1.0. Summary

The analysis of the pipe with fines is modeled using Abaqus software to obtain results for temperature distribution as well as stress distribution by excluding and including the thermal part. The study involved two different types of element types such as DC2D4 (Thermal problem) and CPS4I (Stress problem), finite element method. A parametric mesh density study was performed with six different element breakups in order to obtain converged solution. Results showed the higher element meshes generated better approximations to the actual solution. Even though the coarser meshes were too stiff and underestimated, its results were almost approximately equal to the finer grids. Grid six with number of nodes 16997 was selected in both studies. To obtain the dominant part in the problem, the stress problem was run separately to get how much stress we have from the pressure and then the same problem was run by importing the data from the thermal problem. It is observed that the dominant part is the thermal part.

1.1. Introduction

There are numerous situations where heat is to be transferred between a fluid and a surface. In such cases the heat flow depends on three factors namely (*i*) area of the surface (*ii*) Temperature difference and (*iii*) the convective heat transfer coefficient. The base surface area is limited by design of the system. The temperature difference depends on the process and cannot be altered. The only choice appears to be the convection heat transfer coefficient and this also cannot be increased beyond a certain value. Any such increase will be at the expense of power for fans or pumps. Thus the possible option is to increase the base area by the so called extended surfaces or fins. The situation is depicted in Fig. 4.1. The fins extend form the base surface and provide additional convection area for the heat conducted into the fin at the base. Fins are thus used whenever the available surface area is found insufficient to transfer the required quantity of heat with the available temperature deep and heat transfer coefficient. In the case of fins the direction of heat transfer by convection is perpendicular to the direction of conduction flow. The conduction in fins is considered to be one dimensional though it is essentially two dimensional. This is acceptable as the length along the fin is much larger to the transverse length. The process of heat transfer with fins is often termed as combined conduction convection systems. Common examples of the use of extended surfaces are in cylinder heads of air cooled engines and compressors and on electric motor bodies. In air conditioners and radiators tubes with circumferential fins are used to increase the heat flow. Electronic chips cannot function without use of fins to dissipate the heat generated. Several shapes of fins are in use. These are (*i*) Plate fins of constant sectional area (*ii*) Plate fins of variable sectional area (*iii*) Annular or circumferential fins constant thickness (*iv*) Annular fins of variable thickness (*v*) Pin fins of constant sectional area and (*vi*) Pin fins of variable sectional area. Some of these are shown in Fig. 4.1. The main aim of the study is to design fins to optimize the use of a given amount of material to maximize heat transfer. For this purpose it will be desirable that the fin surface temperature is closer to the base surface temperature. This can be achieved by the use of materials of high thermal conductivity like copper or aluminum. In terms of weight and ease of lubrication aluminum will score over copper though its thermal conductivity will be lower. It will be shown later that there are limitations about the length of the fin in terms of effectiveness of the material used. In order to increase the area for a given volume, thinner fins should be chosen. Fins are found more valuable when the convective heat transfer coefficient is low. This is the case in the case of gas flow and natural convection and fins are more commonly used in these cases [1].

