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Stress Classification Lines for Non-Standard Y-Tee Design in Piping System

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ABSTRACT

All processes fluids in the oil and gas industry are generally transferred from one point to another via complex piping networks. The process temperature and the pressure of the fluids may vary between -21 °C to 816 °C and from atmospheric, 1.013 barg to 431 barg, respectively. Under these conditions, the entire piping networks are exposed to thermal expansion and contraction resulting in mechanical bending, potentially causing mechanical failures. The most common pipe fitting used in piping networks to divide the fluid flow into two different directions would be a standard equal tee which was designed in accordance with ASME B16.9. The equal tee, however, has distinctive flexibility characteristics compared to the pipe, which is well defined in ASME B31.3. These flexibility characteristics generate a stress intensification factor (SIF) through bending moment equations for beam-type element analysis for standard equal tee components. However, due to the limitation of SIF and flexibility characteristics for the non-standard pipe fittings, the stress evaluation could not be done using beam-type element analysis. When using finite element method (FEM) analysis, generally equivalent stresses (Von Mises or Tresca) or principal stresses would be evaluated depending on the stress categorization as explained in ASME codes and standards. In this paper, a stress evaluation method for non-standard Y-tee was developed and demonstrated against ASME VIII Div. 2 Part 5 requirement. The developed SCLs line for the non-standard Y-tee fittings provides the guides for stress assessment against design stress allowable. An optimum design for non-standard Y-tee is proposed at 45° and 60° crotch radii which resulted in lower stress value at SCL lines, hence improving the structural integrity.

1. Introduction

In oil and gas industry, all processed fluids are generally transferred via complex piping networks. The process temperature and the pressure may vary between -21 ^{o}C to 816 ^{o}C and from atmospheric, 1.013 barg to 431 barg, respectively [1]. As such, the piping systems are exposed to thermal expansion and contraction resulting in mechanical bending. The most common pipe fitting used in the piping networks in dividing the fluid flow into two different directions or vice versa, would be a

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standard equal tee designed in accordance with ASME B16.9 [2]. The equal tee, however, has distinctive flexibility characteristics compared to the pipe, which is well defined in ASME B31.3 [3].

In 2012, one of Liquefied Natural Gas (LNG) facilities reported several piping systems transporting two-phase flow process fluid were subjected to strong vibration. Preliminary visual inspection of the entire piping system revealed that the strongest vibration encountered was at the standard equal tee fittings connections between the pipelines. The fluids in each pipeline flow in the opposite directions and collided before it enters the branch pipeline. This collision resulted in a heavy vibration occurring at the center of the standard equal tee, which was later transmitted along with the piping system, damaging several pipes supports attached to the piping network.

To reduce the vibration intensity, several recommendations have been made to minimize the vibration to an acceptable level under the requirement as per the Energy Institute (EI) guidelines handbook [4]. This includes by adding piping component such as valve [5] or using non-standard fitting connection to replace the previous standard equals tee, which able to ease the fluids flow. Nonetheless, the design and development of non-standard Y-tee fittings connections in the piping system are not exclusively specified in any of the ASME standards. The requirements for the development and implementation of such non-standard fittings consist of three main criteria: pressure test, load test, and finite element stress analysis [6]. Besides, the distribution of stresses to the non-standard Y-tee fittings has not yet documented, determined, and identified following ASME VIII BPV Division 2 Part 5, 2019 requirements [7].

In ASME Section VIII code [7], there are no specific approach or methodology defined to guide users for quantification of the membrane and primary stresses of the component. Some users may have different understanding of how to perform stress linearization using SCL method which results in variation of the stress results. There are several published papers related to stress linearization using SCL approach to deduce the stress components. Carlos *et al.,* [8] published a study on a methodology to perform stress linearization and classification based on ASME code on a generic geometry found in many plants, particularly from petrochemical to nuclear plant. Their study used a four-way Wye piping component which was modeled using 3-D elements and subjected to only internal pressure loading. The SCLs for the component is introduced to correctly deduced the component stress based on location of high total stress in FE model. The most interesting part about the SCL lines of their model would be the line across the crotch area of the model identified as line “7” in Figure 1 where high stress profile was observed. The SCL line “7” in their work was adapted from Hechmer and Hollinger [9].

Another interesting work performed by Strzelczyk and Ho [10] introduced a method of stress linearization without actual linearization been performed. Their work was performed on solid element model of a circular nozzle attached to a cylinder vessel subjected to pressure load and thermal expansion. For solid element model, the linearization process of the stresses is time consuming and the results dependent on how SCLs were drawn. In their proposed method, modification on the original model needs to be performed where the vessel must be modeled out of quadratic 20 node solid element. The mesh density must be sufficient for stress concentration representation and the mesh lines in the thickness direction must be more or less normal to the surface.

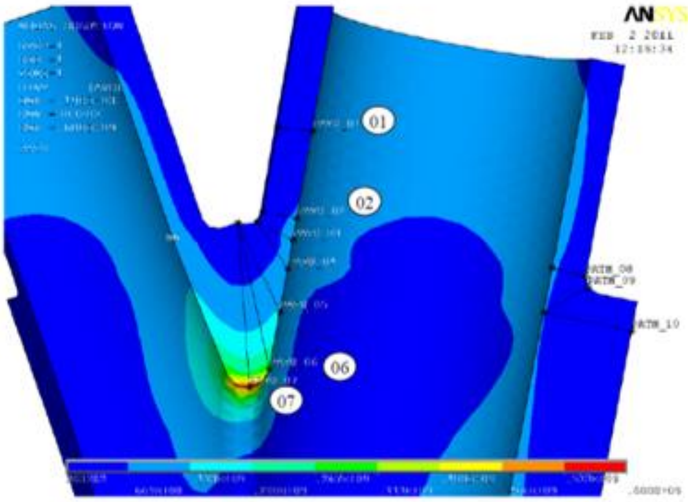


Fig. 1. Stress Classification Lines (SCL) on Y-tee fittings [6]

Based on the aforementioned literature surveys, there is no established procedure or method for constructing SCL for a specific component like piping branch, tee or Y-tee. Hence, this paper will describe the analysis of non-standard Y-tee fittings for stress distributions using ASME VIII [7] as a guideline for the requirements of calculated stress level. The reduction of vibration strength that produces smooth flow patterns around the Y-tee joints, on the other hand, will not be discussed.

