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TRANSIENT THERMAL ANALYSIS OF CENTRIFUGAL PUMP

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ABSTRACT

Pumps at times may be required to undergo critical flow conditions wherein thermal and pressure transients are part of operating conditions. Under such circumstances, a guideline from ASME code is followed to have confidence in design.

High operating temperature affects the design in two ways. The physical properties change drastically with elevated temperature and other important aspect is if it is accompanied by high pressure, the strength of the component need to be checked. The thermal stresses together with stresses developed by mechanical loads like pressure, forces, moments when combined; need to be compared with strength as suggested in ASME Codes.

The situation becomes far more complex when the temperature and pressure levels are fluctuating with time. The transient conditions of temperature and pressure of short duration will generate thermal shock which will further produce very high thermal stress.

A numerical approach is presented in this paper to judge suitability of design under transients in thermal and mechanical loads. The design code followed is ASME Section III Subsection NB. The transients are considered at 126 bar pressure and 350 °C temperature.

KEYWORDS

Finite Element Analysis, Thermal Stress, stress intensity.

INTRODUCTION

Development of special purpose pumps operating at transient temperature and pressure conditions poses challenges for designer. These conditions are unique and demands for special pumps. Depending upon certain applications, pump may undergo transient nature of temperature and pressure. In such application, even mechanical seal arrangement to be used becomes critical.

Mechanical Strength of components operating at high temperature needs to be checked. Stresses induced due to temperature change are called as thermal stresses [2]. Sudden change in temperature condition generates very high thermal stresses. Thermal stresses together with stresses developed by mechanical loads like pressure, forces, moments, self weight etc. leads to very critical situation.

The paper presents an approach to handle this complex situation. The thermal analysis is carried out at steady state [1] temperature to evaluate temperature distribution of pump geometry. The geometry of the pump has been modified in order to reduce the temperature near the pump seal location. Stress analysis of pump geometry is carried out for steady state and transient operating conditions. Analysis results are evaluated as per ASME Section III, subsection NB [3]. The two phase flow formation and chocking of the flow in pump is left to experimental investigation rather than numerical simulation.

NOMENCLATURE

MPa	Stress value in Mega Pascal.
P_L	Primary local membrane stress, MPa
P_b	Primary bending stress, MPa
$P_L + P_b$	Primary local membrane plus bending stress, MPa
Q	Secondary (thermal) membrane plus bending stress, MPa
S_m	Design stress intensity value, MPa
S_a	Total stress, MPa

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THE CHALLENGE

High operating temperature affects the design of pump by changing physical properties of material as well as working fluid. In present case, steady state operating temperature and pressure are 350 ºC and 126 bar respectively. As a part of operation condition, temperature of the working fluid varies in between 100°C to 400°C with pressure ranging from 1 bar to 115 bar as shown in Table1.

TABLE 1: TEMPERATURE AND PRESSURE TRANSIENT CONDITIONS.

Case No.	Time(s)	Temperature variation (ºC)	Pressure variation (bar)
1	10	266 to 400	115 to 1
2	30	266 to 100	115 to 1
3	60	200 to 100	115 to 1
4	600	150 to 100	115 to 1

Operation of mechanical seals becomes difficult for such cases due to change in the viscosity of working fluid with temperature.

Sudden change in temperature and pressure generates very high stresses in the pump components. Non uniform temperature distribution in components also cause thermal stresses. The pump components need to be checked for its mechanical strength when thermal loads act simultaneously with mechanical loads. The thermal transient associated with pressure transient could also lead to generation of two phase flow which is not considered.

ANALYSIS OF INITIAL PUMP GEOMETRY

Practically, it is impossible to study the effect of temperature and its cycles on pump components. Numerically it is very much possible to evaluate the thermal conditions of mechanical components. Commercially available finite element analysis technique can handle complex geometry as well as operating conditions. 3D modeling of the pump is done by using Pro-E, V 3.0. ANSYS work bench V 11.0 is used to carry out finite element analysis.