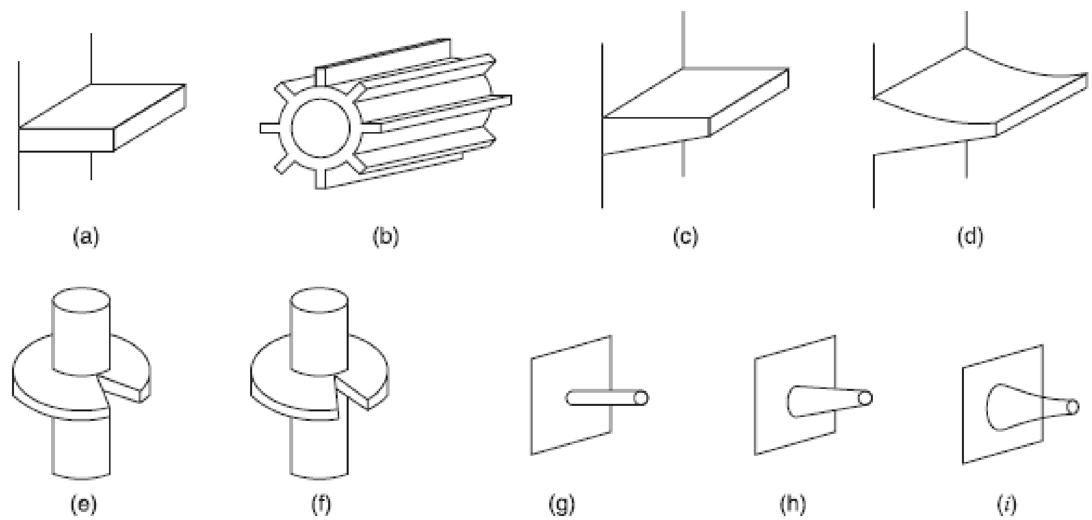


Figure 1. Schematic Diagrams of Different Types of Fins: (a) Longitudinal Fin of Rectangular Profile; (b) Cylindrical Tube with Fins of Rectangular Profile; (c) Longitudinal Fin of Trapezoidal Profile; (d) Longitudinal Fin of Parabolic Profile; (e) Cylindrical Tube with Radial Fin of Rectangular Profile; (f) Cylindrical Tube with Radial Fin of Truncated Conical Profile; (g) Cylindrical Pin Fin; (h) Truncated Conical Spine; (i) Parabolic Spine.

1.2. Problem Description

The parameters of the heat transfer problem to be analyzed are shown in figure 1.

1.2.1 Material Properties

The properties of the model are shown in Table 1.

Table 1: Material Properties for model 2.

Material Properties	Values
Elastic modulus, E	$30\ MPsi$
Poisson's Ratio, v	0.2
Thermal conductivity, K	$0.2\ \frac{BTU}{s.in.F}$
Coefficient of thermal expansion, α	$0.2 \times 10^{-5}/F$

1.2.2 Model in more details

Figure (2) shows the problem in more details.

