PW-IPLH1, PW-IPLH2 and PW-IPLH3) the vibration levels are observed to be lower than the maximum allowable limit. It is also evident from the table that the vibration levels in Condition No.2 are higher than compared with Condition No.1. This is due to the reason that the impact caused due to the sudden closing of the check valve produces a backward hydraulic thrust, which is added to the discharge pressure of the other pump. This results in a buildup of a shock wave that is greater than compared to the simultaneous closing of two check valves.

The typical real time traces of the vibration signal measured on the PW-IPL during two transient conditions are shown in Figs 6 and 7. Generally, all the vertical direction signals from accelerometers placed at selected positions exhibit similar characteristics; the only difference is the amplitude that varies with the monitoring positions. The time scale trace shows the dynamic behaviour of the signal, which is characterized as the impulse response.

6. ANALYTICAL ANALYSIS OF WATER HAMMERING

The phenomenon of water hammer is extremely complex, in the first phase of the stress analysis parameters, such as, the magnitude of pressure and its distribution along a pipe length based on surge-water theory, deflection of the pipe line, natural frequency, and ratio of the maximum combined and allowable stresses are evaluated in the following sections.

6.1 Maximum Pressure Build-up

The velocity of the pressure wave S_v is a fundamental factor in any surge study and this velocity depends on the pipe diameter, wall thickness, material of the pipe walls, as well as density and compressibility of the fluid in the pipe. The piping material of primary cooling system of PARR-1 is stainless steel of type 304L, ASTM-A-312. Knowing the Young's modules and compressibility or bulk modules of water in the temperature range 0-38°C and pressure range of 0-1000 psi, the surge wave velocity and maximum pressure rise for instantaneous valve closure, computed from the relations (4) in section 3.2 are;

$$S_v = 1172\ m/s \quad , \quad \Delta P = 580\ psi$$

When the flow rate is changed within a time scale $0 < t \leq 2L/S_v$ seconds (critical time of pipe), the magnitude of the pressure rise is the same as with instantaneous closure, but the duration of the maximum value decreases as the time approaches $2L/S_v$ seconds. The pressure distribution along the pipeline varies as the time of value closure varies. The pressure decreases uniformly along the line if the closure is in $2L/S_v$ seconds. The maximum pressure at the check valve exists along the full length of the pipe line with instantaneous closure, and for slower rates travels up in pipe a distance equal to $L - (T S_v /2)$ meters, then decreases uniformly. The surge pressure distribution along the conduit is independent of the profile or ground contour of the line as long as the total pressure remains above the vapour pressure of the fluid.

The profile of the pipe leading away from a pumping station may have a major influence on the surge conditions. When high points occurs along the pipe line the surge hydraulic-grade elevation may fall below pipe profile causing negative pressures may be as low as vapour pressure of the fluid. This results in a separation of the liquid column by a zone of vapour for short time. Parting and rejoining of the liquid column can produce extremely high pressures and may cause failure of the pipe line **[7].**

6.2 Deflection

Line movement or deflection is of interest in the design of yielding supports and establishing clearances for the free expansion movement of complex line. The most important reason for limiting deflection is to make pipe stiff enough, that is, of high natural frequency, to avoid large amplitude response under transient conditions. In most of the process systems, the deflection of the pipeline is kept with in the reasonable limit in order to minimize pocketing and to avoid possible sagging. A practical limit for nuclear reactor piping, a deflection as small as 1.25 mm is specified by some designer **[8]**.

The deflection (δ) of the section of the suspended portion of the primary cooling inlet pipe line under investigation can be approximated by the beam relation **[9]** to be about 1 mm.

6.3 Estimation of Natural Frequency

The natural frequency of the test pipe section of primary system of PARR-1 restrained by the supports is calculated to be about 31 Hz. This frequency is higher than the 24.5 Hz, which corresponds to the rpm of the pumps. Therefore, resonance effect due to the unbalance of pump rotating masses is not to be expected in the piping system. An unbalance centrifugal pump will also have a fairly pronounced excitation at twice its rpm, which is called *secondary unbalance force*. When the secondary excitation gets into resonance it can cause failure, which were experienced earlier **[10]**. However, if all the supports are rigid, even the secondary force will not cause resonance in the pipe structure shown in Fig 1. The lower bound due to the possible flexibility (looseness) of supports is estimated to be 16 Hz, which is lower than the rpm.

It appears that the fundamental structural natural frequency of the primary piping system in the rpm range of the pumps and its resonant effect are possible if all supports are flexible.

6.4 Supports Verification

The transient loading condition warrants special consideration because of the added stress, which can be introduced by the rate of application of the surge force. Very rapid loading can raise the yielding point of supports and the localized yielding at the points of stress concentration may lead to a fracture. A computer program was used to calculate maximum tensile and shear loads, which are given in Tables-V and VI. These loads are calculated for two locations on the support hanger (Fig. 8) by taking into account the pipe and vibration data;

Maximum weight load on a support of test section of pipe is	1224 lb. (5445 N)
Maximum transient vibrations load is	1469 lb. (6534 N)

The transient vibrations measured at the selected positions IPL-H1, IPL-H2 and IPL-H3 on the hanger supports are in the range of maximum allowable limit for Case 1. However, the vibration levels exceed the allowable limit for Case 2 (Table-III). In addition to the reason for excessive vibration in Case 2 explained in section *IV-II,* it was revealed that some bolts of the hangers needed tightening, which also result in the low tensile load safety factor (Table-V).

