Fatigue evaluation of 220MWe PHWR coolant channel seal disc

Gupta S.K.(1), Chawla D.S.(2), Bhambra H.S.(1), Kushwaha H.S.(2)
(1) Nuclear Power Corporation, India
(2) Bhabha Atomic Research Centre, India

ABSTRACT

The coolant channel seal disc of Pressurized Heavy Water Reactor (PHWR) is part of sealing plug and is used to close the two ends of 306 coolant channels and prevent escape of heavy water. The sealing plugs are removed (and then reinstalled) on-power from coolant channel for unloading/loading of the fuel. The seal disc is subjected to cyclic loads due to these operations together with reactor pressure/temperature transients. The fatigue analysis of seal disc as per ASME Sec.III Div.1 NB has been carried out using axisymmetric finite element model to evaluate cumulative usage factor.

1 INTRODUCTION

The coolant channel seal disc is a part of sealing plug assembly. It is used to close the ends of coolant channel and prevent escape of heavy water. Two special on power fuelling machines are used to remove the sealing plug from coolant channel for unloading/loading of the fuel and to reinstall it back into the channel. The importance of seal disc can be understood by the fact that there are 612 seal discs in one 220 MWe PHWR unit. During the life time of reactor the seal disc will be subjected to cyclic loads due to startup, shutdown, power setback and also due to refuelling operations. Excessive reversal of stresses may lead to fatigue failure. The seal disc failure will cause loss of coolant accident. Therefore, seal disc is categorised as safety class 1 component and it has to be qualified according to ASME Section III Division 1 NB[1]. For cyclic type of loads, the fatigue analysis is essential to assess the allowable number of cycles and also to check the total usage factor due to different fluctuating loads. To evaluate the allowable fatigue cycles, the analysis is carried out using finite element method[2].

2 MATERIAL AND GEOMETRY

The seal disc is a thin integrally forged 17-4 PH SS flexible disc, heat treated to H 1075 condition. Its geometry is shown in Fig.2. The seal disc has typical variation in its thickness in radial direction, with thick hump at rocker point and a central stem for handling. The thermal and mechanical properties of seal disc material are taken from ASME[1]. A 1 mm
thick annular soft Nickel gasket is electro deposited near the outer periphery of seal disc face in contact with the endfitting and provides good sealing. Fig. 1 shows the location of seal disc in the coolant channel. The seal disc seats on the lapped face of endfitting. A small radial width of 0.5 mm near the outer periphery of seal disc at diameter of 94,996 mm makes contact with endfitting surface and forms the sealed pressure boundary.

3 DETAILS OF CYCLIC LOADS

The Seal disc will be subjected to cyclic loads due to pressure/temperature cycles arising from start-up, shut-down, power set back etc. & the mechanical loads arising from refuelling operation.

Various PHT pressure temperature transients (Table 1) for normal (level A service limit) and abnormal (level B service limit) condition are:

(i) Start up from cold and shut down to cold.
(ii) Trip and restart before poison out i.e system at full power to hot shut down and restart.
(iii) Trip poison out and restart.
(iv) Crash cool down.
(v) Crash cool down due to normal feed supply failure to Steam Generator.

Two special on-power fuelling machines (F/M) are used to install or to remove the sealing plug into/from the coolant channel before commencing loading/unloading of fuel. Pressure is equal on both side of fuelling machine when fuelling machine is clamped to end fitting for refuelling. A specific loading sequence (Fig.4) is adopted during removal and installation of sealing plug [3,4]. The various loading stages are:

Stage 0: Unstressed condition with equal pressure on both sides of seal disc
Stage 1: Load is applied at centre of disc after contact with end fitting
Stage 2: Load of 1814 Kg applied at centre and rocker point of disc
Stage 3: Locking of stage 2 displacement
Stage 4: Fuelling machine is disengaged from channel. Thus, the pressure on F/M side of installed seal disc becomes atmospheric while channel pressure is acting on other side.

Stage 5: a load of 3089 kg applied at centre and rocking point to unload the plug mechanism.

Sealing plug leak test: Pressuring F/M side to 11.25 Kg/sq.cm. & 38 deg.C. as against channel side 102 Kg/sq.cm. & 293 deg.C.

The loading sequence during installation is: Stage 0--stage 1--stage 2--stage 3--sealing plug leak test--stage 4.