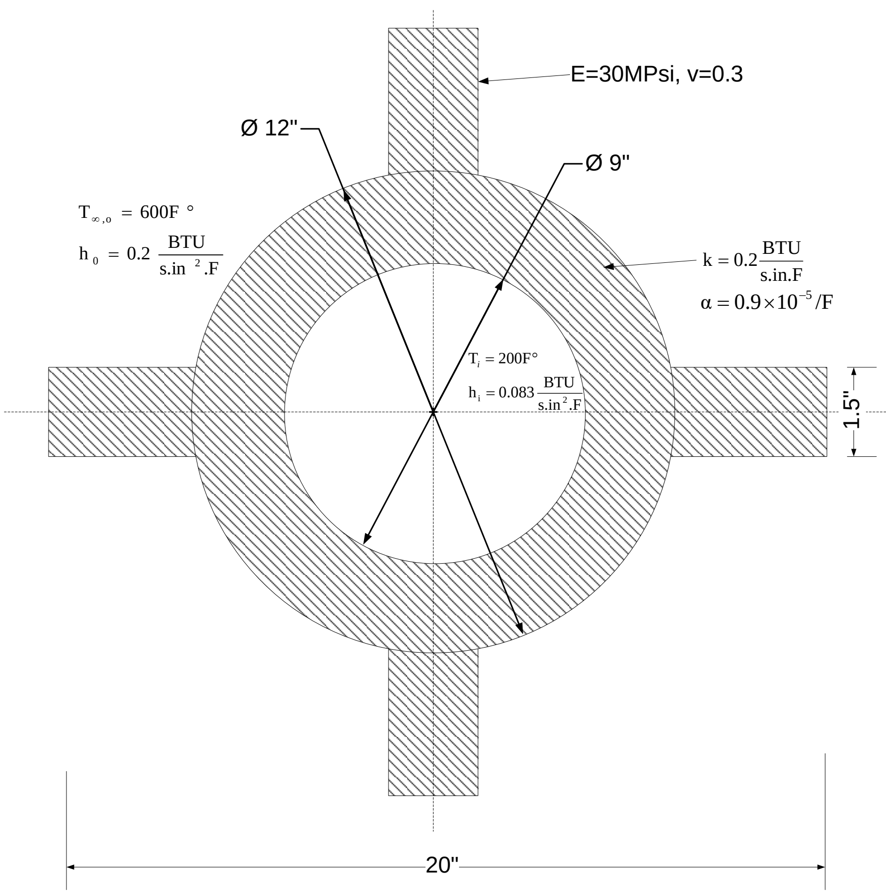


Figure 2: Problem Description.

1.2.3 Boundary Conditions

Two boundary conditions were applied to the problem in order to obtain whether most of the stress comes from thermal part or pressure. Table 2 shows boundary conditions applied to the model.

Table 2: Boundary conditions applied to the model.

First boundary conditions	Inlet	Outlet
Temperature	$200F°$	$600F°$
Convection heat transfer coefficient	$0.083\, BTU/s.in^2.F$	$0.2\ BTU/s.in^2.F$
Second boundary conditions		
Pressure	$1200psi$	Proper boundary

1.3. Coarse Grid

A 4432 Quad-dominated mesh was used in this validation study, and is shown in Figure 2.