During pump operation, components like impeller, shaft, diffuser etc are fully immersed in working fluid (water). These components are at uniform temperature as that of working fluid. Outside pump surface is having ambient stagnant still air of 30ºC. Differential temperature distribution exists across the components like suction casing, discharge head etc. These components are considered for thermal analysis.

Thermal analysis for proposed pump geometry (suction casing, discharge head and motor stool) is carried out. Temperature of 350ºC is applied at pump wetted surface to evaluate temperature distribution. Convective heat transfer coefficient of $5 w/m^{2o}K$ [6] is considered for pump outer surface. A temperature distribution result for pump geometry is shown in Figure 1.

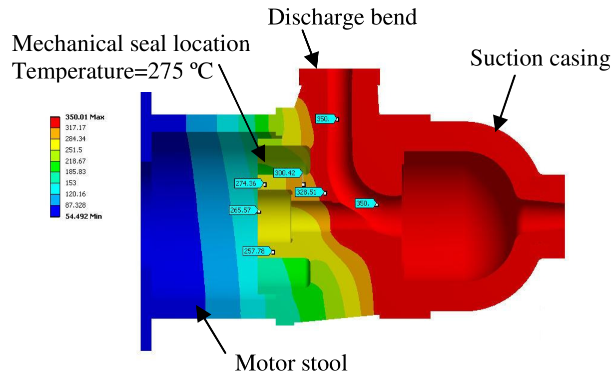

FIGURE 1: TEMPERATURE DISTRIBUTION FOR INITIAL PUMP DISCHARGE BEND GEOMETRY (CROSS SECTIONAL VIEW)

From analysis, it is clear that at pump seal location temperature reaches to 275°C. Operation of mechanical seal affects severely for such high temperature. Such condition affects selection of pump mechanical seal.

ANALYSIS OF MODIFIED PUMP GEOMETRY

To reduce temperature at seal location, pump discharge bend is modified as shown in Figure 2. For modified geometry the temperature at seal location decreased from 275°C to 108°C. Which is much lower compared to the original pump geometry.

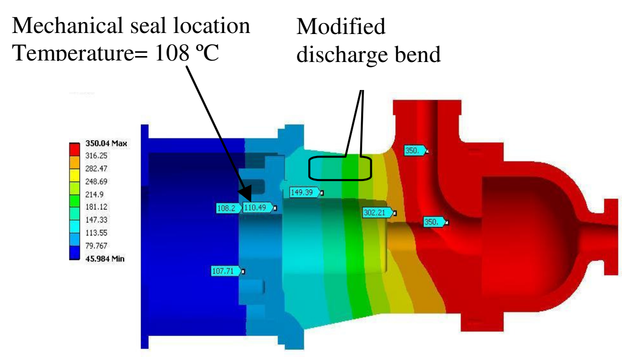

FIGURE 2: TEMPERATURE DISTRIBUTION FOR MODIFIED PUMP DISCHARGE BEND GEOMETRY (CROSS SECTIONAL VIEW)

SELECTION OF MECHANICAL SEAL AND ITS FLUSHING

Transient conditions (as shown in Table 1) of temperature associated with pressure are likely to create problem in maintaining working fluid in liquid state. Thus, the selection of mechanical seal becomes very critical. In the present case, the selection of the suitable mechanical seal has been done by mutual discussion with supplier and rich background from internal resources. Mechanical seal has been selected having back to back arrangement with external seal

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flushing. Mechanical seal Flushing plan 54 as per API 682 is selected [7].

MECHANICAL STRENGTH OF COMPONENTS

Transient nature of thermal and pressure loads reduce strength of pump components drastically. From safety point of view, it is necessary to evaluate effect of temperature and pressure at design stage only. Stress analysis of Suction casing and discharge head is carried out.

