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Assessment of the integrity and functional requirement of moderator pump-motor units

Soni, R.S., Chawla, D.S., Dutta, B.K., Kushwaha, H.S., Mahajan, S.C., Kakodkar, A.
RED, Bhabha Atomic Research Centre, Trombay, Bombay, India

ABSTRACT: The design of various active components in a nuclear power plant calls for a satisfactory analysis of these components for various loadings from the point of view of safety because a designated number of these components must always remain functional. Presented herein is the structural and seismic qualification of one of the active components namely the moderator system pump-motor units for a typical PHWR.

1 INTRODUCTION

The moderator circulation in a typical 235 MWe PHWR is achieved through five moderator pump-motor units. These pumps are vertically mounted single stage centrifugal pumps. Each unit is an assembly of various parts namely pump stool, pump casing, stuffing box, mechanical seals, motor stool, motor frame and pump and motor shafts (Fig. 1). The flow rate in each of the pumps is 440.0 cu.meter/hour at 10.0 Kgf/sq.cm.

The pump casing, which is a double volute casing (with end suction and side discharge), is supported from bottom by the pump stool (Fig. 2). The pump stool in the horizontal plane is not a closed geometry due to the presence of an opening in it in the direction of inlet piping. Complete assembly is secured to the foundation by foundation bolts. The overall length and weight of the assembly are 3570.0 mm and 3148.5 Kgf respectively.

Two antifriction bearings viz the upper ball bearing to take care of radial and axial loads and the lower roller bearing to take care of radial loads, are provided on the motor shaft. In addition, two journal bearings separated by a lantern ring are also provided on the pump shaft just above the impeller for guiding the shaft and minimising the D2O leakage. Two kinds of analyses have been performed to assess the functional operability and structural integrity of these pumps:

(i) Structural qualification of pump casing.
(ii) Seismic qualification of pump-motor unit.

2 STRUCTURAL QUALIFICATION OF PUMP CASING

Due to lack of symmetricity, it is considered essential to prepare a 3-D model of the pump casing. Eight-noded brick and six-noded prism elements are used. Incompatible modes in the shape functions have been considered in 3-D elements to improve their bending behaviour. The casing is modelled by dividing the whole configuration into 39 different two dimensional sections radially. At the bottom of the casing, the pump stool is modelled using eight-noded brick elements, six-noded prism elements and thin plate and shell bending elements. The complete finite element model of moderator pump casing alongwith the stool contains 3740 nodes (Fig. 3).

The analysis is carried out for each of the five pump casings separately for various loads using finite element computer code (NISA 1990). The various loads considered for the analysis are the internal pressure of 0.14 Kg/sq.mm, the piping loads acting on discharge nozzle and suction nozzle, the loads due to motor and motor stool on the top of the pump casing, bolt loads and seismic inertial loads (Chawla 1992). The load combinations have been carried out in accordance with ASME section III Division 1 (ASME 1989).

The maximum stress intensities in different regions of the pump casing for each of the pumps have been calculated. Fig 4 and 5 show the stress intensity contours at some typical locations for design loadings. It has been observed that the stress intensities in all the regions of the pump casing for each pump are well below the allowable stress intensity given in NC-3200. The allowable membrane stress (Sm) of the material is 12.0 Kg/sq.mm. The maximum stress intensity due to design loads is found to be 9.9 Kg/sq.mm in the stool, which is less than the allowable membrane stress intensity (Sm) value of the material. Also the displacements at different locations of pump casing are small and within the available tolerances.

3 SEISMIC QUALIFICATION OF PUMP-MOTOR UNIT

Lumped mass beam modelling of the assembly is carried out by using beam, pipe and truss elements. Fig.6 shows the overall finite element model of the pump-motor assembly which consists of 103 number of nodes. The static and dynamic analyses of the assembly have been carried out by using the Finite Element computer code SAP-IV (Bathe).

Bearings have been modelled using stiffness and/or damping elements. The stiffness of anti-friction bearings have been evaluated using emperical relations which correlate the bearing stiffness with the force on the bearing, number of balls/rollers, angle of contact etc. The stiffness and damping coefficients of fluid-film journal bearings have been calculated using the non-dimensional charts available in the literature (Neelwarne 1992).

Seismic analysis of the assembly has been performed to assess the operability of the pump-motor assembly and to generate the loads at various locations. The analysis has been performed using the applicable floor response spectra for 3% damping during SSE (Fig.7).

