STRESS ANALYSIS OF A POLYMER EXTRUSION DIE
USING
FINITE ELEMENT METHOD,

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by
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CHAPTER I

Introduction

Different types of processes can be used to obtain polymer ribbons with improved mechanical properties, high orientation, elevated melting temperature, and transparency. Some of these processes are; superdrawn filament process (1,2), radial compression method (3), solid state extrusion (4-14), and fixed boundary extrusion process. Although the first three processes are used successfully, they are not feasible in mass industry due to their need to operate at high pressures (around 2000 atmospheres)*, and low rate of output.

Collier and coworkers (15-21) have been designing different types of dies to investigate the development of polymer properties by using a single screw plasticating extruder with specially designed and operated dies employing "Fixed Boundary Extrusion." It was reported that at reasonable pressures, they obtained highly oriented, high modulus, and transparent ribbons of polypropylene and

* 1 atm. = 14.696 psi. = 1.0133 x 10^5 N./sq.m. = 0.10133 MPa.
polyethylene.

Some of these specially designed and operated dies were designed to be used for uniaxial flow and others for biaxial flow. All of the output of these dies are cooled externally with vertical extrusion but one (21), which was designed for horizontal extrusion with an internal cooling system to cool the polymer in the land section. This cooling minimizes the die swell phenomena, controls the temperature of the polymer in the land section of the die, and improves extruder operating conditions.

A major mechanical problem occurred in the previously designed biaxial die using horizontal extrusion (21) because of the internal cooling section, and badly designed supporting bolts. When leakage develops at the contact surfaces of the two parts of a die, the whole process should be stopped. Even though the leakage occurred in weak points where the stresses are high (i.e. near the shaping section), a comprehensive mechanical design of a new die was needed. A comprehensive study involving the stress analysis of die due to pressure, temperature changes (internal cooling in the land section), and torques taken by the supporting bolts and nuts is needed for dies with extrusion pressures between 270-680 atmospheres.
One of the successful techniques which can be used to analyze the stresses in the die is the "Finite Element Method." Its first appearance (22,23) was in the 1950's, when it was used to solve solid and structural mechanics problems. The development of the matrix structural analysis (24) method between 1945 - 1955 presented an efficient solution technique that could be used on digital computers. This helped Argyris and Kelsey 1955 (25) to recognize and generalize their work on the technique of the finite element method. The first appearance of the name "finite-element method" was in the paper of Turner, Clough, Martin and Topp 1956 (26), which solved a plane stress problem of a structure subdivided into triangular elements whose properties were determined from the theory of elasticity equations. It was recognized in 1963 (27-30) that the finite element method was a variation of the Ritz method. In 1965 Zienkiewicz and Cheung (31-32) reported that the finite element method could be applied to conduction heat transfer.

In the summer of 1979 (33) ELAS 75 (34-36), a general purpose digital computer program for solving linear equilibrium problems of structures, was implemented on the Ohio University IBM 370/158. This finite element analysis computer program would be used to achieve the objectives of analyzing the new uniaxial die. Fortunately, there is
another finite element analysis program available in the Civil Engineering Department (37) of Ohio University, which can be used to check the results from the two dimension ELAS program.

Because the ELAS program at Ohio University did not include a plotting subroutine, and there was a need for mesh plotting, two computer programs were written in appendix D. One of these computer programs is for a two dimensional mesh, and the other is for a three dimensional mesh. Most of the significant stress results were plotted using contour graphics computer programs and subroutines (38,39) which are available in the Chemical Engineering Department.

The Octahedral shearing-stress failure theory was used in the die analysis since it gives the most accurate results when compared to experimental data for failure (40). The octahedral shearing-stress is defined as:

$$\tau_{oct} = \frac{1}{3} \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)}$$

in terms of yield stress:
\[ \tau_{\text{oct}} = \frac{\sqrt{2}}{3} \, Se \]

where \( \sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{yz} \) and \( \tau_{zx} \) are the stress components at a point in the direction of \( x, y \) and \( z \) coordinates.

\( \tau_{\text{oct}} \) is the octahedral shearing-stress.

\( Se \) is the effective stress, or the yield stress with a factor of safety 1.5 included.

The die is being used on thesis projects by some of the graduate students in the Chemical Engineering Department, that is continuing the work done on polymer extrusion by the fixed boundary process. The purpose of the die is to measure the temperature, pressure gradients, and the crystal structure in different locations of the die while extruding polymer at different conditions. This will be done by inserting transducers, and passing a Laser beam (or any high intensity light) through glass windows. Therefore, the purpose of the stress analysis to be performed, was to determine the maximum pressure that can be used to prevent the leakage of polymer from the two contact surfaces of the die, and to check whether the glass window will break.
CHAPTER II

Basic Design Considerations

2.1 The functions of the die with a general description.

Most of the dimensions of the die were furnished by the Chemical Engineering Department (38). As mentioned in chapter I, the purpose of the die is to extrude melted polymer horizontally using a single screw plasticating extruder with a constant rate of flow. A maximum pressure was found by the calculations and testing. In addition it was necessary to maintain the temperature above the melting point of the polymer. A die was used having a shaping section with a draw ratio of 6:1. A cooling system was included in the land section (which follows the shaping section) to cool the polymer and minimize the loss of stresses due to die swell phenomena.