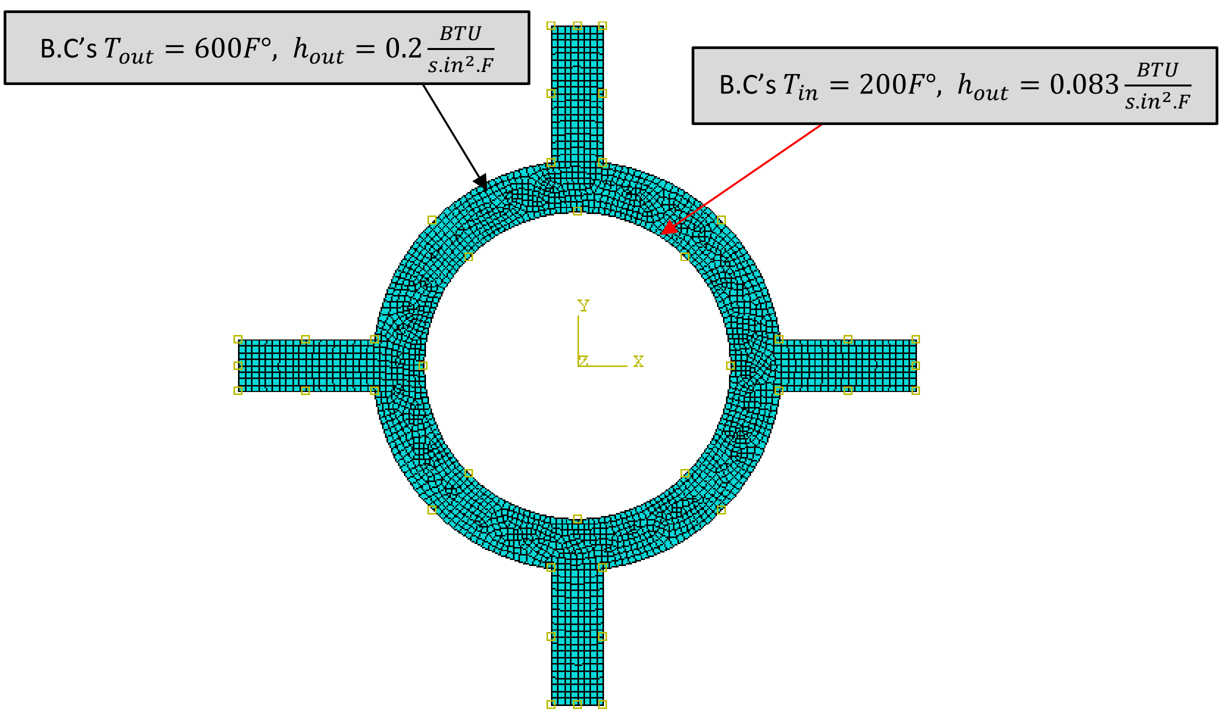


Figure 3: Quad-dominated mesh using first boundary conditions.

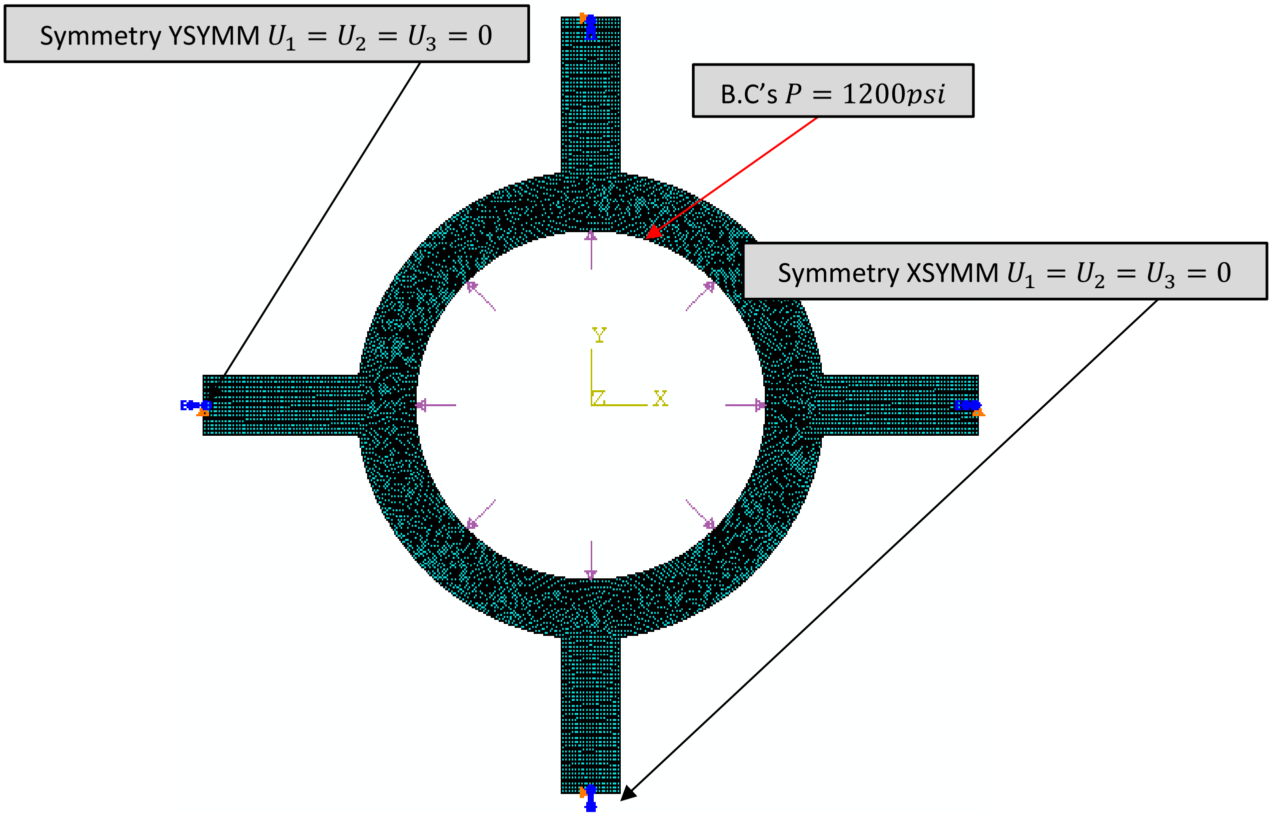


Figure 4: Quad-dominated mesh using second boundary conditions for number of elements 16997.

