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Design and Analysis of Mixing Sphere in Start-up System of Supercritical Boilers

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***Abstract* -** Thickness of the mixing sphere with multi openings in the startup system is unattainable using the design codes. Obtaining the values experimentally is also quite uneconomical. Hence the analysis of sphere with openings was performed to find the allowable thickness values based on maximum stress conditions given in ASME code. The whole start-up system was modeled in CAESAR and the flexibility analysis was carried out for all the static load cases. The reaction forces and moments acting on the connecting nozzles of the sphere were obtained and given as inputs for the finite element analysis of the sphere. The mixing sphere was modeled in ANSYS and analyzed for the maximum stress that occurs for different thickness values. The analysis reveals that the maximum stresses for the given thickness values are within the allowable limit according to ASME design code. The optimum thickness of the sphere and the corresponding allowable pressure were also determined.

Keywords *- Mixing Sphere, Start-up System, Burst test*

I. INTRODUCTION

The start-up system in super critical boilers is used to protect super-heaters from water carry-over by separating water from steam and re-circulating it through the evaporator surfaces during start-up, low-load operation and shutdown of the boiler. The required water flow rate through the evaporator tubes is therefore maintained greater than the evaporation rate to protect them against overheating. The entire system but the furnace connecting sphere and the recirculation mixing sphere, used as intersection points of the piping in the system, can be sized based on the guidelines mentioned in the ASME codes [1]. Thickness of the sphere can be determined by carrying out burst tests which is very expensive. In order to eliminate the necessity of costly destructive testing was proposed to carry out stress analysis of the mixing sphere to check for the safety of the design and to find out the optimum thickness. The analysis of the sphere with openings is very important as the openings constitute a potential source of weakness. The case of an opening in a spherical shell under internal pressure without and with pressure acting at the bore was reported by Schindler [2]. Though tremendous advances have been made in numerical simulation, most of the papers report the case of a smaller diameter of the sphere, single nozzle-vessel connection and stresses on either shell or nozzle. Stresses due to the external loads on the nozzle can be more critical than those due to internal pressure [3].

This paper provides data on practical spherical vessel with multi-openings considering internal pressure and external loadings.

II. START-UP SYSTEM AND THE MIXING SPHERE

The schematic of the start-up system is shown in the Fig.1 starting from separators extending to the economizer inlet link. Water from the economizer enters the system through risers. It flows through the separators where the hot water is separated from steam and is collected in the collecting vessel. From there water flows through the down comer connecting links, furnace connecting sphere where the outlets from the collecting vessels unite, water down comer to recirculation mixing sphere where the temperature of water is reduced before it enters the recirculation pump, and joins the feed line after the globe check valve.

The mixing sphere has three nozzles of which two are inlets and one is an outlet. One inlet is from the feed water line through the diffuser and another is from the down comer. The pressure drop from feed water side has lesser effect on the stresses that occur in the sphere because the feed water pump can always force the flow from that direction creating the head to balance the pressure drop. But the flow from the down comer side is highly sensitive to pressure drop which directly influences the stresses on the sphere. Thermal stresses also will be induced in the components because of the mixing of the fluids which are at different temperatures. The pressure and temperature of the fluid flowing into the nozzle will affect the stress developed. In addition the weight of the piping also is transferred on the sphere. Generally the sphere is self-supported in the pipeline or it may have separate supports. They serve as a mixing chambers and distribution points in the circuit and hence are subjected to thermal and pressure loads. To withstand the loads the sphere is generally made of high wall thickness. Carbon steels and low alloy steels are the materials used to withstand the high temperatures and high pressures that occur in the system.

III. FLEXIBILITY ANALYSIS OF THE START-UP SYSTEM

A. *Introduction*

Flexibility analysis of the system is performed to obtain the exact reaction forces and moments acting on to the nozzle of the mixing sphere using the CAESAR [4] pipeline flexibility analysis software.