The direction which contains the opening in the pump stool is referred here as "flexible direction" whereas the one which does not have this opening is referred as "rigid direction". The frequencies for the assembly are 11.4, 74.87 Hz in flexible direction, 22.58, 75.5 Hz in rigid direction and 168.5 Hz in vertical direction. The assembly is having its fundamental frequency as 11.4 Hz in the flexible direction which falls near to the peak in the applicable floor response spectra. This would mean higher seismic induced forces and therefore, it has been recommended to stiffen the pump stool in this direction. The increase in stiffness shall be such that the frequency is around 21 Hz. The seismically induced forces are shown graphically in the stationary as well as the rotating parts in Figs. 8 and 9 respectively for the flexible direction.

For the smooth functioning of the unit during SSE it is ensured that the total radial clearance available (75 microns) is more than the maximum induced relative deflection (71.1 microns) between the shaft and the bearing. So there will be no rubbing between the shaft and the bearing during an SSE event. Similarly, it has been found that the design clearance of 1.5 mm between the motor-rotor and the motor- stator is more than the induced relative displacement of 0.04 mm between them.

4 CONCLUSIONS

From the above analyses of pump-motor units, it can be concluded that they would remain functional during an SSE event and that the structural integrity of these units will be maintained under all loading conditions.

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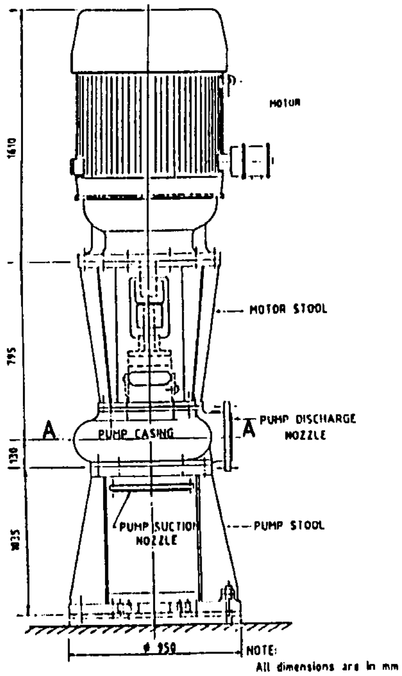

Fig. 1 General arrangement of pump-motor unit.

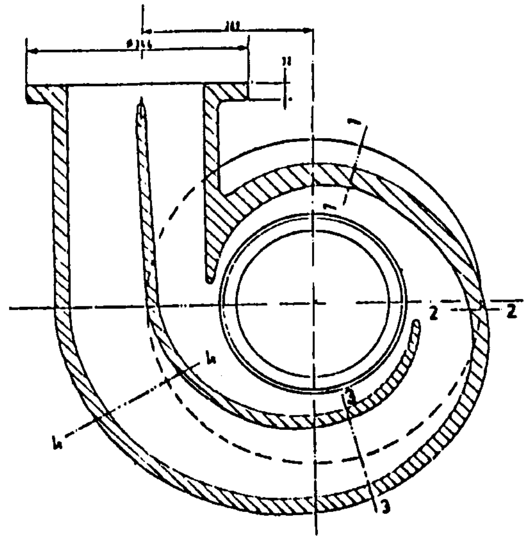

Fig. 2 Sectional view of pump casing at A-A

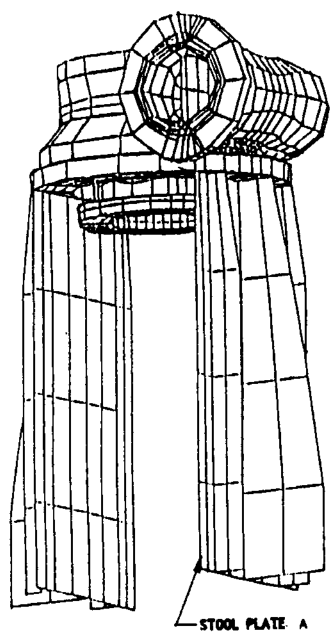

Fig. 3 Finite element model of pump casing.