1.4. Case Setup

The Abaqus case was set up using constant fluid properties and boundary conditions described in Section 1.3.3 and 1.4. Two runs were made after getting the converged solution 2.

1.5. Calculation

The element type that was selected in the heat problem was DC2D4 and in the stress problem was CPS4I (Incompatible mode) to avoid singularity.

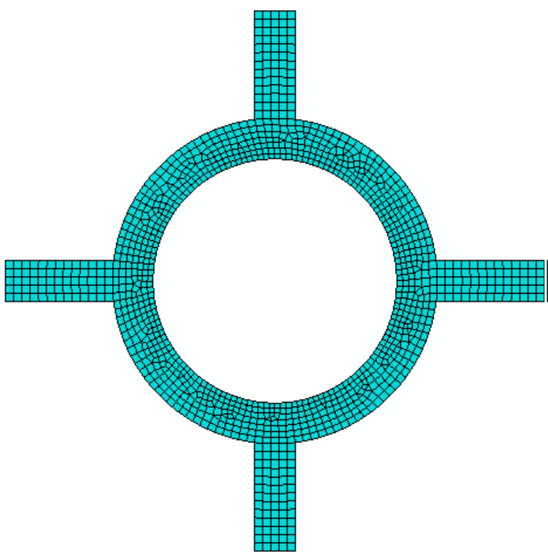
1.6. Grid Convergence Study

Six different resolutions figure (5) were made for the same family of meshing. Based on the selected grids, Abaqus was run on each grid to obtain the value in which we decided to make our study. In our case, the maximum temperature NT11 was measured for each grid resolution table 4. Now, our aim is to obtain the value at zero mesh spacing. To do so, we have to do curve fit to our four values and that leads us to the maximum temperature at zero mesh spacing, which in our case is approximately $NT11 = 599.09$.

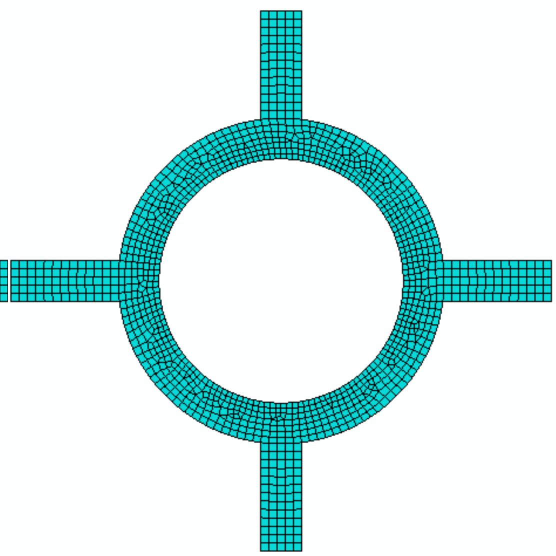
Table 4: gives the results for grid convergence for different number of nodes

	Approximate global size	**Number of elements**	$Max - NT11, F°$	$Min - NT11, F°$	Max-HFL1
Coarse Mesh	0.3	1190	599.082153	416.578247	17.638514
Coarse Mesh	0.2	2494	599.097351	416.561035	17.757521
Coarse Mesh	0.15	4432	599.103516	416.540558	17.788774
Medium mesh 1	0.1	8979	599.108704	416.530823	17.856194
fine mesh 1	0.09	12389	599.109558	416.529907	18.159552
Very fine mesh 2	0.08	16997	599.110474	416.526520	18.738518