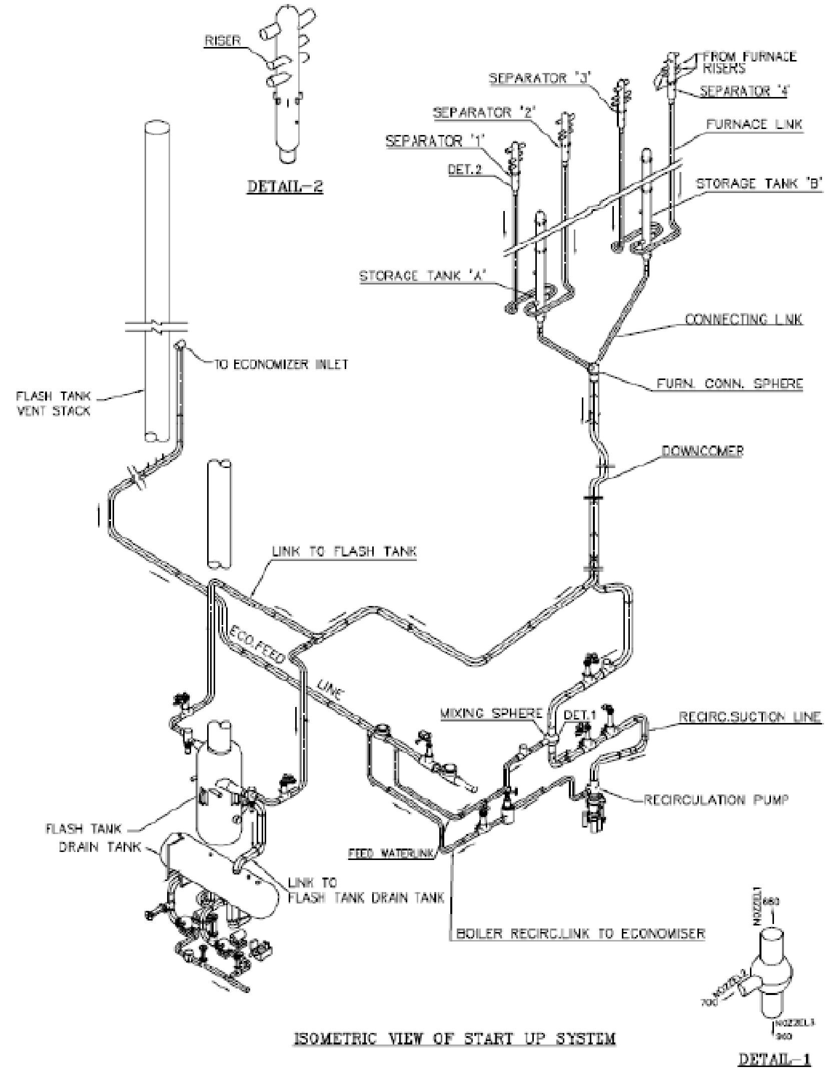

Figure-1 : Start-up System

B. Modelling

The piping system along with furnace connecting sphere, recirculation mixing sphere, recirculation pump, valves and fittings were modeled in CEASER as shown in Fig. 1. The pipes were modeled as elements with end nodes by specifying the co-ordinates and the direction. The geometric details of the pipes namely diameter and thickness were also given as input. The bends and fittings namely elbows and Tee's were defined using necessary tools available in CEASER. The spheres, pump and valves were modeled as rigid bodies. The nozzles in the mixing spheres were defined as nodes where results are required.

C. Materials

The material properties [5] and the operating conditions, i.e., the internal fluid pressure and the temperature, of each component specified for the supercritical boiler considered are given in Table 1.

D. Loads and Boundary Conditions

Various types of loads act on the start-up system both internally as well as externally. Internal loads include the internal pressure of the fluid (Table I) and the self weight of the piping and the components. External loads include forces and moments from the economizer. Displacement boundary conditions were applied to the separator which connects the furnace to the start-up system through risers as shown in Fig. 2. The displacement with the flash tank is also given as boundary condition to the start-up system is shown in Fig. 3. The separator is arrested in the vertical direction by supporting from the structure by rigid rods. The weights of components are modeled as rigid bodies as shown in Table II. The forces and moments developed in the economizer inlet link are applied on the start-up system. A separate analysis is carried out on the economizer inlet links (not included) to calculate the forces and moments that occur. A force of 71172 N and a moment of 85417 Nm are applied equally in the x, y and z directions on the start-up system.