The present study aimed to evaluate the integrity of the Y-tee fittings against internal and external loading due to pressure and piping thermal displacement, respectively using ABAQUS software. The distribution and reduction of stresses for the non-standard Y-tee fittings is determined in accordance with ASME VIII [7]. Moreover, the optimum design of the Y-tee is also demonstrated by modifying the crotch radius angle and the effect of variable crotch angle on the primary and secondary stresses are evaluated.

2. Theoretical Background

2.1 Pressure Distribution

Conventionally, pipe stress analysis was performed to determine the complete piping system's flexibility characteristics. CAESAR II software is commonly used to perform the analysis where the stresses are evaluated with reference code and standard allowable value. Since the current piping system was subjected to thermal expansion due to the processed fluids pressure and temperature, it creates bending in the piping system. This bending generates thermal expansion stress, S_E ranges from ambient to operating temperature that can be determined from Eq. (1) and the value should not exceed the allowable displacement stress ranges S_A from Eq. (2) [3].

$$S_E = \sqrt{(|S_a| + S_b)^2 + (2S_t)^2}, \tag{1}$$

$$S_A = f(1.25S_c + 0.25S_h), \tag{2}$$

where S_a is axial stress, S_b is bending stress and S_t is torsional stress. The *f* value is stress range factor generated from Figure 302.3.5 of ASME B31.3 [3], S_c is cold allowable stress at ambient temperature whereas the S_h is hot allowable stress at operating temperature of the materials.

In CAESER II, the pipe stress analysis was performed using beam element configuration. In beam element analysis, the calculated S_E values can be obtained at all branch connections or pipe fittings including the Y-tee. However, the calculated S_b in Eq. (3) requires values of stress intensification factor *(SIF)* in out-plane, i_o as well as in-plane, i_i which can be determined from experimental or analytical means as describes in ASME B31J [10,11] for non-standard piping components. The M_i and M_o are the moment in-plane and out-plane respectively and Z_e is the section modulus of the pipe [3].

$$S_b = \frac{\sqrt{(i_i M_i)^2 + (i_o M_o)^2}}{Ze} \tag{3}$$

Since the beam element analysis has limited ability to compute the bending stresses, S_b around the Y-tee for stress evaluation due to the lack of i_i and i_o information, a detailed finite element analysis using solid elements model is recommended. However, detailed finite element modelling often leads to significant computational effort if a complete piping system is modelled. Simplification on the modelling technique was identified to utilize a local model (Y-tee) whereas the input for boundary conditions due to thermal expansion were deduced from global model, *i.e.,* CAESER II analysis.

2.2 Design-by-analysis Requirement of Y-tee

The design-by-analysis requirement as defined in ASME VIII Div. 2 Part 5 [7] is used to evaluate components for plastic collapse, local failure, buckling and cyclic loading. The detailed design procedures utilizing the stress analysis results from three alternative analysis methods. *i.e.*, elastic stress analysis, elastic-plastic analysis, and limit load analysis. The design-by-analysis requirement as defined in ASME [7], are based on protection against the failure modes, namely, Protection against Plastic Collapse, Protection against Local Failure, Protection against Collapse from Buckling, and Protection against Failure from Cyclic Loading. The requirement components are to be evaluated for each applicable failure mode.

In current work, it was assumed that no plastic deformation on Y-tee as it was not subjected to compressive stress field under applied design load. Hence, the component is only evaluated for Protection against Plastic Collapse and Protection against Local Failure by means of elastic stress analysis method.

2.2.1 Protection Against Plastic Collapse

In assessment for protection against plastic collapse, the results from an elastic stress analysis with defined loading conditions are categorized and compared with the limiting stress value. The basis of categorization procedures includes the following: (a) the equivalent stress is deduced at locations in the component and compared to the allowable equivalent stress, and (b) maximum distortion energy yield criterion is used to determine the allowable equivalent stress.

For maximum distortion energy yield criterion, the equivalent stress is defined as the von Misses stress, σ_e, which defined as Eq. (4) for 3-D surface planes.

$$S_e = \sigma_e = \frac{1}{\sqrt{2}}\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}, \tag{4}$$

where the symbol notations for 1, 2 and 3 are referred to the 3-dimensional axis of *x, y,* and *z* as per Figure 2.

There are three basis equivalent stress categories, and their limits are to be satisfied for plastic collapse assessment: (1) General primary membrane equivalent stress, P_m, (2) Local primary membrane equivalent stress, P_L, and (3) Primary membrane (general or local) plus the primary bending equivalent stress, $P_L + P_b$. All of these stress categories are to be evaluated for specified design loads and allowable stress as per defined in ASME VIII Div.2 Part 5 Table 5.3 [7].

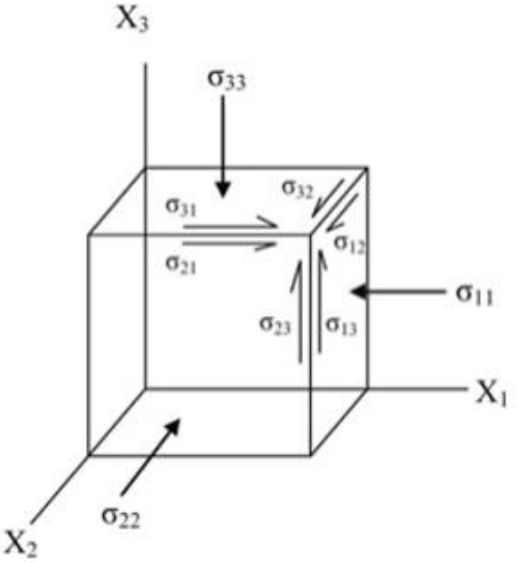


Fig. 2. Stress tensor in 3-D dimensional space

2.2.2 Protection Against Local Failure

In addition to evaluation on protection against plastic collapse, the local failure protection criterion shall be satisfied for the components. These requirements are applied for all components where the thickness and the configuration are established by using design-by-analysis rules.