The thermal stresses together with stresses developed by mechanical loads, when combined are required to evaluate as per conditions given by ASME section III, subsection NB. Variations of temperature and pressure conditions with respect to time are shown in Table1. ASME section III, subsection NB requirements for level A [3] and level B [3] service loads is $P_L + P_b + Q \leq 3\ S_m$ [3]. Level A service conditions are steady state operating loads and level B service conditions are transient operating loads.

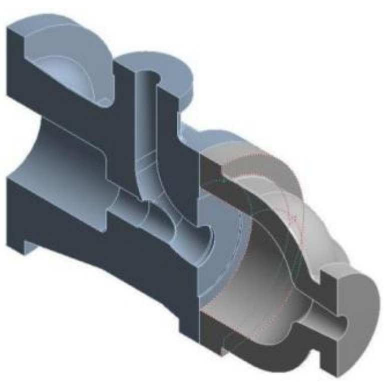

FIGURE 3: SYMMETRIC HALF MODEL OF PUMP GEOMETRY CONSIDERED FOR ANALYSIS

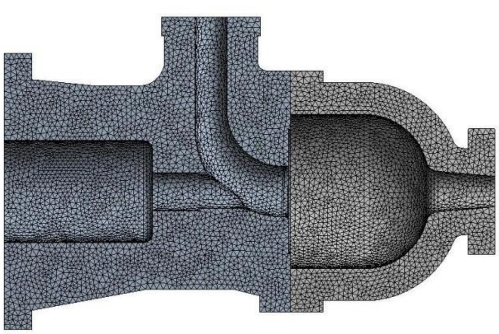

FIGURE 4: MESH MODEL OF PUMP CONSIDERED FOR ANALYSIS

For ease of finite element analysis, symmetric half model is considered as shown in Figure 3. Finite element meshing of geometry is carried out by using 10 node tetrahedron elements. Good quality mesh is ensured with total number of nodes as 0.2 million approximately.

QUALIFICATION FOR LEVEL A SERVICE LOADS

At steady state pump is handling working fluid at temperature 350ºC and pressure 126 bar. Steady state operating condition comes under level A [3] service loads. Thermal stress analysis of suction casing and discharge bend is carried out at 350ºC temperature. For finite element simulation, temperature of 350 ºC is at pump wetted portion. Pump geometry is having symmetric boundary condition.

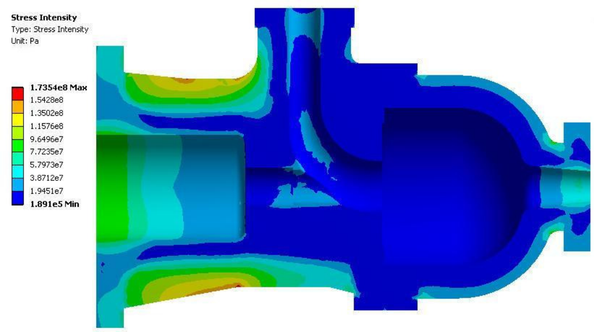

FIGURE 5: STRESS INTENSITY DISTRIBUTION AT STEADY STATE TEMPERATURE OF 350ºC

Stress intensity [3] distribution for thermal stress at steady state temperature is shown in figure 5. It shows maximum value of Q is 173.6 MPa. Maximum stress location is at outer ribs of discharge bend as shown in Figure 6.

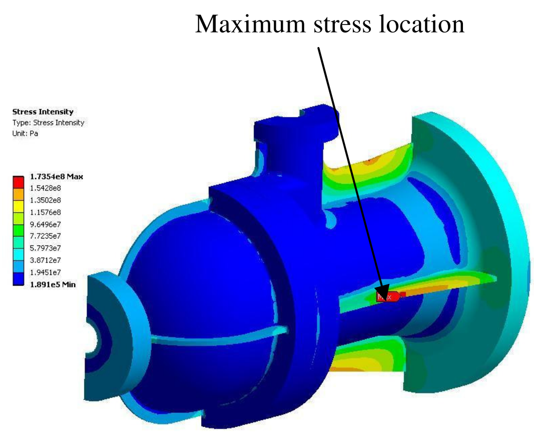

FIGURE 6: MAXIMUM STRESS INTENSITY LOCATION AT STEADY STATE TEMPERATURE OF 350ºC

Design level stress [3] analysis of pump is carried out separately by considering mechanical loads such as internal pressure, forces, moments and self weight. From Design level stress analysis, value of P_L+P_b is obtained as 112.97 MPa.