6.5 *Stress Analysis*

The stress analysis of the IPL test section was performed by dividing the pipe into separate mathematical model based on anchor points. Maximum peak displacement measured during the two transient vibration conditions (Table-IV) was used for the estimation of maximum combine stress. The ratios of maximum combined stress to the allowable stress for key monitoring points are presented in Table-VII. The results shown in the table are within the acceptable limits since the stress ratio is less than unity.

7. CONCLUSIONS

Based on the results obtained from the vibration analysis of the test section of inlet pipeline of the primary cooling system of PARR-1, the following conclusion are made;

1. The primary piping system of PARR-1 has been classified into VMG-2 group defined in the ASME code as the requirements for Preoperational and internal startup vibration testing of NPP piping system.
2. The maximum vibration levels measured on the PW-IPL in the steady state operating conditions were within the allowable vibration limits.
3. The transient vibrations measured at the selected positions on the suspended section of the PW-IPL are in the range of allowable vibration limits. However, the vibration levels on the hanger supports exceed the allowable limit.
4. The maximum pressure build due to water hammering is independent of the pipeline length but depends upon the valve closure time. The increase in the valve closing time reduces the impact of the pressure surge wave.
5. The deflector of the pipeline is less than the limit specified in the code for nuclear reactor.
6. The estimated natural frequency of the test pipe section restrained by supports is higher than the reactor coolant pump rpm, hence, reduced the possibility of resonance frequency.
7. The estimated tensile and shear loads on the hanger rod is less than the allowable limit. However, these loads on the bolt connection are higher and the safety margin is quite low.

It is recommended that in order to minimize the vibration load the pipe structure and supports should be rigid and tight. In the next phase of the analysis, it is planned to perform stress analysis of the piping system taking into account the distributed load.

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9. ACKNOWLEDGMENT

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Table-I: Vibration Measurements at Various Locations on PW-IPL of PARR-1 using Vibration Meter

Locations	Acceleration (g)	Displacement (μm) p-p	Velocity (mm/s)
PW-P1-S1	0.23	4.2	1.2
PW-P1-D1	0.48	7.4	1.4
PW-D2	0.20	4.0	1.4
PWHX1-O	0.32	6.0	1.4
PWHXS-O	0.20	4.0	1.2
PW-IPL-1	0.50	10.0	2.0
PW-IPL-2	0.05	9.0	1.6
PW-IPL-3	0.30	9.3	1.2
PW-IPL-4	0.20	9.8	1.8
IPL-H1	0.05	3.0	0.8
IPL-H2	0.13	3.5	1.0
IPL-H3	0.20	30.0	6.5

Table-II: Steady State Vibrations of PW-IPL at the Key Monitoring Points Using Triple Axis Accelerometer

Location	Direction	Max. Velocity (mm/s)	SRSS (mm/s)	$V_{allowable}$ (mm/s)
	X	4.0		
PW-IPL1	Y	8.0	$\sqrt{X^2+Y^2}$ 8.94	
	Z	2.0		
	X	6.0		
PW-IPL2	Y	7.5	$\sqrt{X^2+Y^2}$ 9.6	13
	Z	2.8		
	X	2.8		
PW-IPL3	Y	9.5	$\sqrt{X^2+Y^2}$ 9.9	
	Z	2.0		
	X	5.0		
PW-IPL4	Y	9.5	$\sqrt{X^2+Y^2}$ 11.8	
	Z	1.3		

Table-III: Maximum Peak Velocity Measurements on PW-IPL During Transient Vibrations

Locations	Condition No. 1 Max. Velocity (mm/s)	Condition No. 2 Max.Velocity (mm/s)
PW-IPL-1	8	9.5
PW-IPL-2	8.5	10
PW-IPL-3	10	11
PW-IPL-4	9	11
IPL-H1	8	23
IPL-H2	8	50
IPL-H3	6	19

Table-IV: Maximum Peak Displacement Measurements on PW-IPL During Transient Vibrations

Locations	Condition No. 1 Max. Displacement (μm)	Condition No. 2 Max. Displacement (μm)
PW-IL-1	20	50
PW-IL-2	30	70
PW-IL-3	60	60
PW-IL-4	22	580
IL-H1	50	380
IL-H2	50	800
IL-H3	40	190

Table -V: Maximum Tensile Load at Two Different Positions on the Support

S. No.	Locations on Support	Tensile Load (N) Calculated	Tensile Load (N) Max. Allowable	Min. Safety Factor
1.	Hanger Rod	11979	24432	2
2.	Bolt Connection at Location 1	17424	16365	0.9

Table-VI: Maximum Shear Load at Two Different Positions on the Support

S. No.	Locations on Support	Shear Load (N) Calculated	Shear Load (N) Max. Allowable	Min. Safety Factor
1.	Bolt Connection at Location 1	11979	12800	1.07
2.	Bolt Connection at Location 2	5989.5	14616	2.4

Table-VII: Stress Ratio for the PW-IPL during Transient Conditions

Transient Conditions	PW-IPL1	PW-IPL-2	PW-IPL 3	PW-IPL 4
Condition No. 1	0.44	0.43	0.62	0.51
Condition No. 2	0.56	0.57	0.72	0.71