The loading sequence during removal is: Stage 4--stage 3--stage 5--stage 0.

4 FINITE ELEMENT MODEL

Axisymmetric finite element model (Fig.3) consisting of 671 nodes and 188 eight noded axisymmetric elements is used. The mesh convergence study was carried out and it was found that the refinement of mesh does not show any appreciable change in results[3]. The finite element based results are also compared with experimental stress analysis results [5] and agreement is found to be satisfactory. This validates the finite element mesh used in the analysis. The contact location and width of the support on the endfitting is determined by measuring the impression of contact surface experimentally[3]. The finite element computer code NISA [6] is used for the thermal and stress analysis.
5 THERMAL ANALYSIS

The analysis of seal disc is carried out in 3 phases viz. thermal analysis, stress analysis and fatigue cycle and usage factor computation. In the first phase, thermal analysis is carried out to determine the temperature distribution in the seal disc for various temperature transients.

The F/M side of seal disc is subjected to ambient air temperature of 38 deg.C. The heat transfer by natural convection is considered on F/M side of seal disc. The channel side surface of seal disc is in contact with primary heat transport coolant heavy water. The coolant flows from feeder pipe on upstream end into the annulus between end fitting and liner tube and thereafter through holes in the liner tube and holes in the orifice of shielding plug towards the fuel bundles (Fig.1). After flowing over the fuel bundles the coolant returns through holes in the orifice of shielding plug and holes of liner tube on downstream end and thereafter flows out through annulus between liner tube and end fitting into the feeder pipe. The natural convection heat transfer on channel side is considered since the coolant flow near the seal disc is stagnant due to following reasons.

(i) Seal discs act as a blind end for coolant flow.
(ii) The annulus opening across shield plug is restricted and is approximately 1 m long.
(iii) The distance between orifice of shielding plug and seal disc face is approx. 1.5 m long.

The heavy water in contact with seal disc is conservatively assumed to be at same pressure and temperature as channel heavy water. The convective heat transfer coefficients on channel side and on F/M side is computed from average temperature of Heavy water (Table 1) Galerkin time integration scheme is used in the analysis. A small time step is chosen to obtain accurate solution accuracy. For each transient the time step was selected separately.

The typical isotherms in seal disc for typical transient are shown in fig.5. Fig.6 shows the temperature history of few nodes for transient C. Different sections and locations of nodes are shown in figure 3. Since seal disc is very thin plate, the temperature variation along the thickness is negligible for most portion of the seal disc. The variation in temperature with thickness is noted at and near the stem of seal disc.

6 STRESS ANALYSIS

6.1 Analysis for pressure/temperature transients

Stress analysis, for various pressure-temperature transients are carried out. During these transients, the seal disc is in installed condition (stage 4). The seal disc is supported by end fitting at diameter of 94.996 mm, while rocker point is in locked condition having deflection of 0.195 mm. It is assumed that seal disc can have free thermal expansion in radial direction. The friction between nickel gasket and end fitting face and also between rocker point and seal plug body is not considered. These assumptions do not cause any appreciable change in the results, since the computed stresses match nicely with those found experimentally[5].

The linear elastic analysis is carried out at various time instants for different transients. The computed temperatures from thermal analysis are used for stress analysis. The pressure variation is also considered. The pressure on the F/M side is atmospheric.

Fig.7 shows the stress intensity contours in seal disc at typical time instants for one of the temperature pressure transients. The stress variation with time at selected location is shown in Fig.8 for transient E. At typical time instant, the comparison of stress intensity distribution at different sections is shown in Fig.9. It can be seen that stresses are higher at section 1-1 & 2-2 as compared to section 3-3 & 4-4 of seal disc.
It is seen that major contribution of stress variation is due to pressure loading. The thermal stresses are quite less in magnitude. This is because the temperature difference across two surfaces of seal disc is very small since seal disc is very thin. For example in pressure temperature transient C, there is sudden drop in pressure and then sudden rise back to same pressure. It can be seen from fig. 8 that the stress level comes down due to fall in pressure and stress level also picks up once the pressure rises to previous level. The pressure loading tries to deflect seal disc towards F/M side, while the thermal expansion of seal disc tries to deform seal disc in the opposite direction. A typical stress distribution for various thermal cycles can be seen for section 3-3 in fig.8. The maximum bending stress occurs at thinnest part of seal disc. The stress intensity variation with time is shown in Fig. 10 and 11.