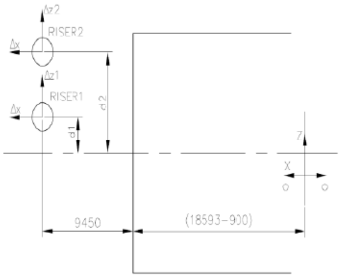

Fig. 2: Thermal expansion at riser pipes

Riser tube expansions

Design temperature of riser, T = 482.2°C

Coefficient of thermal expansion $\alpha = 6.58$ mm/m

$\Delta 1z = d1 * \alpha = 2.99 * 6.58 = 19.67$ mm;

$\Delta 2z = d2 * \alpha = 8.172 * 6.88 = 53.78$ mm;

$\Delta x = dx * \alpha = -(18.593 + 4.45) * 6.58 = -145.7$ mm.

(Minus sign indicates that the displacement of risers is away from the furnace).

Flash tank expansions

T1 = Temperature at Link1 (390.6°C), $\alpha_1 = 5.09$ mm/m

T2 = Temperature at Link2 (349°C), $\alpha_2 = 4.55$ mm/m

TABLE I: MATERIAL PROPERTIES AND OPERATING CONDITIONS AT EACH COMPONENT

S.No.	Component	Size (dxt) mm x mm	Material Grade[5]	E in MPa	v	σ_y in MPa	Internal Pressure in MPa	Temperature in °c
1	Separator	558.8x63.5	SA 335 P91	182992.1	0.31	415	285	480
2	Storage tank	660.4x73	SA 335 P91	182992.1	0.31	415	285	480
3	Furnace down comer	355.6x50.8	SA 335 P12	173999.4	0.31	275	285	480
4	Connecting sphere	812.8x63.5	SA 387 GR.12	171998.8	0.31	230	285	480
5	Water storage down comer	558.8x76.2	SA 106 GR.C	177000.2	0.3	275	285	480
6	Economizer Feed line	457.2x55.5	SA 106 GR.C	177000.2	0.3	275	285	480
7	Recirculation pump suction line	508x74.6	SA 106 GR.C	177000.2	0.3	275	314	390.6
8	Boiler recirculation link to economizer	355.6x 52.3	SA 106 GR.C	177000.2	0.3	275	314	390.6
9	Start-up feed water link	323.8x49.2	SA 106 GR.C	177000.2	0.3	275	314	390.6
10	Feed water link to mixing sphere	323.8x 49.2	SA 106 GR.C	177000.2	0.3	275	314	390.6

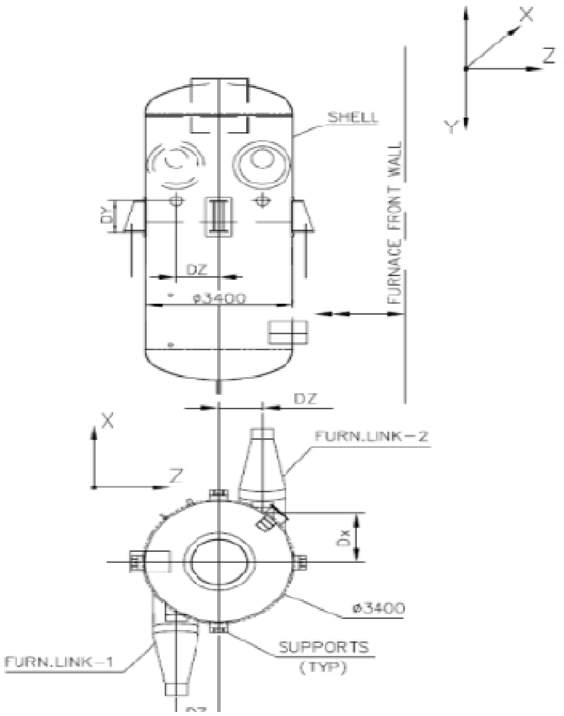

Fig. 3 : Thermal expansions at flash tank.

$\Delta x_1 = Dx1 * \alpha_1 = 1.375 * 5.09 = 7.03$ mm/m;

$\Delta y_1 = Dy1 * \alpha_1 = 25.84 * 5.09 = 13.4$ mm/m

$\Delta z_1 = Dz1 * \alpha_1 = 1 * 5.09 = 5.09$ mm/m.