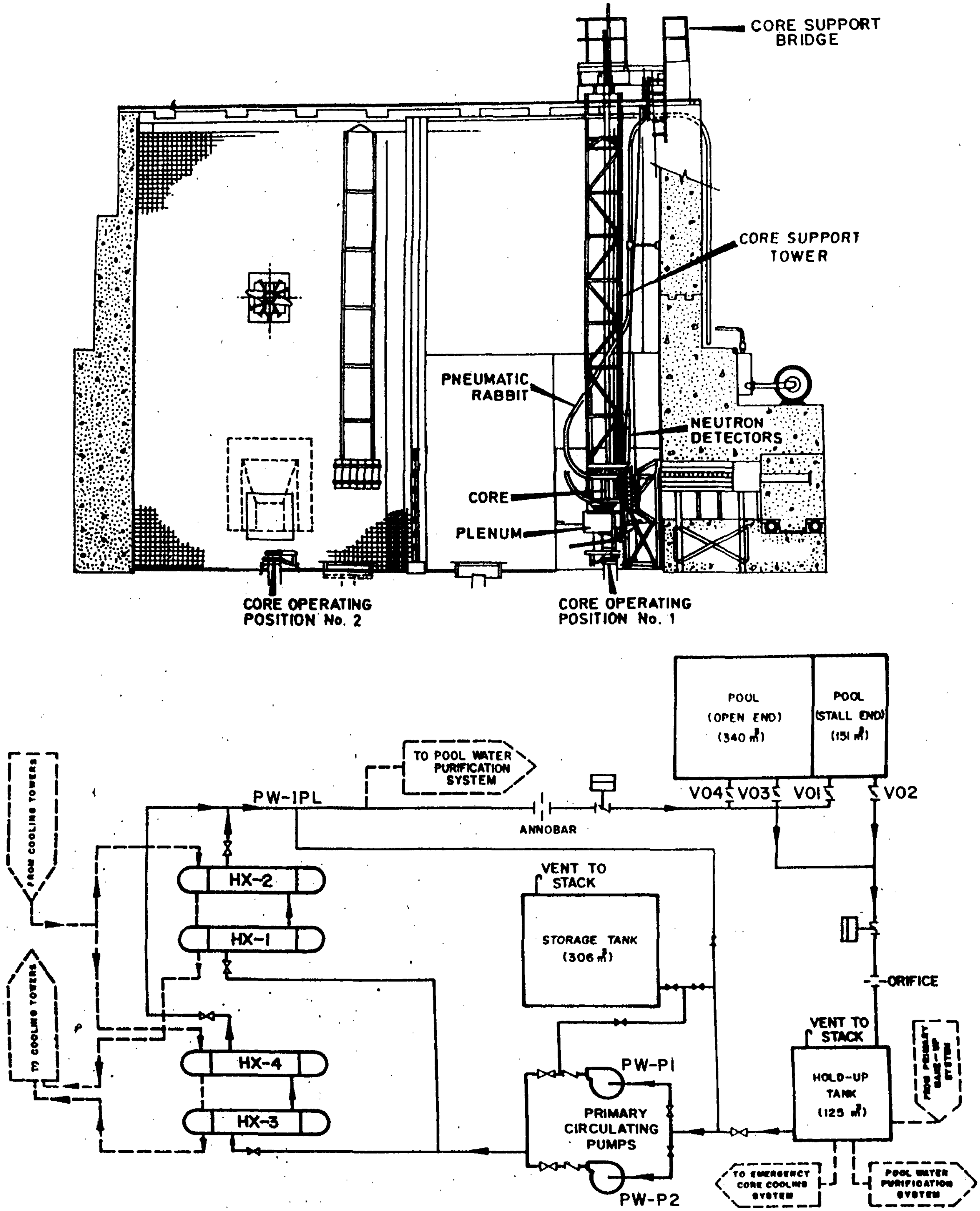

FIG. 1 PRIMARY COOLING SYSTEM AND VERTICAL SECTION OF PARR-1 (9 MW)