The assessment for protection against local failure can be established based on elastic analysis or elastic-plastic analysis, which the latter provides more accurate estimation. In current work, the results from elastic analysis were used to provide approximation of the protection against local failure. The assessment uses triaxial stress limit which is the algebraic sum of the three linearized primary principal stresses obtained from Design Load Combination 1 $(P + P_S + D)$ where P is the design internal and external pressure, P_S is design static head and D is the deadweight load. The triaxial stress limit is defined as per Eq. (5) [7].

$$(\sigma_1 + \sigma_2 + \sigma_3) \leq 4S, \tag{5}$$

where S is the allowable stress based on the material of construction and design temperature.

3. Methodology

The stress analysis of the Y-tee was performed in two steps: global piping system model and local model of Y-tee. Single step simulation for global piping system can be performed, but subject to element type limitation and larger computational hours. Pipe stress analysis using CAESAR II [12] was performed for the global model. The modelling utilized beam type elements; hence computational hours are greatly reduced. However, using beam type elements, the details of Y-tee geometry and profile is not possible to model and often requires reference SIF for the fittings for more accurate prediction. Nonetheless, the reference SIF for non-standard Y-tee in current work is not available from standard or literatures.

To overcome this limitation, ABAQUS [13,14] software was used to model the Y-tee as local model using solid element representation. It is also possible to model the global piping system in ABAQUS but subjected to extensive computational hours for converge solutions. Having said that, a two-step modelling technique was employed in current work, where the results from global CAESAR II model were used for boundary input in local Y-tee model in ABAQUS.

Figure 3 shows the complete piping system at the site facility. The process liquid travels from pressure vessel V-1000 and V-2000 via piping line numbers 18” P21025 and 18” P21023, respectively, moving towards the Y-tee. Before the fluids enter the Y-tee, there is concentric reducer 20” x 18” on each side of the Y-tee. The processed liquid would pass the Y-tee and flow straight to Column C501 via piping line number 20” P21026. The process conditions of the piping system are shown as per Table 1.

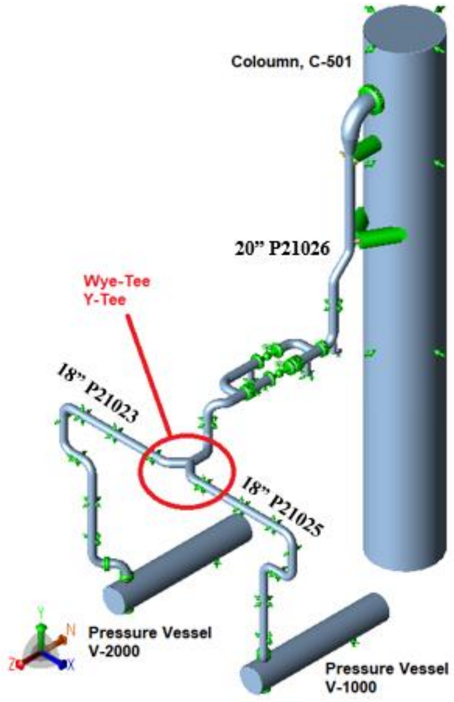


Fig. 3. Global piping system modelled in CAESER II

Table 1
Test model specification and test conditions

Line Number	Operating Temperature, °C	Operating Pressure, barg
18” P21023	135	11
18” P21025	135	11
20” P21026	135	11

3.1 Piping System Input in CAESAR II

The detail closed-up of Y-tee connection is illustrated in Figure 4 where Nodes 4175, 4375 and 4410 were the reference of boundary conditions of the Y-Tee. The nodes were identified as an anchor with the connecting nodes [12], term that is used for imaginary anchor pipe support in CAESAR II for extracting forces and moments for the output result purposes.

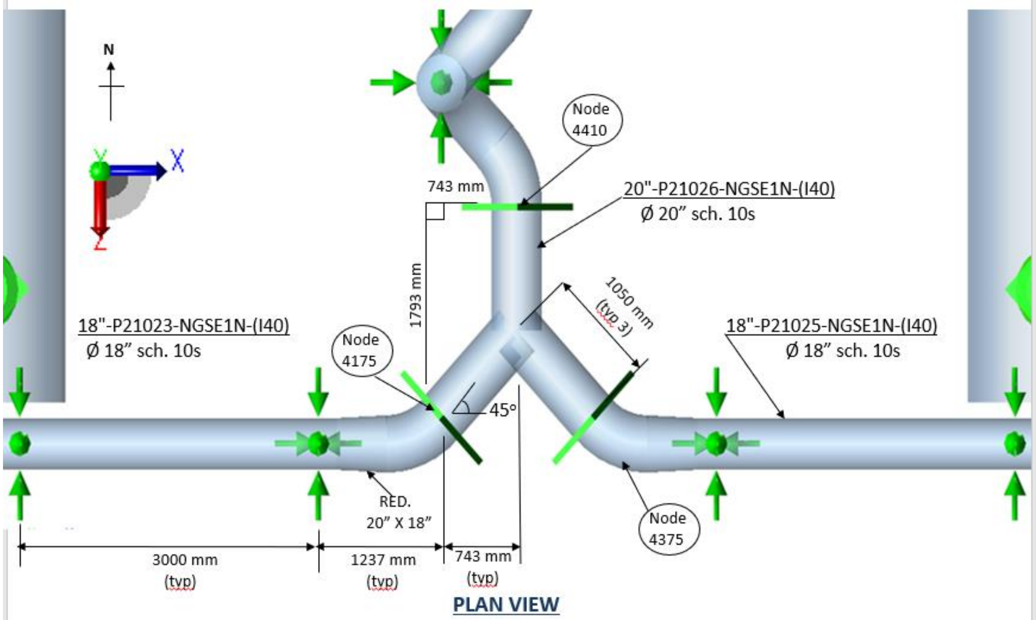


Fig. 4. CAESAR II plan view of the Y-tee

The calculated forces, moments, and the displacement of these three boundary conditions were extracted from CAESAR II restraint summary outputs and used for the local stress evaluation in ABAQUS. However, the modelling techniques in CAESAR II for the Y-tee did not follow the exact contour of the Y-tee geometry. This is the most common limitation in various beam type element software application. Thus, for the system to be accurately analyzed, the value of SIFs is needed.

