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DESCRIPTION

Consists of info concerning the stress analyses performed for the TMI-1 Decay Heat Pump Shafts.....

ENCLOSURE

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October 27, 1977
GQL 1477

Director of Nuclear Reactor Regulation
Attn: R. W. Reid, Chief
Operating Reactors Branch No. 4
U. S. Nuclear Regulatory Commission
Washington, D.C. 20555

Dear Sir:

Three Mile Island Nuclear Station, Unit 1 (TMI-1)
Docket No. 50-289
Operating License No. DPR-50
Decay Heat Pump Shafts

During a telephone conversation on October 19, 1977, between the NRC (Mr. Zwetzig) and Met-Ed (Mr. Stevens), the NRC asked Met-Ed to provide a summary of the stress analyses performed for the TMI-1 Decay Heat Pump Shafts. The NRC requested that the information applicable to 80 gpm operation be provided by October 26, 1977, and that information pertinent to operation at 135, 550, and 3,000 gpm be provided by November 2, 1977.

The information given below is primarily directed at answering the NRC questions regarding 80 gpm operation. However, information for the full flow range is also discussed to some extent to show trends for load and stress changes as a function of flow rate. Further information regarding operation at 135, 550, and 3,000 gpm will be provided in the November 2, 1977 submittal.

1. Steady-State and Alternating Loads for Bending and Torsion

   a. Bending Loads

      (1) Steady-State

      At a flow of 80 gpm, Worthington calculated a maximum steady-state radial load of 260 pounds acting at the centerline of the impeller (which corresponds roughly to the end of the shaft). Loads for higher flows will be less than this value. Worthington indicated that these load calculations are based on models developed by them using test data for their pumps.

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(2) Alternating

Worthington has indicated that there is no reliable method for accurately calculating actual alternating radial loads at low flow rates such as 80 and 135 gpm. However, an upper bound was established by MPR by calculating the radial load required to bend the shaft enough for the impeller and wear rings to contact. This maximum load was calculated to be 670 pounds acting at the centerline of the impeller.

b. Torsion

(1) Steady-State

The maximum steady-state torsional load occurs when the pump is drawing the maximum horsepower, i.e., at the maximum pump flow rate being considered (3,000 gpm). The horsepower at 3,000 gpm is 340 HP; this corresponds to a torque of 11,900 lb-in. At a flow rate of 80 gpm, the horsepower is approximately 150 HP, which corresponds to a torque of 5,250 lb-in.

(2) Alternating

With typical pumps, alternating torques are normally about 1% of steady-state torques. This would give very low torsional stresses, so an upper bound estimate of alternating torque was calculated by MPR by assuming that the full pump discharge head periodically acts across one impeller vane. This resulted in an upper bound alternating torque of 11,300 lb-in., i.e., essentially 100% of full power steady-state torque, and is much higher than expected to realistically occur. It is assumed that these upper bound loads could occur at either 80 or 135 gpm. but that they should not occur at flows over 20% of the rated flow, i.e., over 600 gpm.

2. Stress Concentration Factors

a. Bending

Per Page 108 of "Stress Concentration Design Factors", by R. E. Peterson, John Wiley and Sons, Inc., 1953, the stress concentration factor for this type of keyway in bending is 1.79.

b. Torsion

Per Figure 100 in "Stress Concentration Design Factors", the theoretical stress concentration factor in torsion for a sharp-edged keyway could be over 5. Therefore, a fatigue strength reduction factor of 5.0 was used, in accordance with ASME Section III Design Rules, which state that no fatigue strength reduction factor greater than 5.0 need be used.

3. Calculated Stresses for 80 gpm Operation

Using the above estimated loads and stress concentration/strength reduction factors, the following steady-state and alternating stresses are calculated to be acting at the keyway during operation at 80 gpm:

a. The steady-state bending stress due to a steady 260 pound load is 1,270 psi.

b. The alternating bending stress due to a fluctuating 670 pound load is 3,260 psi.

c. The steady-state torsional stress due to a 5,250 lb-in. steady torque is 9,270 psi. 1/

d. The alternating torsional stress due to a torque varying between 0 and 11,300 lb-in. varies from 0 to 19,960 psi (i.e., an alternating torsional stress of 9,980 psi). 1/

4. Comparison of Alternating Stresses with Fatigue Endurance Limit

The alternating bending and torsional stresses can be combined, and result in an effective alternating stress intensity of 21,660 psi. 2/ Per the ARMCO catalog, the endurance limit for 17-4 PH in the H1150 heat treat condition is 90,000 psi for both 10^7 and 10^8 cycles. Since the endurance limit does not change when going from 10^7 to 10^8 cycles, it is reasonable to conclude that a true endurance limit has been reached, and that 90,000 psi also applies to cycles over 10^8 cycles. Following standard ASME Section III practices, the allowable alternating stress range for this high cycle range would be a factor of 2 less than the best-fit tested value, i.e., would be 90,000/2 = 45,000 psi. Since the upper limit combined alternating stress of 21,660 psi is well below this value, no fatigue damage is calculated to occur.