$(Dx_1 = Dx2 = Dx; Dy1 = Dy2 = Dy; Dz1 = Dz2 = Dz)$.

$\Delta x_2 = Dx2 * \alpha_2 = 1.375 * 4.55 = 6.25$ mm/m;

$\Delta y_2 = Dy2 * \alpha_2 = 25.84 * 4.55 = 11.98$ mm/m;

$\Delta z_2 = Dz2 * \alpha_2 = 1 * 4.55 = 4.55$mm/m.

E. Analysis and Results

The flexibility analysis of the start-up system was carried out as per boiler piping code B31.1 2007 [6].The required reaction forces and moments acting on the nozzles viz., at nodes 960 and 700 of the mixing sphere were retrieved from the corresponding nodes for the case of operating conditions and presented in Table III.

IV. FINITE ELEMENT ANALYSIS OF THE SPHERE

Three-dimensional linear, static finite element analysis of the mixing spheres with three nozzles and a man-hole door was performed by use of the commercial analysis software, ANSYS [7]. Elastic stress analysis method from the ASME code [1] that has been conservatively established to prevent a plastic collapse is opted. According to the method, stresses are computed by performing an elastic stress analysis, classified into categories, and limited to allowable values. The structural linear analysis of the system at room temperature was performed ignoring the reinforcement of the sphere-nozzle intersection.

TABLE II : WEIGHTS OF THE COMPONENTS

Product	**Connecting sphere**	**Mixing sphere**	**valves**	**Recirculation pump**
Weight (kg)	2280	8061	45761	11316

TABLE III : ELEMENT FORCES AT OPERATING LOADS

Node	**F_x in kN**	**F_y in kN**	**F_z in kN**	**M_z in kN-m**	**M_y in kN-m**	**M_z in kN-m**
700 (nozzle2)	-0.39	13.98	5.16	-11.53	-10.27	1.574
960 (nozzle2)	-0.36	4.39	3.23	46.81	-3.72	6.68

A. Modeling

The mixing sphere with nozzles and a man-hole door has been modeled in ANSYS. The thickness chosen for the study varies from 100 mm to 145 mm with increments of 5mm to determine the optimized thickness. Other geometric details of the sphere with nozzles are shown in Fig. 4. Solid 45 eight node element with plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities was used to model the sphere. The element is defined by eight nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions [7].

B. Materials

High carbon steel of a specific grade is used for the component to withstand the high temperatures and pressures. The material is modeled by assigning the mechanical properties of the material to the component. The grade of the material and its mechanical properties are presented in Table IV.

C. Meshing

Because of the different geometrical shapes, the entire component is divided into volumes for meshing purpose. Free volume meshing was performed for the sphere whereas mapped meshing was done for the nozzles. In order to achieve accuracy in determination of stresses, fine meshing was carried out at the sphere-nozzle intersection corners as shown in Fig. 5. The model consists of 16022 elements and 20175 nodes.

D. Loads and Boundary Conditions

Due to the symmetry of the geometry, only half of the structure has been considered for analysis. Symmetric boundary conditions were fully employed on x-y plane (z=0) as shown in Fig. 6. Loads on the system include the internal pressure and the external reaction forces and moments on the nozzles. An internal pressure of 135.45 MPa was applied on the internal surface of the component as shown in Fig 6. The reaction forces and moments (Table III) from output of CAESAR analysis are retrieved and applied on the cross-sectional areas of the nozzles 2 and 3. For this purpose a master node was created at the centre of the circular cross-section of each nozzle at the open end. A rigid beam element, BEAM 3 [8], was created between the master node and the nodes on the nozzle face for proper distribution of forces as shown in Fig. 6. An equivalent pressure is imposed on the nozzle faces to avoid the longitudinal stresses caused by the openings in the sphere. Equivalent pressures of 25.8 MPa and 28.64 MPa are applied on the nozzle 2 and 3 respectively. Nozzle 1 is fixed in all the directions.