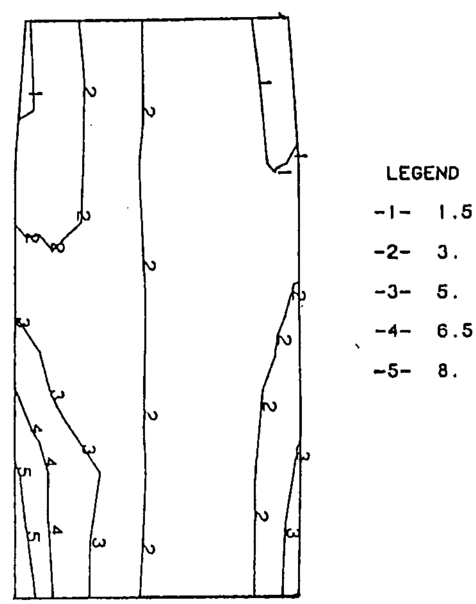

Fig. 4 Stress intensity (Kg./sq. mm.) contours in pump stool plate A.

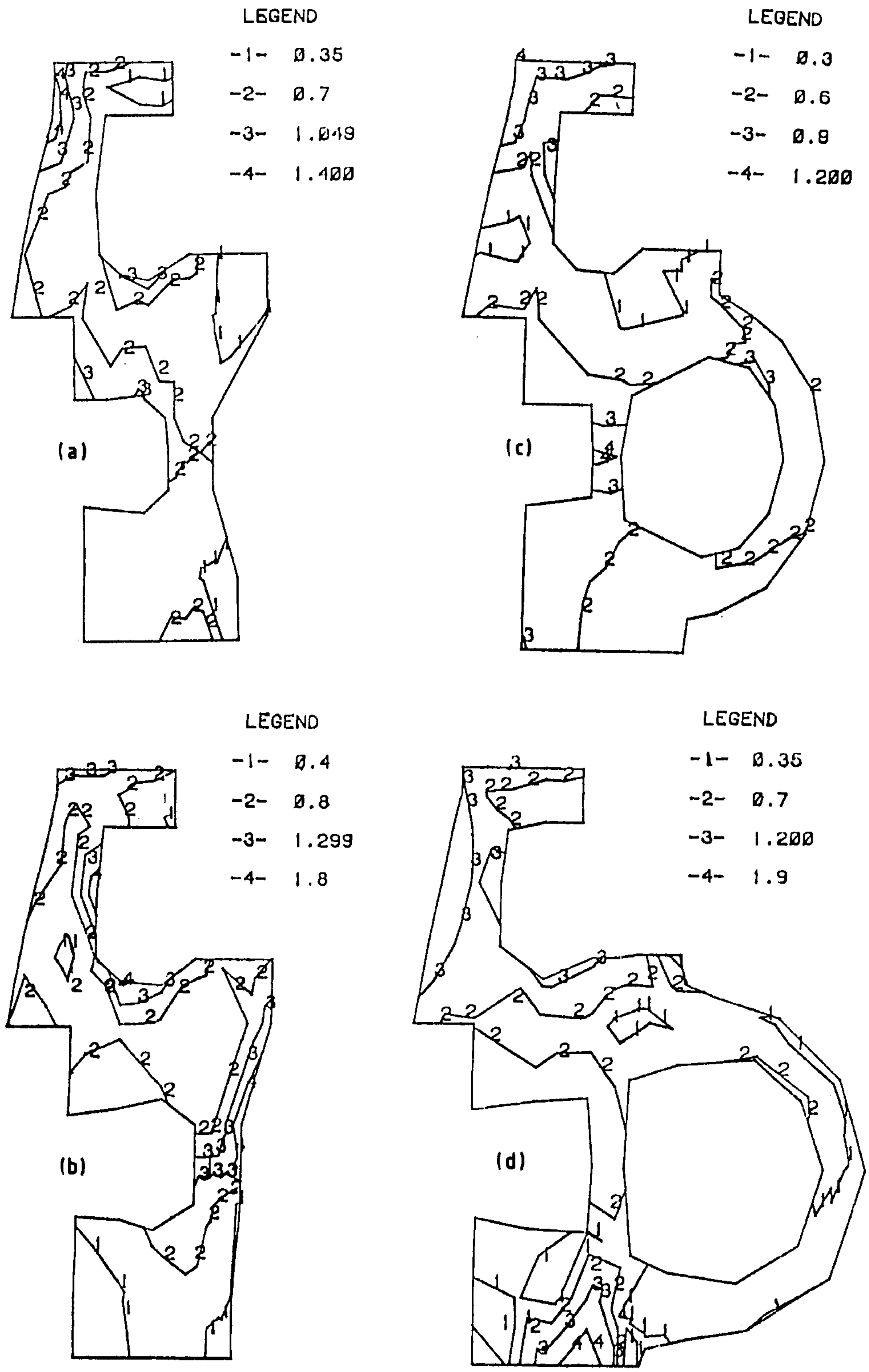

Fig. 5 Stress intensity (Kg./sq. mm.) contours in pump casing at section (a) 1-1 (b) 2-2 (c) 3-3 (d) 4-4 (Refer figure 5)