The material properties used in the analysis for the piping system is shown as per Table 2. All of data properties for the materials are available from CAESAR II version 2019 database system.

The piping system operates below than 7000 cycles and therefore the factor *f* for the allowable stress is set to 1 and included in Eq. (2). Since the system is operating below than 7000 cycles, no fatigue considerations are performed to the piping system.

Amir and Faizul [15] have explained in great details of the steps involved when performing pipe stress analysis using CAESAR II software. Similar approach and method described in their paper was adapted in current work.

Table 2
Piping material properties

Specifications	Parameters
Pipe Size, inch	18, 20
Pipe Schedule, mm	4.78, 5.53
Insulation Thickness, mm	40
Insulation Density, kg/m^3	136.16
Fluid Density, kg/m^3	1064
Corrosion Allowance, mm	0
Class Rating, lb	150
Material (Pipe)	API 358 304L CL1
Material (Y-Tee)	A351 CF3

3.2 Finite Element Analysis (FEA) in ABAQUS

The FE model was developed in ABAQUS using the Y-tee geometry information as in Figure 5. The same material properties as in Table 2 were used in material data input. In this Y-tee model, a mesh element with structured hexahedral shape was selected as it is the most stable, and easier for nodes convergence during mesh post-processing stage. However, using the hexahedral mesh type can be somewhat complicated.

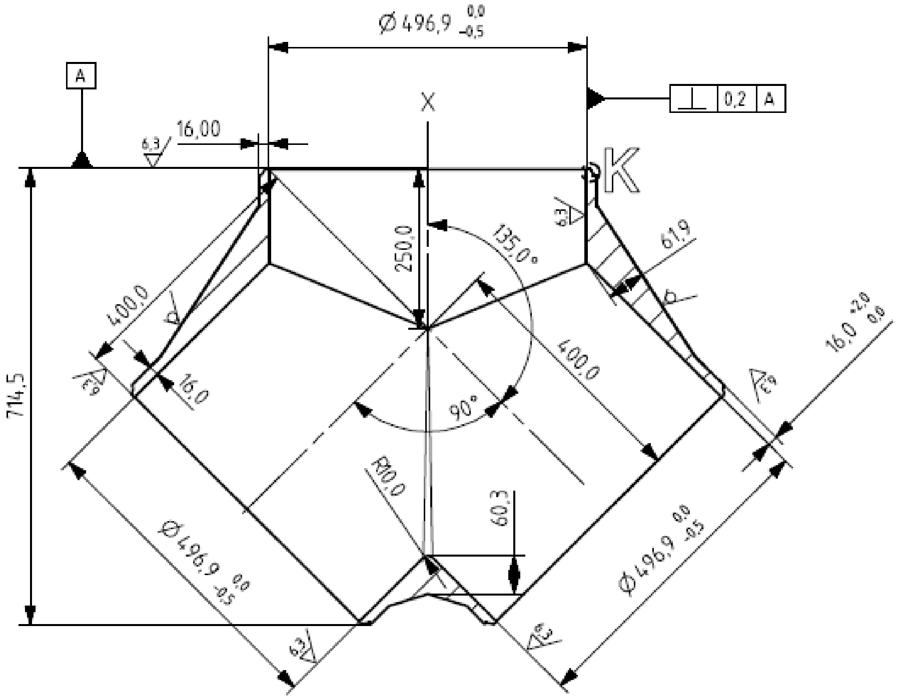
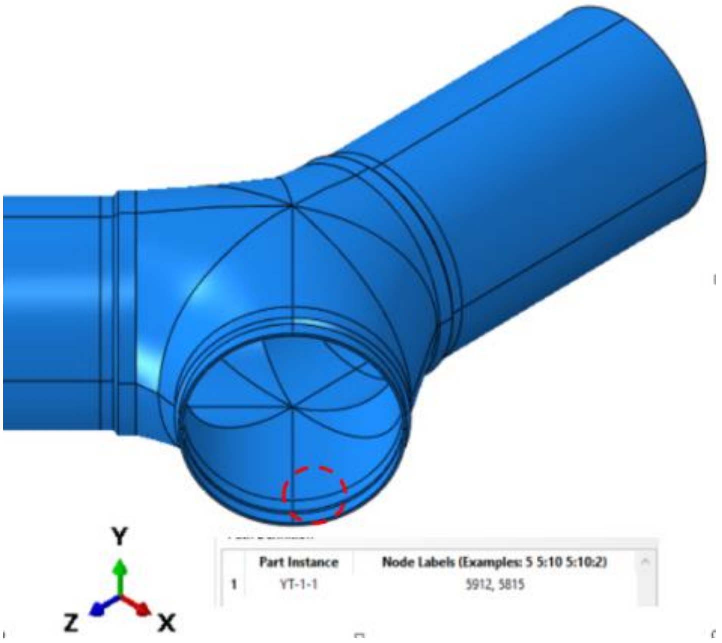


Fig. 5. Y-tee detailed geometries

Jason and Chris [16] explained in great details including several essential steps for mesh selection upon completion of the full solid primitive geometry model. In selecting the mesh size, all three crotch areas of the Y-tee should be smaller than the other elements in the model. These are the important stress concentration areas that will be evaluated against ASME VIII [7] requirements. The three pipes attached to the Y-tee utilized mesh size 30 with Single-Bias techniques to link it with a series of mesh sizes selected at the crotch areas.

Niemi [17] explained that stresses obtained in FEA simulations are highly dependable on its meshing type and size. To account for meshing dependency, a mesh sensitivity analysis was performed to identify the most suitable mesh size for the Y-tee model. A series of mesh sizes of 5, 8, 10, 13, 15, 18, 20, 23 have been developed at the 3 crotch areas of the Y-tee to be coupled with the connecting pipe. Internal pressure loading of 11 barg was imposed to the Y-tee and the graph stress versus mesh size was developed.

Validation of the Y-tee model was performed based on the comparison of the theoretical results in hoop and longitudinal stress against those obtained in the ABAQUS model. One of the pipe spools shown in Figure 6 that is connected to the Y-tee was selected for the validation steps. From the hexahedral mesh elements of the model, two geometric orders of linear and quadratic shape functions were selected for the analysis.