TABLE IV. WEIGHTS OF THE COMPONENTS

Material	**E (MPa)**	**ν**	**σ_y in MPa**	**σ_{allow} MPa**
SA 299 GR.A	177000	0.3	275	135.45

E. Results and Discussion

1) *Thickness optimization:* Stress analysis was carried out to find the optimum thickness, on the basis of trial and error, for a particular practical internal pressure of about 32 MPa. The thickness was varied from 100 mm to 145 mm in steps of 5 mm. The variation of equivalent stress for the thicknesses considered are shown in Fig. 7 in which the induced stress for the thickness of 140 mm is minimum and hence the optimized thickness of 140 mm has been selected for the mixing sphere of the start-up system.

2) *Determination of Maximum Pressure:* The study was extended to determine the maximum allowable pressure of the material, the allowable stress limit being $2\sigma_y$ for the particular optimized thickness of 140 mm. The pressure was varied from 3 kg-f/mm2 (29.43 MPa) to 5 kg-f/mm2 (49.05 MPa) in steps of 0.25 kg-f/mm2. The variation of equivalent stress induced for these internal pressures can be observed from the Fig. 8. The internal pressure of 4 kgf/ mm2 (39.24 MPa) is the maximum pressure for which the induced stress is 542 MPa which is near to the allowable stress limit of 550 MPa ($2\sigma_y$).

3) *Validation with analytical Formula:* According to C.J. Dekker [9] reactions from connected piping may give rise to high stresses and these stresses are in addition to the pressure induced stresses.

The various stresses occurring at nozzle/vessel junctions are to be categorized as follows

- Membrane stresses (local) due to internal pressure are primary stresses.
- The bending stresses due to internal pressure are secondary stresses.

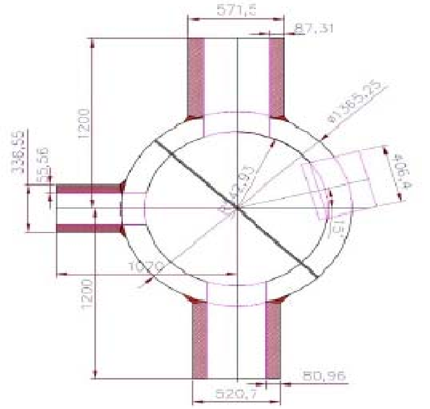

Fig. 4: Dimensions of the sphere

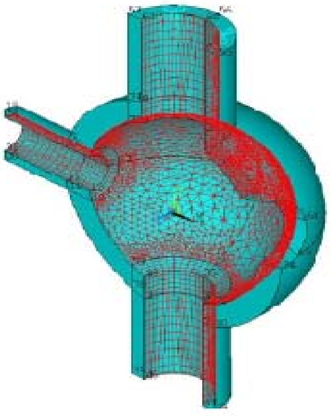

Fig. 5: Half symmetric FE model with internal pressure.

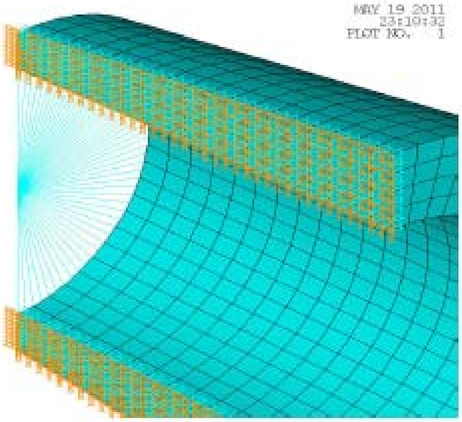

Fig. 6: Half symmetric FE model of nozzle with beam element.

- Membrane stresses (local) due to external loads belong to the local primary membrane stress category whether the origin of the external loading is mechanical or thermal.
- Bending stresses due to external loads belong always to the secondary stress category.