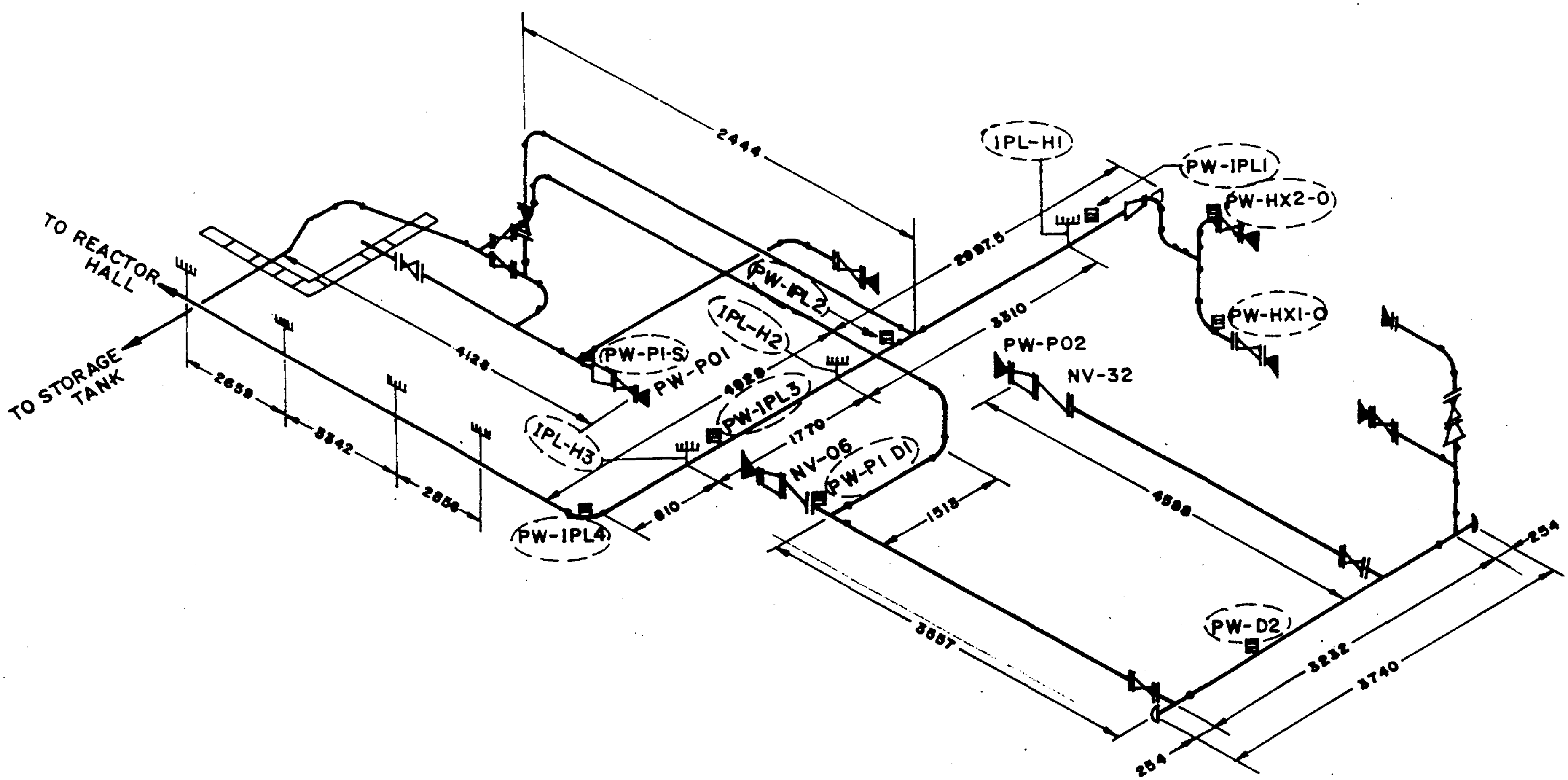

FIG.2 ISOMETRIC OF PRIMARY WATER INLET PIPELINE (PW-IPL)