Number of elements 1190



Number of elements 2494



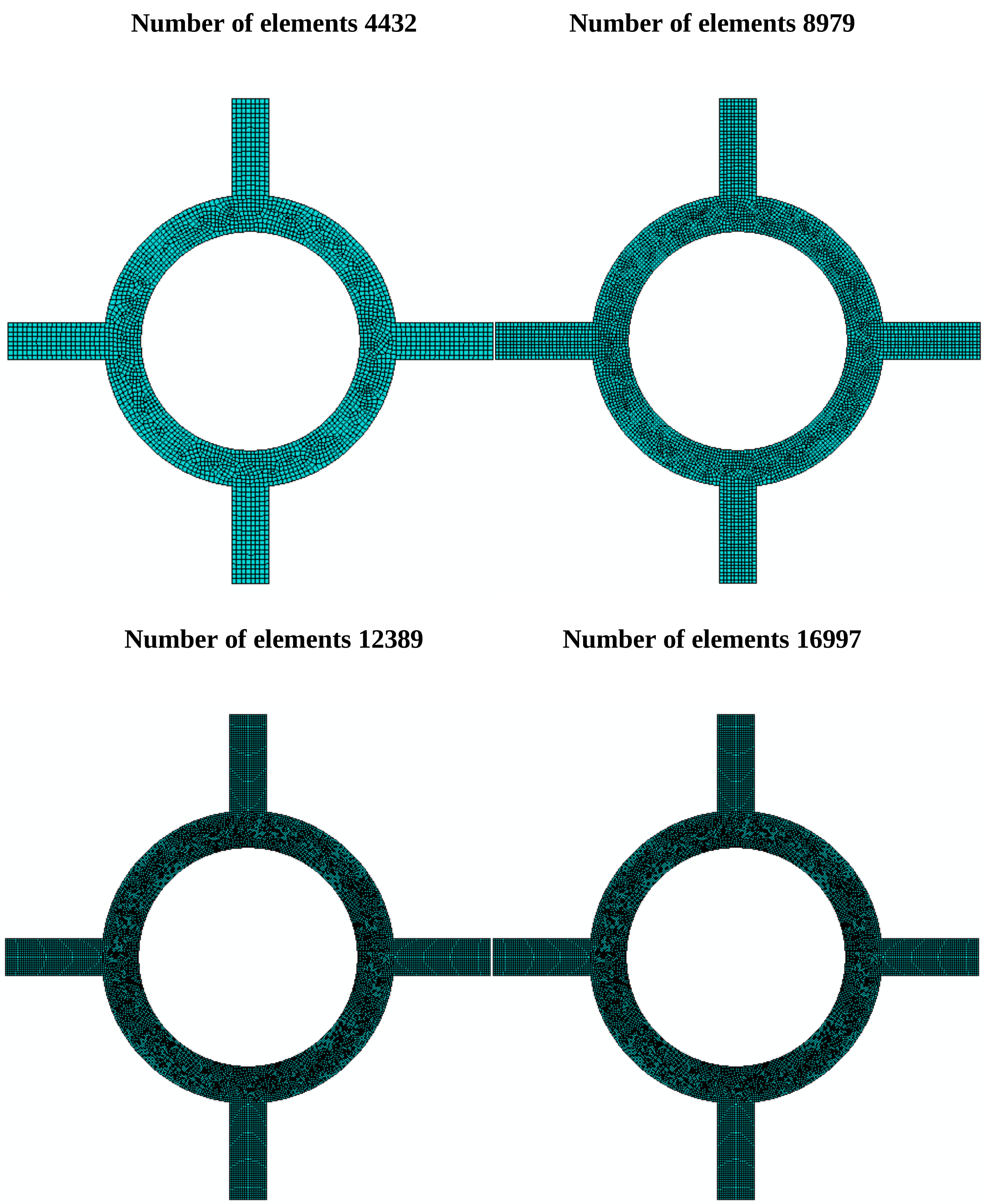


Figure 5: shows coarse, medium, and fine mesh for the same family of grid.