	Part Instance	Node Labels (Examples: 5 5:10 5:10:2)
1	YT-1-1	5912, 5815

Fig. 6. Model/mesh validation at weld joint

The type of loadings to be applied at nodes 4175, 4375 and 4410 were extracted from CAESAR II results as summarized in Table 3. The pipe internal pressure of 11 barg also included in the local model. For the local model in ABAQUS, the displacement loadings were used as the boundary condition at each node.

Table 3
Boundary condition results obtained from CAESAR II analysis at three locations (Operational Case)

Loading types	Vectors	Node 4175	Node 4375	Node 4410
	F_x	694	-8985	-8291
Force [N]	F_y	23070	951	15190
	F_z	-6516	-20256	-26772
	M_x	877	4068	-28198
Moment [Nm]	M_y	-1373	7315	1282
	M_z	-12982	-1029	-30433
	D_x	2.552	5.616	4.640
Displacement [mm]	D_y	-1.22	-0.638	-4.244
	D_z	-0.028	0.299	-3.297

Stress Classification Lines (SCL) was developed in accordance with the understanding of stress categories and the limitation of equivalent stress as specified in ASME VIII [7]. The identification of SCL lines for the Y-tee fittings is shown in Figure 7 based on recommendations made in [8] and [9]. The SCL lines for 1, 3 and 5 were mainly for evaluation of stresses on the pipe shell whereas lines 2, 4 and 6 were for welding lines on the piping side. On the other hand, the 7, 8 and 9 SCL lines were for welding lines of the Y-tee whilst 10, 11 and 12 were selected for the Y-tee crotch areas.

For computing the solution of the Y-tee model, a load combination as in ASME VIII [7] was defined. Subject to no occasional/environmental loadings such as wind, earthquake, snow, and ice, only Load Case 1 *(P + Ps + D)* and Load Case 3 *(P + Ps + D + T)* were used in the stress evaluations where P is internal and external pressure at 11 barg, *Ps* is design or operating static head which is equivalent to 0, *D* is imposed displacements as accordance to Table 3 and *T* is thermal loading at 135

^{0}C. Based on the stress categories as well as the load combination, a table with series of stress evaluations as defined in ASME VIII [7] was further developed.

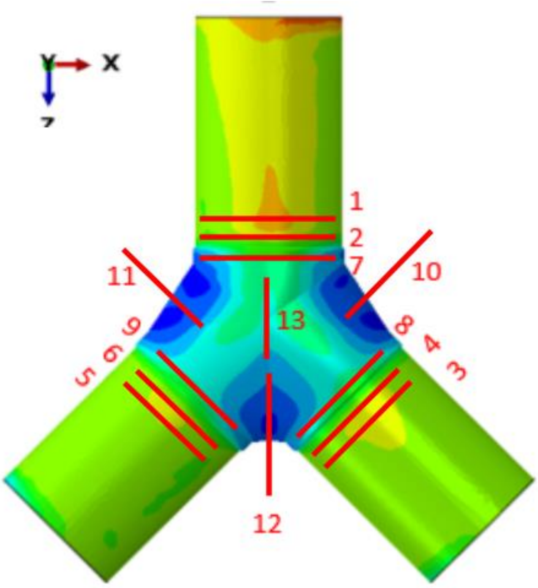


Fig. 7. Y-Tee SCL lines developed for Y-tee design

3.3 Optimum Design of Y-tee

The optimum design of the Y-Tee can be achieved by modifying the crotch radii (*i.e.,* SCL12 from Figure 7). Sensitivity analysis on the crotch radii was performed to determine the effect on the stress values, hence the optimum radii can be deduced. Four different angles were considered that included 45°, 50°, 55° and 60° as shown in Figure 8. Load Case 1 and 3 were considered, in which for both cases, the primary, secondary and peak stresses were extracted for comparison at each SCL lines.

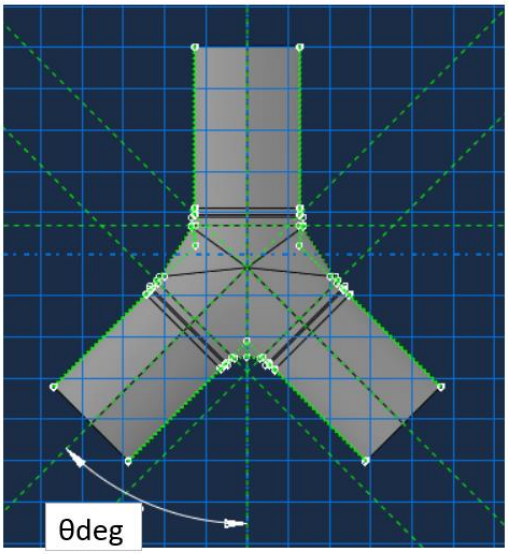


Fig. 8. Crotch radii angle defined by θ deg

Results of the selected each crotch radius angle shall be tabulated to determine the optimum design of the Y-tee. With the results, the correlations between crotch radius angle against stresses for the Y-tee components are demonstrated.

4. Results and Discussion

4.1 Mesh Sensitivity Analysis

From the mesh sensitivity analysis, two locations have been identified for sensitivity evaluations as shown in Figure 9.

For each location, the stress identification was between the internal and external wall of the Y-tee crotch areas. Figures 10 and 11 show the results obtained from mesh sensitivity analysis for various sizes of hexahedral mesh elements. From Figure 10, the result revealed that mesh element size between 5 to 11 was acceptable and applicable as the load variations were below than 10%. Whereas Figure 11 indicates that the acceptable mesh element sizes was between 7 to 11. Results for location L1 can be duplicated for the other side of Y-tee crotch area. Thus, element size 10 will be selected for both location L1 and L2 as well as the other crotch area for FE analysis.