6.2 Analysis for refuelling loads

The stress analysis of seal disc is also carried for various stages of loading during its installation and removal [3,4]. During installation and removal, seal disc is also subjected to thermal stresses due to hot PHT coolant at 293 deg.C on channel side and heavy water of 38 deg.C on Fuelling machine side. Analysis are carried out considering these thermal conditions along with mechanical loads. Figure 12 shows the stress intensity contours for stage 4 loading during refuelling.

7 FATIGUE CYCLES AND USAGE FACTOR COMPUTATION

The maximum fluctuating stress intensities are determined for different pressure temperature transient and refuelling loads. The allowable fatigue cycles are evaluated using the fatigue curve I-9-2 of ASME section III Division 1 Appendices[1].

The usage factor is calculated by using the actual number of cycles and allowable fatigue cycles for various pressure temperature transient and also refuelling loads. The total usage factor of seal disc is found as .016 which is much less than unity.

8 CONCLUSION

The fatigue analysis of Indian 220 MWe PHWR seal disc is carried out for pressure/temperature transients and refuelling loads. It is found that the total usage factor is much less than unity.

REFERENCES


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### TABLE 1 DIFFERENT TEMPERATURE-PRESSURE TRANSIENTS

<table>
<thead>
<tr>
<th>S.N.</th>
<th>TEMPERATURE AND PRESSURE TRANSIENTS</th>
<th>cycles</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 (a)</td>
<td>TEMP(C) 44 (0 Hrs)--44 (0.2 Hrs)--263 (2.5 Hrs)</td>
<td>300</td>
</tr>
<tr>
<td></td>
<td>263 (8.5 Hrs)--272 (8.7 Hrs)--272 (12.7 Hrs)--278 (12.9 Hrs)--278 C (16.7 Hrs)--284 C (16.9 Hrs)--284 (20.7 Hrs)--289 (20.9 Hrs)--289 (24.7 Hrs)</td>
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<tr>
<td></td>
<td>293.5 (24.9 Hrs)</td>
<td></td>
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<tr>
<td></td>
<td>PRES (Kg/Sq.Cm) 0.22 (0 Hrs)--22 (0.2 Hrs)--100 (0.2 Hrs)--100 (24.9 Hrs)</td>
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<tr>
<td></td>
<td>(b) TEMP(C) 293 (0 Hrs)--284 0.06 Hrs)--284 (2.06 Hrs)--264 (2.1 Hrs)--116 (3.9 Hrs)--44 C (6.06 Hrs)</td>
<td>600</td>
</tr>
<tr>
<td></td>
<td>PRES. (Kg/Sq.Cm) 100 (0 Hrs)--100 (6.06 Hrs)--22 (6.1 Hrs)</td>
<td></td>
</tr>
<tr>
<td>2 (a)</td>
<td>TEMP(C) 293 (0Mins.)--262.2 (0.1Mins.)--262.2 (22Mins)</td>
<td></td>
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<tr>
<td></td>
<td>PRES. (Kg/Sq.Cm) 100 (0 Mins)--60 (0.1 Mins)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>100 (2 Mins)--100 (22 Mins)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>(b) TEMP 262.2 C (0Mins)--293 C (10 Mins)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>PRES. (Kg/Sq.Cm) 100 (0 Mins)--100 (10 Mins)</td>
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<tr>
<td>3 (a)</td>
<td>TEMP(C) 293 (0 Hrs)--262.2 (.03 Hrs)--200 (2 Hr)--200 (36 Hrs)</td>
<td>300</td>
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<td></td>
<td>PRES. (Kg/Sq.Cm) 100 (0Hrs)--60 (0.016 Hrs)--100 (0.033 Hrs)--100 (36 Hrs)</td>
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<tr>
<td></td>
<td>(b) TEMP(C) 200 (0 Hrs)--262.2 (0.68 Hrs)--262.2 (2.68 Hrs)--293 (2.83 Hrs)</td>
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<tr>
<td></td>
<td>PRES. (Kg/Sq.Cm) 100 (0 Hrs)--100 (2.83 Hrs)</td>
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<tr>
<td>4 (a)</td>
<td>TEMP(C) 293.4 (0 Minutes)--262.2 (.1 Minutes)--120 (4.8 Minutes)--38 (21.4 Minutes)</td>
<td>45</td>
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<tr>
<td></td>
<td>PRES (Kg/SqCm) 100 (0 Mins)--60 (.1 Mins)--100 (.5 Mins)--100 (18.9 Mins)--22 (19 Mins)--22 (21.4 Mins)</td>
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<tr>
<td></td>
<td>Same as 1 (b)</td>
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</tr>
<tr>
<td>5 (a)</td>
<td>TEMP(C) 293.4 (0 Mins)--262.2 (0.1 Mins)--121 (4.6 Mins)--38 (24 Mins)</td>
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<tr>
<td></td>
<td>PRES. (Kg/Sq.Cm) 100 (0 Mins)--55 (0.1 Mins)</td>
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<td></td>
<td>22 (4.6 Mins)--0 (24 Mins)</td>
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<td>Same as 1(b)</td>
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### TABLE 2: USAGE FACTOR EVALUATION FOR SEAL DISC