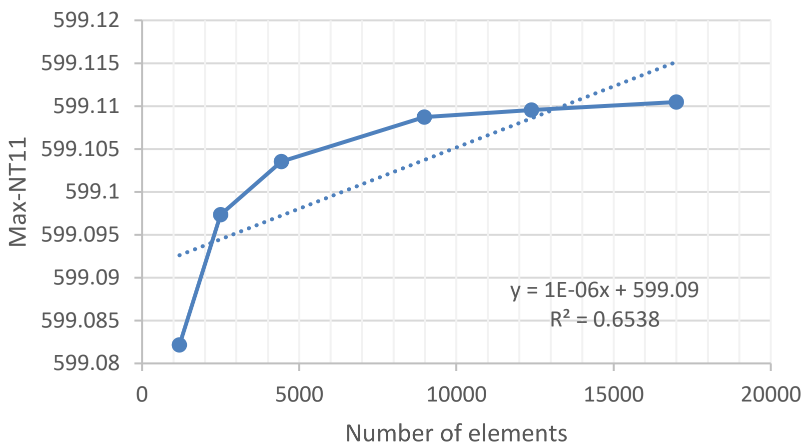


Figure 6: Grid Convergence for the pipe with fines.

1.7. Results

1.7.1 Heat problem

As we mentioned before, a grid convergence study was performed in order to obtain the converged solution. In this study, the very fine mesh was chosen with number of elements of 16997 to get the required results such as the temperature distribution and heat flux etc. Figure (7) shows the temperature distribution.

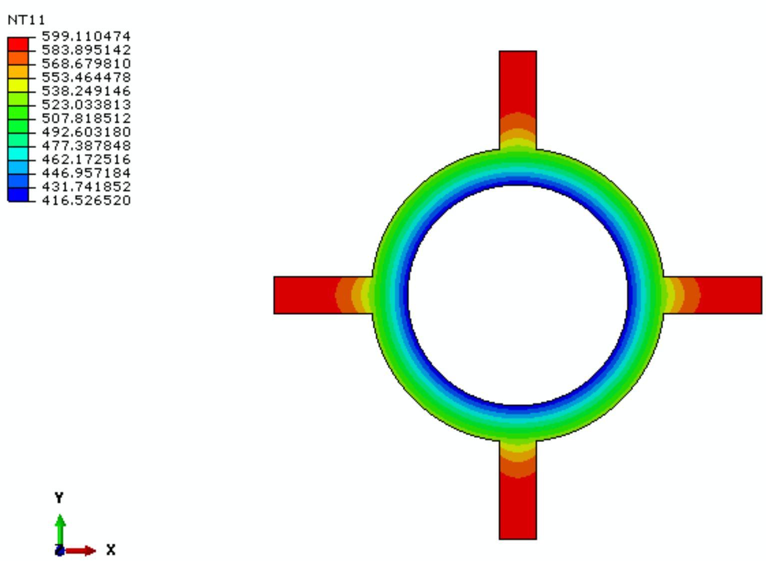


Figure 7: temperature distribution for number of elements 16997.

1.7.2 Stress Problem

In order to obtain which the dominant part in model 2 problem, we have to run each problem separately. In model 2, pressure boundary condition at the pipe was provided, and our first aim was to see how the pressure is going to effect the pipe. In this case, we have to provide extra boundary condones as shown in figure 4 in which we fixed both fines at the top and bottom as well as the lift and right fines. Figure (8) shows stress distribution results around the pipe and fines.