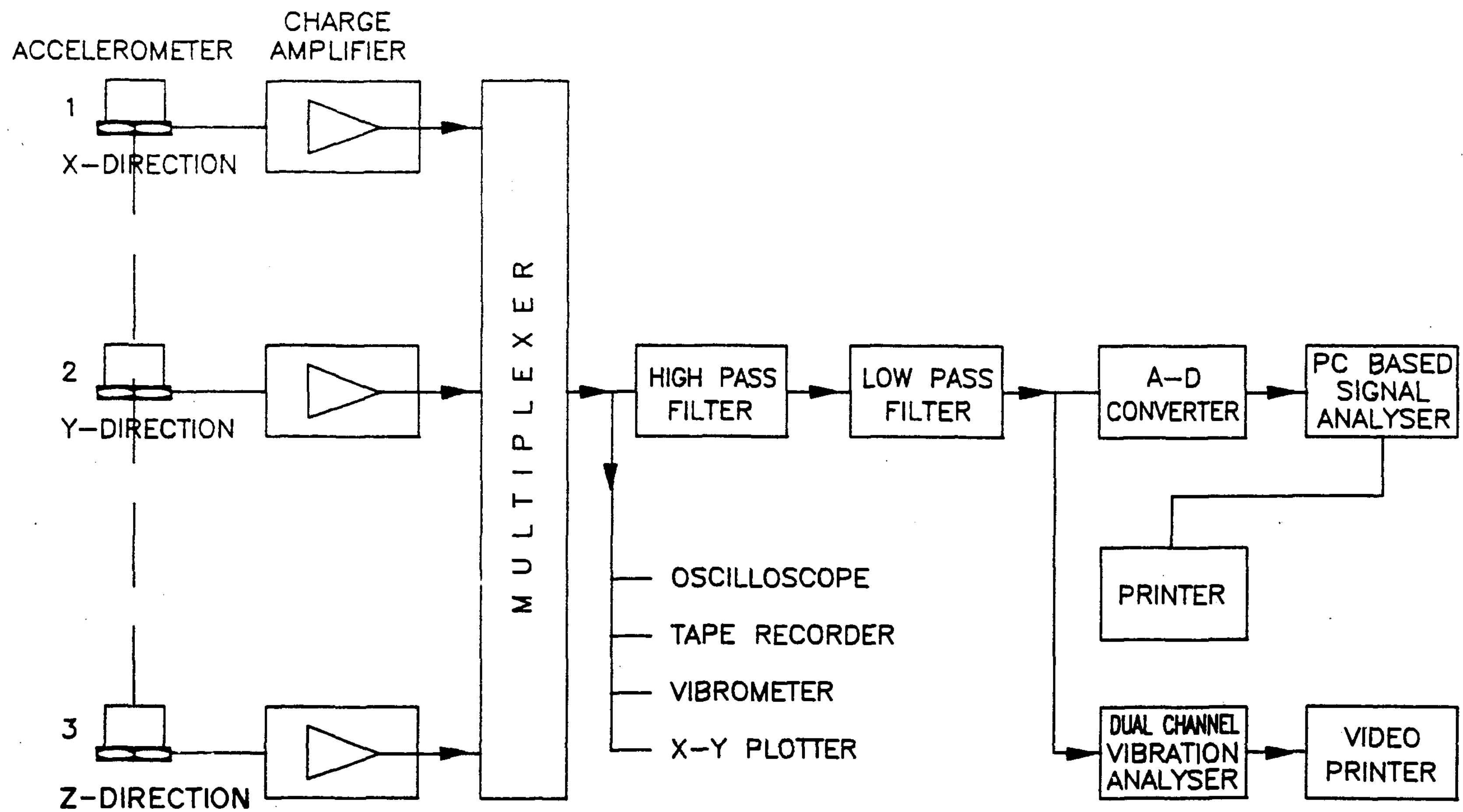

FIG. 3 EXPERIMENTAL SET-UP

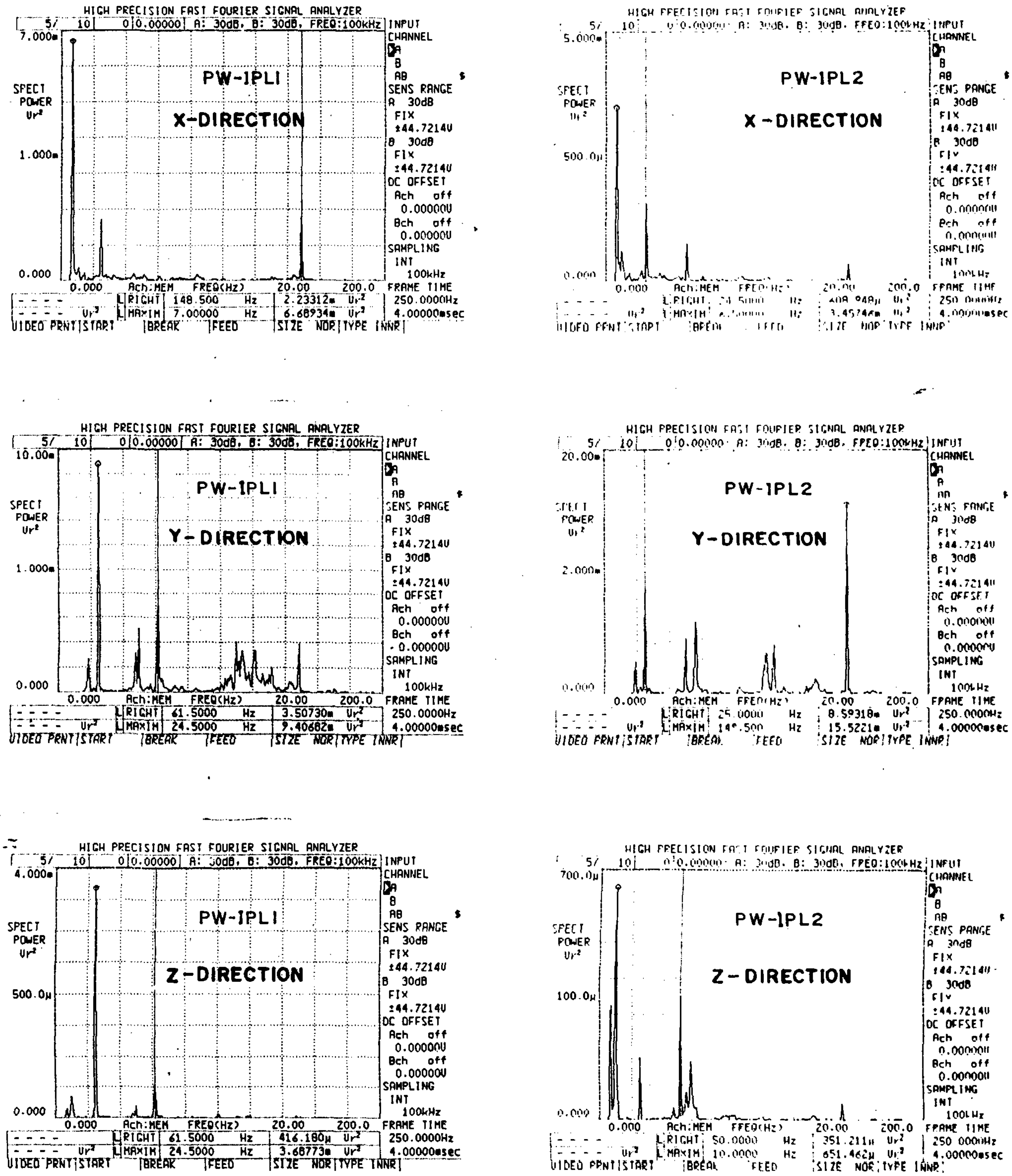

FIG. 4 **VIBRATION SPECTRA OF HANGER SUPPORTED TEST SECTION OF PW-IPL FROM TRIPLE-AXIS SENSOR AT LOCATIONS PW-IPL1 & PW-IPL2**