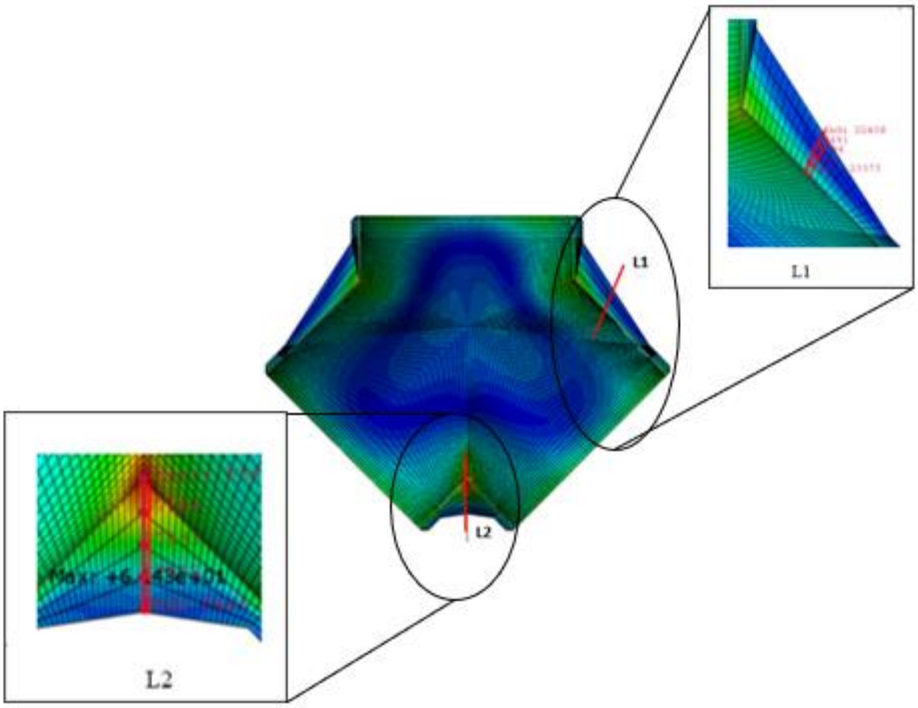


Fig. 9. Mesh sensitivity locations of L1 and L2

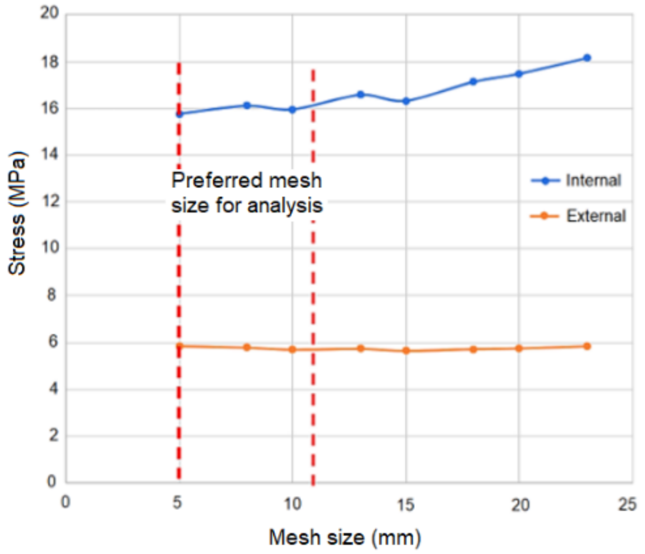


Fig. 10. Mesh sensitivity results at Location L1 of Y-tee

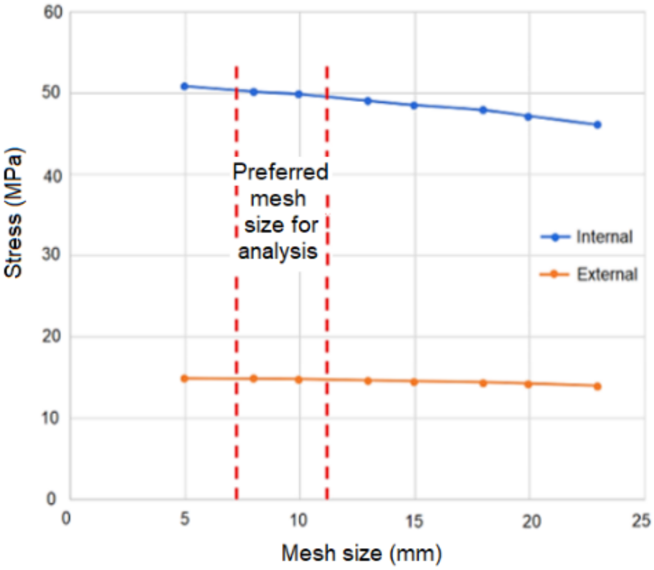


Fig. 11. Mesh sensitivity results at Location L2 of Y-tee

4.2 Optimum Design of Y-tee

Stress components from the FE analysis using *Hexahedral* elements with first and second geometrical order *i.e.,* linear, and quadratic functions were compared with the theoretical results obtained for both hoop and longitudinal stresses. The percentage differences were tabulated as in Table 4. The lowest variations of 3.6% were observed in the linear function. Thus, the linear type of formulation was selected for the FE analysis for better accuracy and computational cost.

Table 4
Results of the validation analysis

Components	Analytical (A)	FE Analysis		% Difference	
		Linear (L)	Quadratic (Q)	L vs A	Q vs A
Hoop stress [MPa]	44	45.60	46.30	3.60	5.20
Longitudinal stress [MPa]	22	23.55	24.46	7.00	11.20

4.3 Protection against Plastic Collapse

For protection against plastic collapse, there are two evaluations needed as per Load Case 1 and Load Case 3. ASME PTB-3 [18] recommends the results to be tabulated as per Table 5 and 6 for both cases. Noted that the simulation was performed at operating temperature of 135 °C.

Based on Figure 7, SCL 1, SCL 3 and SCL 5 were located on the pipe shell. The evaluation of the stresses is mainly focusing on the primary membrane stress, P_m due to pressure has excluded discontinuities. From Load Case 1 results, the average calculated von Mises stresses across SCL 1, SCL 2, SCL 3 lines were computed at 44.5 MPa, 44.7 MPa and 42.2 MPa, respectively against the allowable stress of 115 MPa.

Since the primary local stress, P_L was having similar allowable stress as combination between $P_L + P_b$, the evaluation of single P_L was not needed. Therefore, the rest of the SCL lines would be compared against allowable stress of 172.5 MPa based on combination between $P_L + P_b$. From Table 5 and 6, it shows that all SCLs stresses for Load Cases 1 and 3 were within the allowable limit. With these results, it is now confirmed that the current as-built Y-tee design has fulfilled the requirements of ASME VIII [7] for protection against plastic collapse for both load cases.