The ASME code [10] formula given below is the acceptability criteria used to validate the results obtained from finite element analysis.

$$PL + Pb + Q < 3Sm \qquad (1)$$

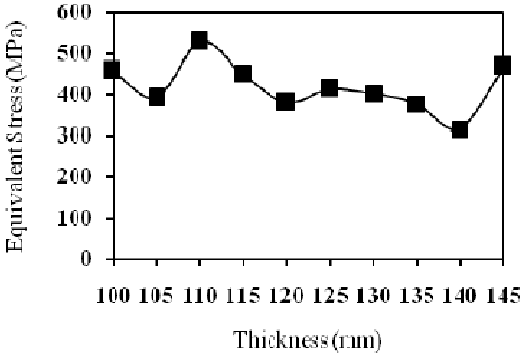

Fig. 7: Variation of stress for various thicknesses

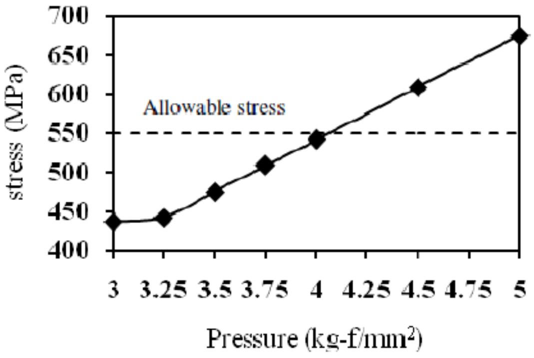

Fig. 8: Variation of stress for various pressures

The allowable stress for the chosen material, as per SME code [5] is 135.5 MPa (Sm). The membrane, bending and peak stresses were obtained for the internal pressure of 135.45 MPa at the nozzle-sphere junction by creating path across the thickness of the junction. The induced membrane (PL), bending (Pb) and peak (Q) stresses are 126 MPa, 27 MPa and 4.11 MPa respectively. The values were substituted in Eq. (1) to check whether the induced stresses are within 3Sm. The sum of membrane, bending and peak stresses is 156.36 MPa. The allowable limit is 406.5 MPa (3Sm). The finite element analysis is therefore validated.

V. CONCLUSION

The mixing sphere of the start-up system for a super critical boiler was investigated using finite element linear static analysis. The start-up system was modeled and analysed in CEASER to get the forces and moments acting on the mixing sphere and given as input in the finite element analysis. The analysis was performed to find the optimum thickness, for a particular internal pressure, for which the induced stress is within the allowable stress. The analysis was extended to obtain the maximum pressure for the safe thickness at which the induced stress is within the allowable limit. The finite element results namely membrane, bending and peak stresses were substituted in the code formula to verify if the induced stresses were within the limit prescribed by code. The design is was found to be safe. The experiments on the sphere are expensive and time consuming, the numerical method such as finite element method can hence be used for the analyses and design mixing sphere.

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NOMENCLATURE

E	Young's modulus
ν	Poisson's Ratio
σ_y	Yield stress
σ_u	Ultimate stress
σ_{allow}	Allowable stress
F_x	Reaction Force in x direction
F_y	Reaction Force in y direction
F_z	Reaction Force in z direction
M_x	Moment in x direction
M_y	Moment in y direction
M_z	Moment in z direction
$\Delta 1z$	thermal expansion of risers 1 in z- direction
$\Delta 2z$	thermal expansion of risers 2 in z- direction.
Δx	thermal expansion of risers 1 and 2 in x-direction
d1	distance from the boiler axis to the centerline of riser1
d2	distance from the axis to the centreline of riser2
dx	distance from the boiler axis to the axis of riser1&2
Dx	distance from centre line of flash tank to furnace link in x-direction
Dy	distance from centre line of flash tank to furnace link in y-direction
Dz	distance from centre line of flash tank to furnace link in z-direction
Dx_1	distance from centre line of flash tank to furnace link 1 in x-direction
Dy_1	distance from centre line of flash tank to furnace link 1 in y-direction
Dz_1	distance from centre line of flash tank to furnace link 1 in z-direction
Dx_2	distance from centre line of flash tank to furnace link 2 in x-direction
Dy_2	distance from centre line of flash tank to furnace link 2 in y-direction
Dz_2	distance from centre line of flash tank to furnace link 2 in z-direction
P_b	Primary bending equivalent stress
P_L	Local Primary membrane equivalent stress
Q	Secondary equivalent stress resulting from operating loadings
S_m	allowable stress intensity as per ASME Section III, Division 2, 2007, MPa
C_L	Center line
EXP	Expansion value

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