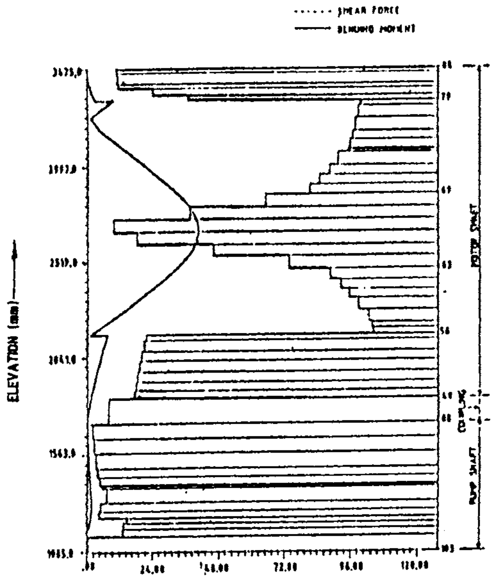

Fig. 9 Shear force (Kg.) and Bending moment (Kg.,/m) for rotating parts in flexible direction due to SSE

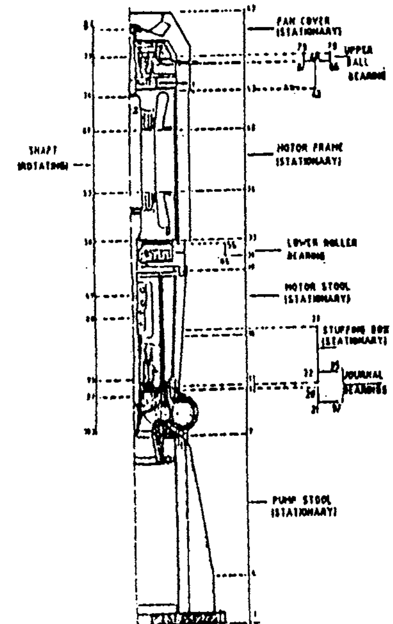

Fig. 6 Finite element model of pump-motor assembly.

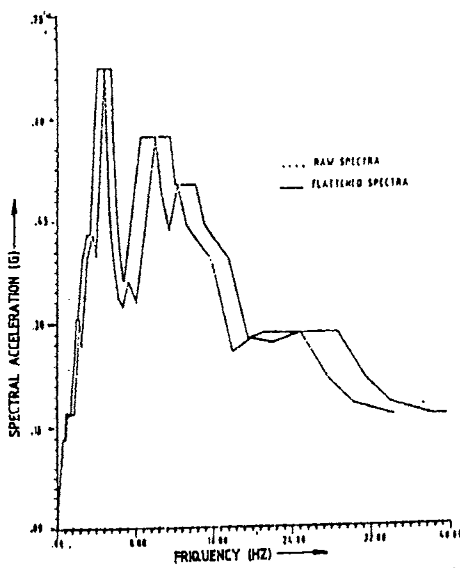

Fig. 7 Floor response for SSE-3% damping EW -95.1m floor.

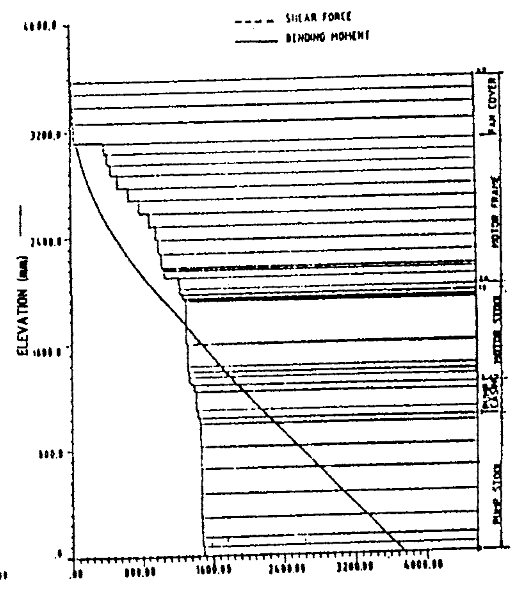

Fig. 8 Shear Force (Kg) & Bending moment (Kg-m) diagram for static parts in flexible direction due to SSE.