As per ASME section III, subsection NB total stress S_a is P_L+P_b+Q. Which gives S_a = 286.57MPa (173.6+112.97). This value of S_a is compared with allowable stress value of material at that temperature. Allowable stress is $3S_m$ [3]. S_m is design stress intensity value obtained from ASME section II part D. From this, allowable stress value is 681.00 MPa. Total stress (S_a) obtained from steady state operating condition is less than allowable stress

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QUALIFICATION FOR LEVEL B SERVICE LOADS

Transient operating conditions comes under level B [3] service loads. Transient thermal stress analysis for cases 1 (as per Table 1) is carried out. Temperature changes from 266 ºC to 400 ºC within 10 seconds in wetted portion of pump. Physical properties like young's modulus, thermal conductivity, thermal coefficient of expansion etc changes with respect to temperature. For analysis, variations of above properties are considered.

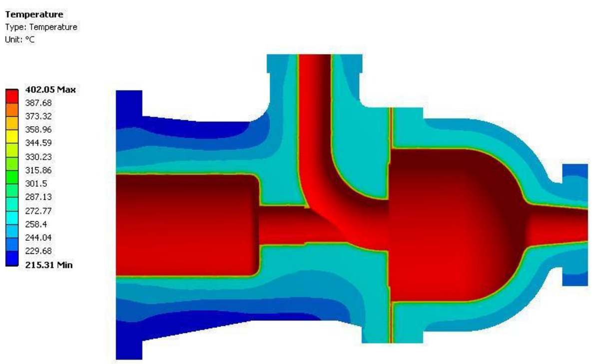

FIGURE 7: TEMPERATURE DISTRIBUTION DUE TO TRANSIENT TEMPERATURE OF 266 ºC TO 400 ºC

From thermal analysis result as shown in Figure 7, it is observed that transient condition changes temperature of inner surfaces only. Non uniform temperature distribution in suction casing and discharge bend leads to thermal stress. Highly generated thermal stresses for inner surfaces are shown in Figure 8. Maximum value of Q is 449.46 MPa at inner surface of discharge bend which in shown in Figure 9.