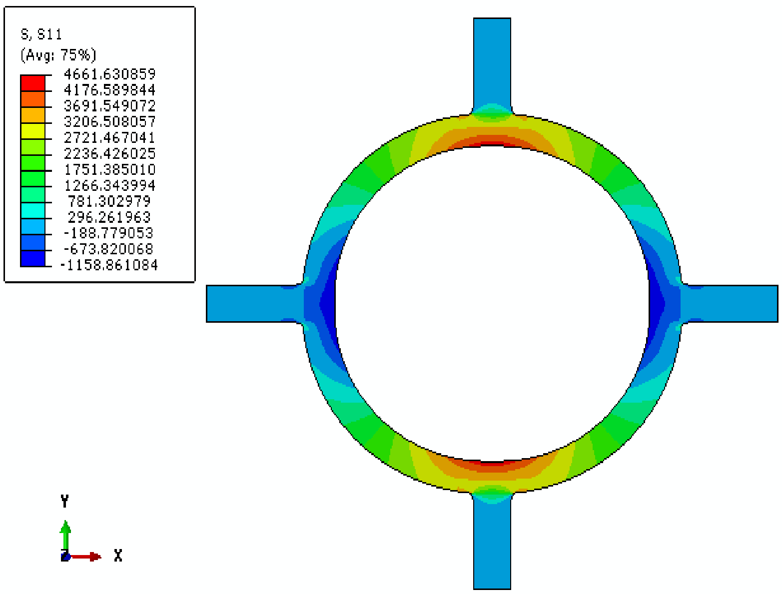


Figure 8: stress distribution for number of elements 16997.

1.7.3 Stress and Thermal

In this problem, our aim is to see which the dominant part in our problem. Therefore, the data such as the temperature distribution for number of elements 16997 was imported to the stress problem, and Abaqus was run in this case. One of the most significant things that was observed is that the stress has increased tremendously, which tells that the dominant part is the thermal, and we can see that by comparing both figures 8, 9 respectively.

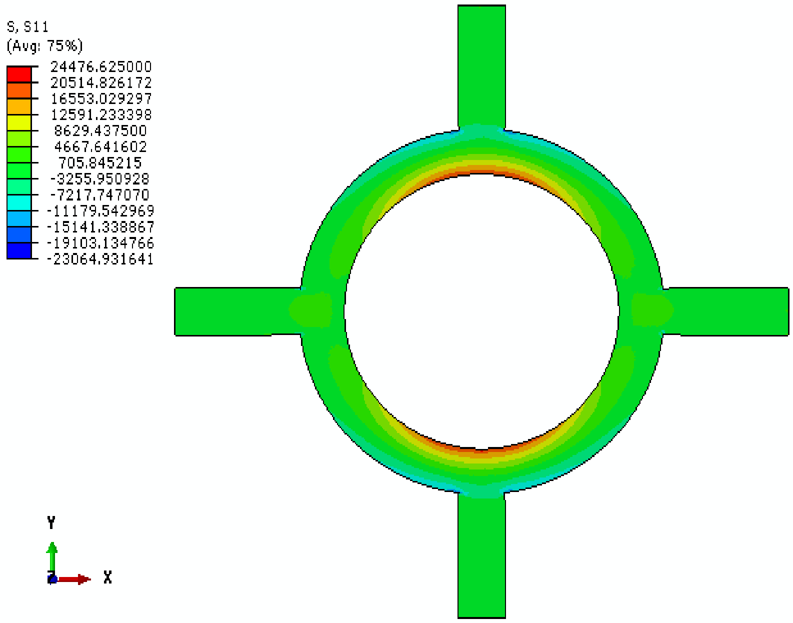


Figure 9: shows stress distributions adding the thermal part for number of elements 16997.

$$Max[S, S_{11}]_{Stress\ problem} = 4661.630859\ psi$$
$$Max[S, S_{11}]_{Stress\ problem+thermal} = 24476.625\ psi$$

1.8. Conclusion

Abaqus has been tested for a good example to compare results by adding temperature to the stress analysis. While the study for the stress problem shows that the maximum are given by $4712.8866211\ psi$, the stress distribution has increased dramatically when we added the thermal part. This proves that the thermal is the dominant part.

1.9. References

1. Incropera DeWitt VBergham Lavine 2007, Introduction to Heat Transfer, 5th Ed.