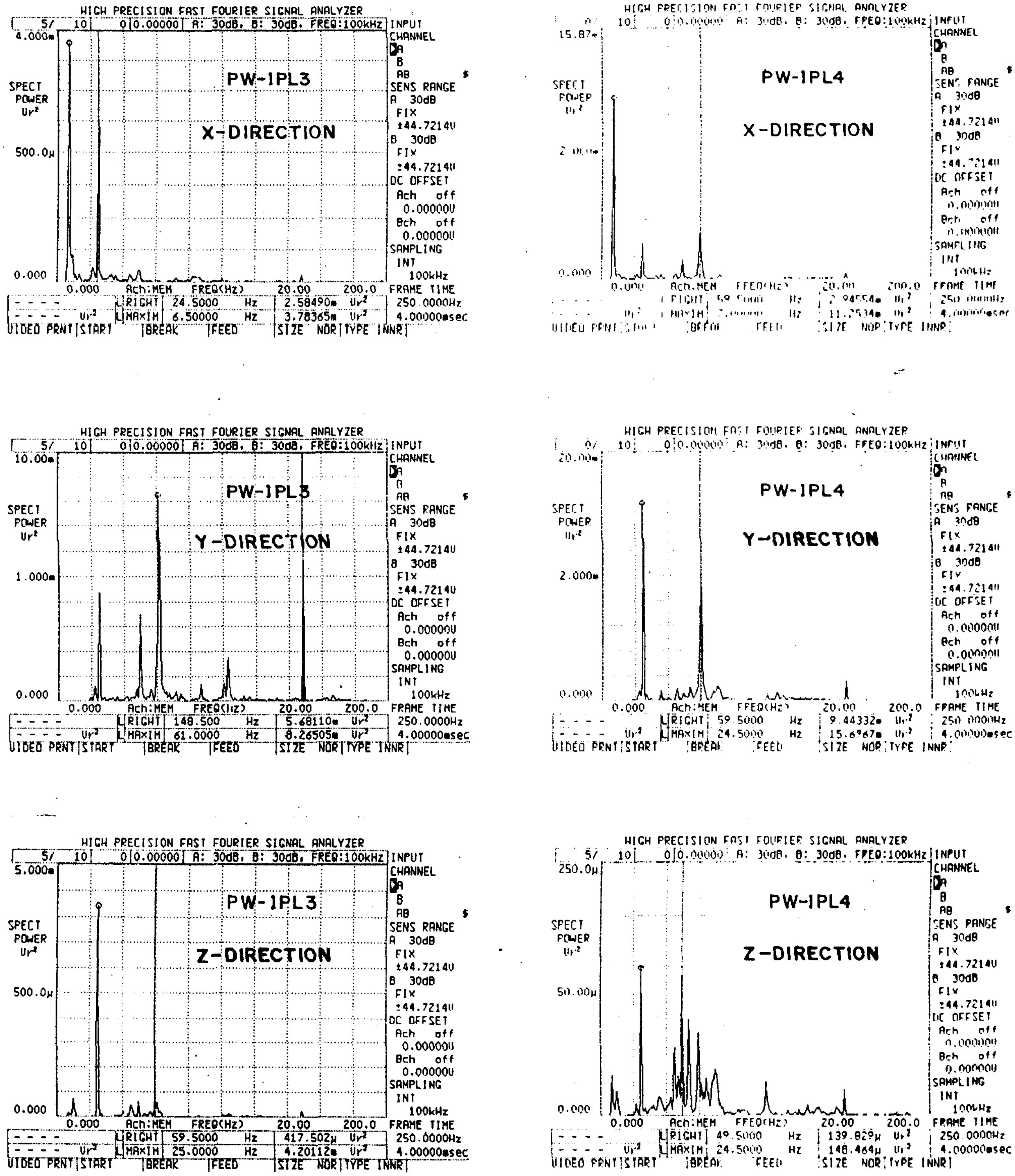

FIG. 5 VIBRATION SPECTRA OF HANGER SUPPORTED TEST SECTION OF PW-IPL FROM TRIPLE-AXIS SENSOR AT LOCATIONS PW-IPL3 & PW-IPL4