Table 5
SCLs results for Load Case 1 ($P + P_S + D$)

SCL No.	Location	Equivalent Linearized stresses (MPa)			Stress Evaluation Status	
		P_m	P_L	P_L+P_b	$P_m<S_m^*$	$P_L+P_b<1.5S_m^*$
1	20"-P21026 to Y-tee (away from discontinuity)	44.5	N/A	N/A	PASSED	N/A
2	20"-P21026 to Y-Tee	N/A	N/A	33.7	N/A	PASSED
3	18"-P21025 to Y-tee (away from discontinuity)	44.7	N/A	N/A	PASSED	N/A
4	18"-P21025 to Y-Tee	N/A	N/A	32.2	N/A	PASSED
5	18"-P21023 to Y-tee (away from discontinuity)	42.2	N/A	N/A	PASSED	N/A
6	18"-P21023 to Y-Tee	N/A	N/A	31.3	N/A	PASSED
7	Y-Tee to 20"-P21026	N/A	N/A	37.6	N/A	PASSED
8	Y-Tee to 18"-P21025	N/A	N/A	33.2	N/A	PASSED
9	Y-Tee to 18"-P21023	N/A	N/A	32.1	N/A	PASSED
10	Crotch Radius	N/A	N/A	44.6	N/A	PASSED
11	Crotch Radius	N/A	N/A	43.1	N/A	PASSED
12	Crotch Radius	N/A	N/A	62.1	N/A	PASSED
13	T-Tee (Centre)	N/A	N/A	22.6	N/A	PASSED

*S_m is allowable stress of 115MPa for A358 304L (Pipe) and 138 MPa for A351 CF3 (Y-Tee)

Table 6
SCLs results for Load Case 3 ($P+P_S+D+T$)

SCL No.	Location	Equivalent Linearized stresses (MPa)		Stress Evaluation Status	
		$P_L + P_b + Q$	P_L+P_b	$P_m<S_m^*$	$P_L+P_b<1.5S_m^*$
1	20"-P21026 to Y-tee (away from discontinuity)	141.3	150.6	PASSED	PASSED
2	20"-P21026 to Y-Tee	N/A	33.7	N/A	PASSED
3	18"-P21025 to Y-tee (away from discontinuity)	44.7	N/A	PASSED	N/A
4	18"-P21025 to Y-Tee	127.7	140.7	PASSED	PASSED
5	18"-P21023 to Y-tee (away from discontinuity)	42.2	N/A	PASSED	N/A
6	18"-P21023 to Y-Tee	N/A	31.3	N/A	PASSED
7	Y-Tee to 20"-P21026	191.6	211.8	PASSED	PASSED
8	Y-Tee to 18"-P21025	N/A	33.2	N/A	PASSED
9	Y-Tee to 18"-P21023	135.2	148.2	PASSED	PASSED
10	Crotch Radius	N/A	44.6	N/A	PASSED
11	Crotch Radius	97.7	110	PASSED	PASSED
12	Crotch Radius	N/A	62.1	N/A	PASSED
13	T-Tee (Centre)	113.1	129.1	PASSED	PASSED

*S_{PS} is allowable limit on the primary plus secondary stress range of 345 MPa for A358 304L (Pipe) and 414 MPa for A351 CF3 (Y-Tee).
*S_a is alternating stress limit of 482 MPa.

4.4 Protection against Local Failure

From Eq. (5) of ASME VIII [7], the Protection against Local Failure is referring to the highest principal stress that occurred in one of the SCL 1 to SCL 13. The result of the evaluation is tabulated in Table 7 for both Load Case 1 and 3.

From Table 7, SCL 12 for Load Case 1 showed that the results for σ_1 was 65.32 MPa, σ_2 (6.23 MPa) and σ_3 (0.54 MPa) whilst SCL 8 for Load Case 3, σ_1 was computed at 220.32 MPa, σ_2 (42.40 MPa) and σ_3 (4.00 MPa). Based on Eq. (5), the summation of all principal stresses for SCL 12 and SCL 8 are 72.08 MPa and 266.72 MPa, respectively. These two values were then compared against the allowable stress at 552 MPa. It was found that both results were well below the allowable stress.

Based on the results, the current as-built Y-tee design has satisfied the requirements of ASME VIII [6] for protection against local failure for both Load Case 1 and 3.

Table 7
Results for protection against local failure assessment

Load Case	Highest Stress, SCL Line		Principal Stress, MPa	Allowable stress, MPa	Status
1	SCL 12	σ_1	65.32		
		σ_2	6.23		
		σ_3	0.54		
		$\sigma_1 + \sigma_2 + \sigma_3$	72.08	552	PASSED
3	SCL 8	σ_1	220.32		
		σ_2	42.40		
		σ_3	4.00		
		$\sigma_1 + \sigma_2 + \sigma_3$	266.72	552	PASSED

4.5 Optimum Design for Y-tee

An optimum design study has been performed for Y-tee by varying the crotch radii angle as shown in Figure 8. The effect of varying crotch radii angle on the primary and secondary stresses at SCL lines was determined. From the study, clear trend of stress reduction was observed when the Y-tee angle varied from 45° to 60° for Load Case 1. The stresses, $P_L + P_b + Q$ were at minimum when crotch radii angle at 60°. The trend was observed at all SCL lines as shown in Figure 12.

For Load Case 2, the effects of crotch radii angle variation were assessed in terms of S_{PS} $(P_L + P_b + Q)$ and S_a $(P_L + P_b + Q + F)$ values as depicted in Figure 13 and 14, respectively. As the crotch radii angle increased from 45° to 55°, there was an increase trend observed for S_{PS} and S_a stresses. At 60°, the stress values were reduced to the stress values of 45° Y-tee. Hence, both 45° and 60° Y-tee showed minimum stress values at SCL lines. For Y-tee design, in order to satisfy the plastic collapse and local collapse assessment, the S_{PS} and S_{PL} $(P_L + P_b)$ shall be at minimum whereas for the protection against cyclic, the S_a would be at minimum.