1/ "Formulas for Stress and Strain", 4th Edition, McGraw Hill, Inc., Copy Right 1965, Page 194, Chapter 9, Table 9, Raymond J. Roark.

2/ "Formulas for Stress and Strain", 4th Edition, McGraw Hill, Inc., Raymond J. Roark, Copy Right 1965, Page 95, Table 2, Case No. 3 for axial stress combined with sheer.

5. Vibration Analyses

a. Bending

In the pump design review meeting of September 20, 1977, Worthington stated that the calculated critical speed for lateral vibrations is about 5,300 rpm. This speed is much higher than the nominal rotating speed of 1,800 rpm, and no problems are expected due to lateral vibrations. This was confirmed by the low vibrational levels detected during recent pump tests, as previously reported to the NRC.

b. Torsion

In the design review meeting of September 20, 1977, Worthington indicated that their experience has been that motor driven pumps which do not include a gear reducer are not subject to torsional vibration problems. Further, Worthington indicated that there are no realistic mechanisms for imposing significant torsional loads. Accordingly, torsional critical speed analyses have not been performed to date. Further, it is noted that the vibration measurements of the pump indicate that no significant vibrations are occuring. However, to provide further assurance that torsional critical speeds are not a problem, an analysis will be initiated and is expected to be completed by November 30, 1977.

c. Allowance for Resonance Effects

As discussed above, no specific allowance has been made for resonance effects. If the torsional critical speed is calculated to be close to some driving frequency, then possible resonance effects will be evaluated.

In addition to the request for the summary of stress analysis provided above five (5) questions were asked by the NRC during the telephone conversation of October 19, 1977 and are answered below:

1. How many spare shafts are on hand?

   Three (3); two (2) with documentation, one (1) without.

2. Do spare shafts have keyways of the proper radii?

   Yes, all three (3) shafts have 0.06 inch radii in the keyway.

3. (a) Would installation of spare shafts require shutdown?

   Under present Technical Specification, yes because of the estimated time for shaft replacement.

   (b) What is the time required to replace a shaft and restore the pump to its proper condition?

   Our best estimate is 70 hours for one pump providing no problems are encountered.

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(c) Are we looking at the most expeditious ways to make replacement? For example, pump replacement?

Yes, several ways to replace the pump shafts have been considered including pump replacement. The answer to 3c. and b. above are a result of considering these options.

4. Our October 15, 1977 letter refers to replication and hardness testing. Is this for the installed or spare shafts?

It is for the installed shafts. In our letter of October 5, 1977, GQL 1368, Met-Ed indicated its intention to investigate the feasibility of metallographic replication and hardness testing to verify the heat treated condition of the installed TMI Decay Heat Pump Shafts. We have scheduled, concurrent with the vibration and UT testing during the first week of November, the following tests to verify to the extent possible, the heat treated condition of the installed shafts.

a. A BHN Hardness Test will be performed on both shafts at the pump coupling end.

b. A Resistivity Measurement Test will also be conducted on both shafts with a "TEVO" tester.

c. A Magnetic Permeability - Eddy Current Tests will be performed.

The results of the above three tests should give us a good indication of the heat treat condition of the existing shafts.

5. What is Met-Ed's specific proposal for shaft replacement, orifice removal, and "piggy back" operation?

Based on the results of the three tests described in 4. above, Met-Ed will evaluate both long term operation with the present shafts and the advisability of scheduling shaft replacement during the 1978 refueling. Since removal of the recirculation line orifices can be accomplished without requiring unit shutdown it has been scheduled during November as a routine maintenance item. No change to the long term recirculation cooling (piggy back) mode of operation is deemed necessary or planned as a result of this current analysis of the decay heat pump shaft adequacy.

As a result of a meeting with the NRC on September ' 1977, Met-Ed committed to accomplish vibrational testing and ultrasonic inspection of the installed decay heat pump shafts on a monthly basis. This commitment was documented in GQL 1234 dated September 8, 1977. Subsequent testing and inspection results have all met the acceptance criteria agreed to by Met-Ed and the NRC.

Based on the excellent test and inspection results, information resulting from the evaluation of pump design and decay heat system design which is documented in GQL 1368 dated October 5, 1977, and summarized above, and discussions with the NRC by telephone on October 19, 1977 Met-Ed requests that the commitment be modified from a monthly inspection and test as follows:

a. Conduct quarterly vibration testing and ultrasonic inspection with an allowance of ± 25% of the inspection interval. The inspection interval allowance has been established consistent with TMI-1 Technical Specification Section 4 and current regulatory practices.

b. Conduct vibration testing and ultrasonic inspection when the decay heat pumps are shutdown after accumulating in excess of one hour run time in any mode or combination of modes. This inspection will commence within five working days of pump shutdown.

This inspection frequency is to commence with the first test being conducted during the first week in November. All parameters of the vibration test and ultrasonic inspection except frequency of the inspection are to remain the same as those specified in GQL 1234, dated September 8, 1977.

Sincerely,

J. G. Herbein

J. G. Herbein
Vice President

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