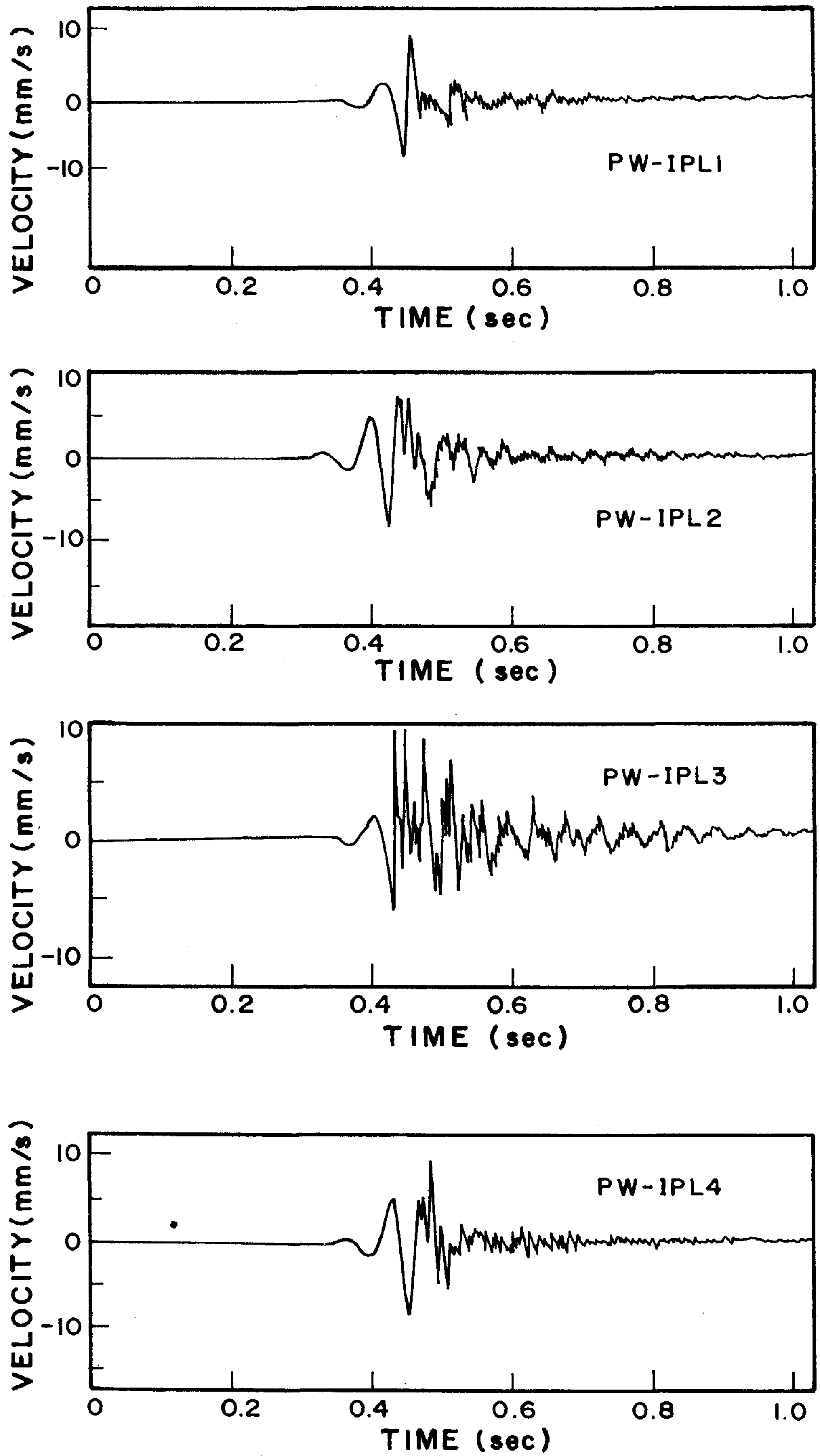

FIG. 6 REAL TIME SIGNAL OF ACCELEROMETER PLACED VERTICALLY AT DIFFERENT POSITIONS ON PW-IPL IN CASE NO. 1

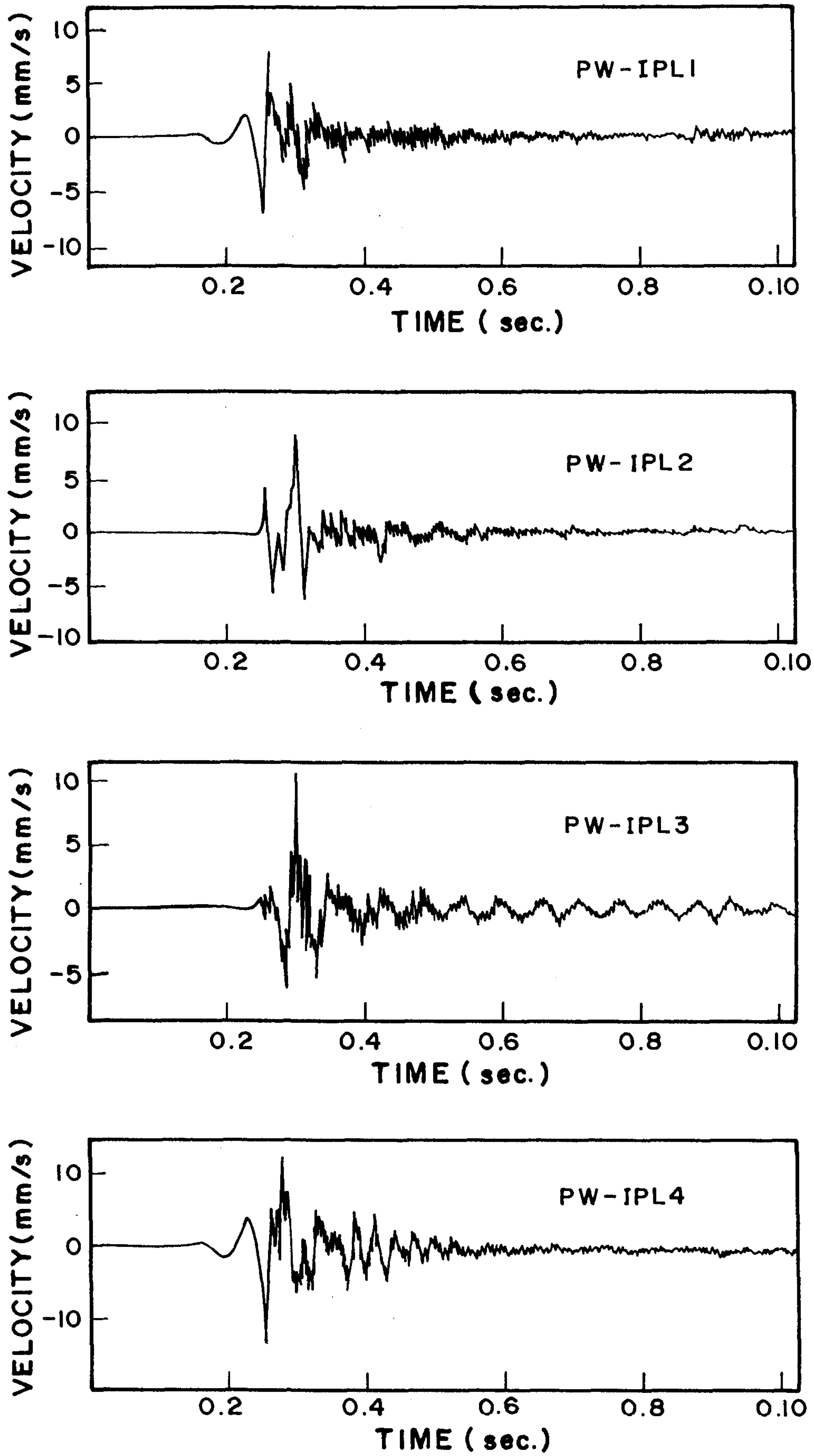