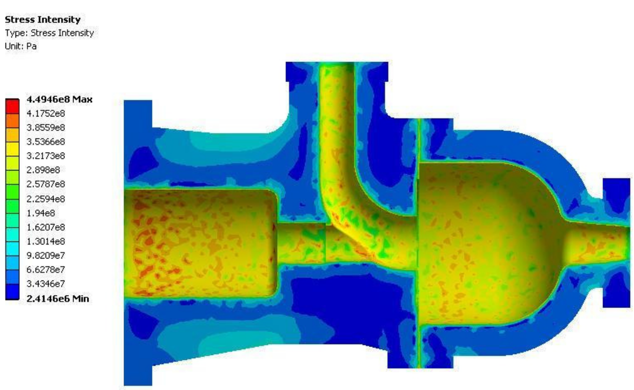

FIGURE 8: STRESS INTENSITY DISTRIBUTION DUE TO TRANSIENT TEMPERATURE OF 266 ºC TO 400 ºC

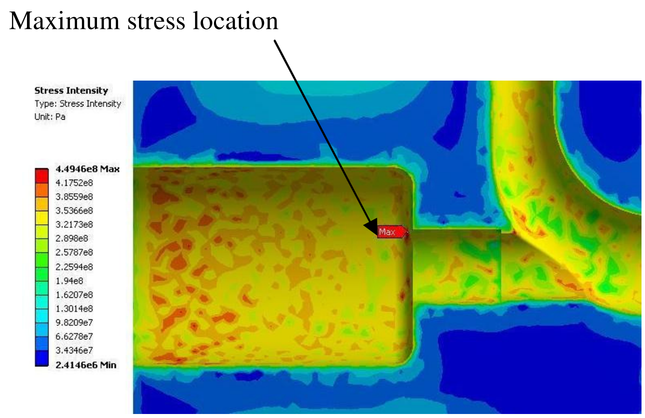

FIGURE 9: MAXIMUM STRESS INTENSITY LOCATION DUE TO TRANSIENT TEMPERATURE OF 266 ºC TO 400 ºC

Stress analysis for transient pressure load of 115 bar to 1 bar is carried out. It gives maximum value of P_L+P_b as 0.8693MPa. For worst condition, value of P_L+P_b obtained from design stress analysis is considered to calculate S_a. Total stress (S_a) is 562.43 MPa (112.97+ 449.46).
On similar basis, values of Q for remaining cases are evaluated. Maximum value of Q for case 2, case 3 and case 4 are 574.03 MPa, 340.31 MPa and 191.05 MPa respectively. Values of S_a for all four cases are evaluated by considering P_L+P_b+Q. These values of S_a and allowable stress of material are shown in Table 2. Total stress (S_a) obtained from transient operating conditions is less than allowable stress.

CUMULATIVE FATIGUE DAMAGE

Transient operating conditions (case 1 to case 4) of pump are cyclic in nature. This cyclic nature of loads may lead to fatigue failure of pump components. Maximum number of cycles that material can withstand for specified operating conditions is evaluated as per design fatigue curve [4].

TABLE 2: QUALIFICATION FOR LEVEL B SERVICE LOADS.

Stress	Case1	Case2	Case3	Case4
S_a (MPa)	562.43	687.00	453.28	304.02
Allowable stress ($3S_m$) (MPa)	681.00	705.00	726.00	744.00

TABLE 3: MAXIMUM STRESS VS NUMBER OF CYCLES FOR LEVEL A SERVICE LOADS.

Level A service loads.	Maximum stress	Number of Cycles as per design fatigue curve	Number of cycles Required
Steady state operating load	286.57	41000	10000

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TABLE 4: MAXIMUM STRESS VS NUMBER OF CYCLES FOR LEVEL B SERVICE LOADS.

Level B service load	Maximum stress	Number of Cycles as per design fatigue curve	Number of cycles required
Case 1	562.43	1000	50
Case 2	687.00	600	200
Case 3	453.28	5000	1000
Case 4	304.02	40000	1750

Number of cycles are evaluated and checked with required number of cycles for level A and level B service load is shown in Table 3 and 4. Cumulative fatigue damage for all above service loads is calculated as follows,

Cumulative fatigue damage= fatigue damage due to service loads A + Cumulative fatigue damage due to service loads B.

Cumulative fatigue damage = {(10000/41000) + (50/1000) + (200/600) + (1000/5000) + (1750/40000)}

Cumulative fatigue damage = 0.87 < 1

CONCLUSION

To verify the temperature distribution at mechanical seal location, thermal analysis is carried out with initial geometry. It is observed that temperature at seal location is very high for its normal operation. Thus length of discharge bend is increased to reduce temperature at seal location which improves life of mechanical seal during its operation.

Stress analysis results shows contribution of thermal stress is more in total stress. Transient nature of temperature affects severely at the inner surface of pump geometry. Maximum value of total stresses obtained from both steady and transient operating conditions are validated as per ASME section III, subsection NB.

As cumulative fatigue damage factor is less than one, pump will operate for required number of cycles. Hence operation of pump is safe under steady and transient conditions.

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