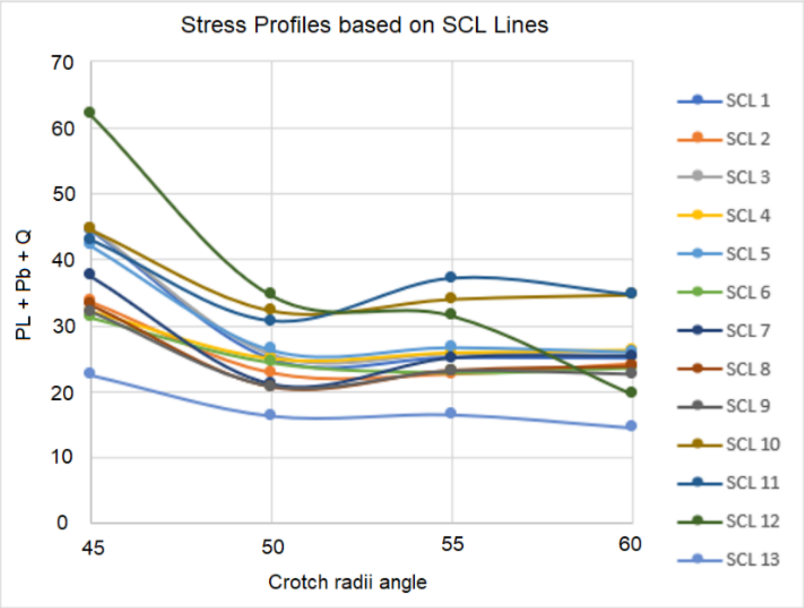


Fig. 12. The variation of primary and secondary stresses, $P_L + P_b + Q$ vs. the crotch radii angle obtained from Load Case 1

With the step-by-step methodology explained in this paper with proof of results and findings from the analysis, the design process of the Y-tee can be further optimized instead of using the conventional approach of factory testing. The approach of SCLs lines developed in current work provided a method of evaluating the integrity of Y-tee design in accordance with the ASME standard. Furthermore, the current work also shows that the Y-tee design can be further improved to suit the application and compliance to the codes and standards.

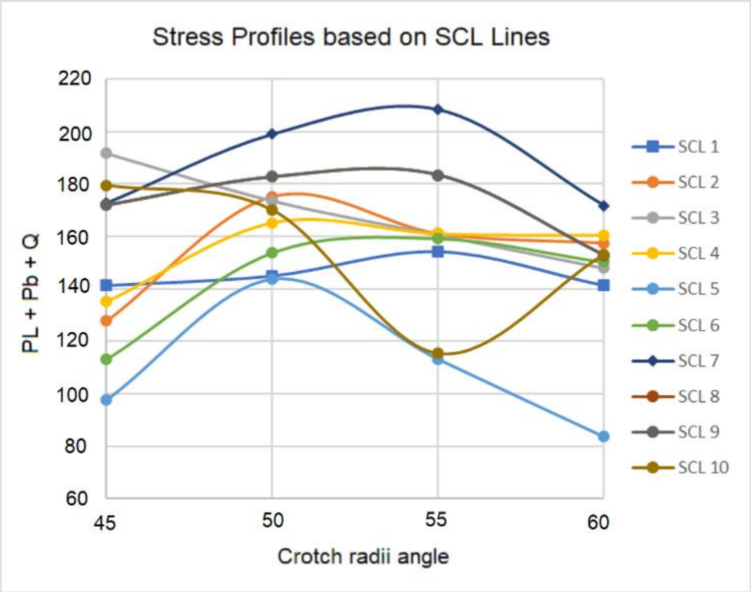


Fig. 13. The variation of primary and secondary stresses, $P_L + P_b + Q$ vs. the crotch radii angle obtained from Load Case 2

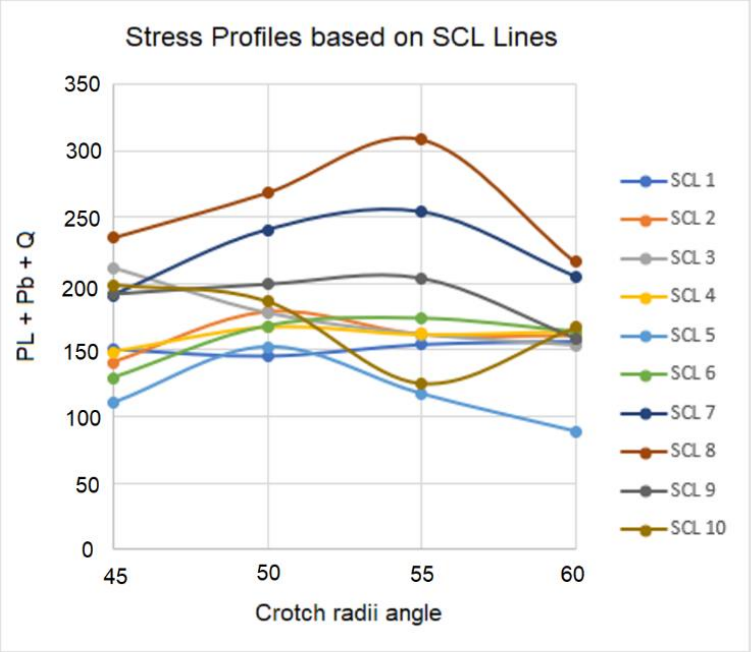


Fig. 14. The variation of primary and secondary stresses, $P_L + P_b + Q + F$ vs. the crotch radii angle obtained from Load Case 2

5. Conclusions

A computational analysis on the non-standard Y-tee fittings has been performed to evaluate the stress distribution and integrity of the design. Some concluding observations from the study are given below.

i. The conventional method of stress analysis of the non-standard Y-tee fittings using CAESER II often requires assumption of SIFs which only available by laboratory testing.

ii. The developed SCLs line for the non-standard Y-tee fittings provided guides for stress assessment against design stress allowable in accordance with ASME VIII Div.2 Part 5.
iii. The optimum design of non-standard Y-tee was found at 45° and 60° crotch radii which resulted in lower stress value at SCL lines.

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