FIG. 7 **REAL TIME SIGNAL OF ACCELEROMETER PLACED VERTICALLY AT DIFFERENT POSITIONS ON PW-IPL IN CASE NO. 2**

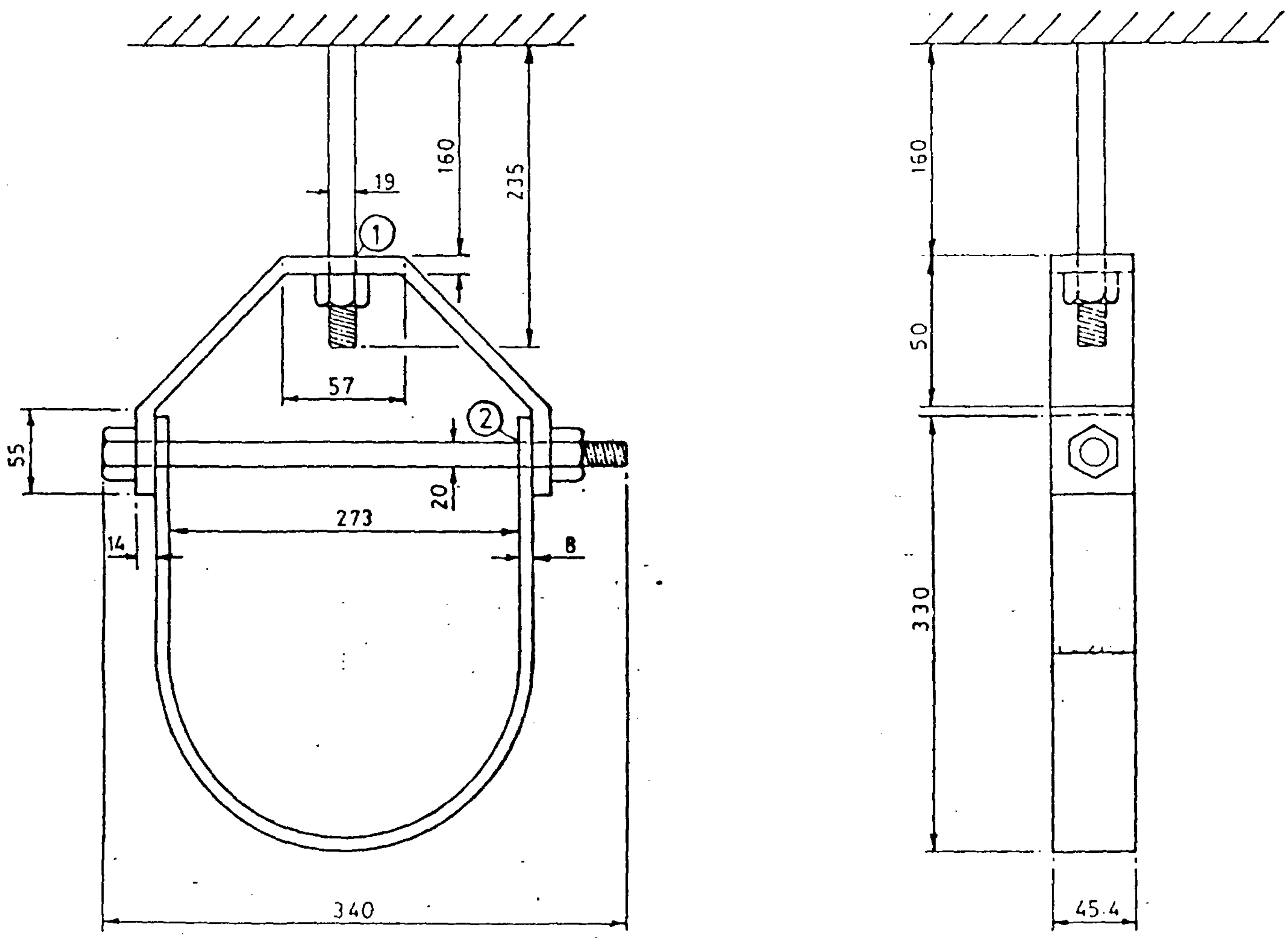

FIG. 8 PIPE SUPPORT HANGER