<table>
<thead>
<tr>
<th>Type of fluctuating load</th>
<th>Actual no. of cycles n</th>
<th>Maximum S.I. RANGE (Kg/mm²)</th>
<th>Fluctuating S I (Kg/mm²)</th>
<th>Fatigue cycles N</th>
<th>Usage factor n/N</th>
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<tr>
<td>press.temp cycle 1</td>
<td>300</td>
<td>36.19</td>
<td>18.09</td>
<td>1000000</td>
<td>0.00003</td>
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<td>press.temp cycle 2</td>
<td>600</td>
<td>13.03</td>
<td>6.52</td>
<td>1000000</td>
<td>0.00006</td>
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<tr>
<td>press.temp cycle 3</td>
<td>600</td>
<td>32.84</td>
<td>16.42</td>
<td>1000000</td>
<td>0.00003</td>
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<td>press.temp cycle 4</td>
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<td>36.14</td>
<td>18.07</td>
<td>1000000</td>
<td>0.00004</td>
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<td>press.temp cycle 5</td>
<td>10</td>
<td>48.53</td>
<td>24.27</td>
<td>2000000</td>
<td>0.00001</td>
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<tr>
<td>Refuelling cycle 1</td>
<td>80</td>
<td>85.54</td>
<td>42.77</td>
<td>55000</td>
<td>0.0145</td>
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<tr>
<td>Refuelling cycle 2</td>
<td>80</td>
<td>43.48</td>
<td>21.74</td>
<td>250000</td>
<td>0.0003</td>
</tr>
</tbody>
</table>

Cumulative usage factor = 0.016

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FIG. 1 SCHEMATIC OF COOLANT CHANNEL.

FIG. 2 GEOMETRY OF 220 MWe PHWR SEAL DISC.

FIG. 3 FINITE ELEMENT MODEL OF SEAL DISC.

FIG. 4 DIFFERENT STAGES OF LOADING DURING REFUELLING.

LEGEND:

→ LOADING

← SUPPORT
Temperature (deg C)
8  290.3
7  289.8
6  289.2
5  288.7
4  288.1
3  287.6
2  287.0
1  286.5

FIG. 5 TEMPERATURE CONTOURS FOR TRANSIENT C AT TIME .003 HRS.

FIG. 6 TEMPERATURE HISTORY OF SELECTED NODES FOR PRESSURE TEMPERATURE TRANSIENT C.

Stress intensity
5  50.0
4  40.0
3  30.0
2  20.0
1  10.0

FIG. 7 STRESS INTENSITY (Kg/sqmm) CONTOURS FOR TRANSIENT C AT TIME .001 HRS.
FIG. 8 STRESS DISTRIBUTION ALONG SECTION 3-3 FOR PRESSURE TEMPERATURE TRANSIENT E

FIG. 9 STRESS DISTRIBUTION ALONG DIFFERENT SECTION FOR PRESSURE TEMPERATURE TRANSIENT E AT TIME = 0.002 Hrs

FIG. 10 STRESS VARIATION DURING DIFFERENT TRANSIENTS AT NODE 389

FIG. 11 STRESS VARIATION DURING DIFFERENT TRANSIENTS AT NODE 389

Stress intensity
5  55.5
4  44.4
3  33.3
2  22.2
1  11.1

FIG. 12 STRESS INTENSITY (Kg/sqmm) CONTOURS FOR STAGE 2 OF REFUELING OPERATION.