The die was constructed to measure the temperature and pressure gradients between the entrance section of the die, the beginning of the shaping section, and the beginning of the land section. Therefore, it was decided to use six combined temperature and pressure transducers type (TPT 432A) (41), three on each part of the die. Another major
purpose for this die is to measure and compare the changes that will occur in crystalization, orientation, and structure of the polymer by using a laser beam or any high intensity light passing through glass windows, which are inserted inside the die and supported by hollow plugs. Six glass windows were included in the die, three in each part of the die located beside the above mentioned transducers.

The glass window 1.25 in. (3.175 cm.) in length, 0.5 in. (1.27 cm.) in large diameter, 0.25 in. (0.635 cm.) in small diameter, of type 7720 Borosilicate Pyrex (42) was used with teflon (PTFE) washers 1/16 in. (0.16 cm.) thick on the top of the glass and 1/32 in. (0.08 cm.) thick on the bottom (43). The teflon washers help to prevent the breakage of the glass and to seal those regions of the die. The two parts of the die should be connected by sets of bolts, nuts and washers to prevent leakage between their surfaces. Heat treatment of the die was not recommended (44) since there is a lack of symmetry in die geometry due to different sizes of holes.

It was recommended (45) that the inside surfaces of the channel have a surface roughness of 16 μin. in order for the polymer to pass through the die with sufficiently low friction at the interfaces of the channel surfaces. Due to the function of the channel, the direction of grinding
should be parallel to the direction of flow. The contact surfaces of the die should have a good surface finish in order to have proper sealing. Therefore, a 16 μin. surface roughness was used with the direction of grinding multidirectional. The rest of the surfaces do not need a fine surface finish because of their relative insignificance. The clearances, allowances and tolerances were given according to the type of machining and the importance of that part (44,45). The die was connected to the extruding system by means of an adapter which was used in a horizontal extrudate die.

Figure 1 shows the diagram of one part of the die including the glass windows and teflon washers fixed by the hollow plugs. There is a double size figure included appendix A, in which the dimensions can be read clearly. Figures 2-5 show photographs of the two parts of the die assembled and taken apart, and with transducers or without. Different views are shown to help in understanding the structure of the die and its accessories.

2.2 Materials selected and their mechanical properties

a) The body of the die

The material which was chosen for the body of the
Figure 1. A Diagram of Half a Die.
Figure 2. A Photograph Showing the Top and Bottom of the Die.

Figure 3. A Photograph Showing the Two Halves of the Die Together.
Figure 4. A Photograph Showing Die Entrance and Transducer Location.

Figure 5. A Photograph Showing Another View of the Assembled Die.
die was AISI 410 stainless steel, because it was readily available. The mechanical properties of AISI 410 stainless steel (46) at 400°F (204°C) are:

\[ E = 30 \times 10^6 \text{ lb./sq. in.} \quad (20.7 \times 10^{10} \text{ N./sq. m.}) \]
\[ \nu = 0.3 \]
\[ \alpha = 6.2 \times 10^{-6} \text{ in./in. (F)} \quad (11.2 \times 10^{-6} \text{ m./m. (K)}) \]
\[ Sy = 40,000 \text{ lb./sq. in.} \quad (27.6 \times 10^7 \text{ N./sq. m.}) \]
\[ Se = 26,600 \text{ lb./sq. in.} \quad (18.4 \times 10^7 \text{ N./sq. m.}) \]

Where \( E \) is the modulus of elasticity, \( \nu \) is the Poisson's ratio, \( \alpha \) is the coefficient of thermal expansion, \( Sy \) is the yield stress of the material, and \( Se \) is the effective stress (or the yield stress with a factor of safety 1.5 included).

b) **Supporting bolts, nuts and washers**

To eliminate the effect of differential thermal expansion between the bolts and the die arising during the heating of the die from room temperature to the operating temperature (400°F or 204°C), bolts were used that were also made from AISI 410 stainless steel. The bolts, nuts and washers were heat treated to assure high yield strength as follows (46):

\[ Sy = 135,000 \text{ lb./sq. in.} \quad (93 \times 10^7 \text{ N./sq. m.}) \]
\[ \sigma_e = 90,000 \text{ lb./sq. in.} \quad \left( 6.2 \times 10^7 \text{ N./sq. m.} \right) \]

c) Glass window plugs

For the glass window plugs, it was obviously reasonable to use the same type of material that was used for the die, and for the same reason as above. This should minimize the breakage of the glass windows and also simplify the stress analysis calculations in the region of the die.

d) Glass windows

The type of glass which was used for the windows was Corning Glass code 7720 Borosilicate (42) which has the following mechanical properties:

\[
\begin{align*}
E &= 9.1 \times 10^6 \text{ lb./sq. in.} \quad \left( 6.27 \times 10^{10} \text{ N./sq. m.} \right) \\
\nu &= 0.2 \\
\alpha &= 2.0 \times 10^{-6} \text{ in./in.}(F) \quad \left( 3.6 \times 10^{-6} \text{ m./m.}(C) \right)
\end{align*}
\]

Tensile working stress = 4,000 lb./sq. in. \quad \left( 28 \times 10^6 \text{ N./sq. m.} \right), safety factor is already included.

The glass windows are the weakest parts of the die since the material can not bear high tensile stresses, but it can withstand very high compressive and shear stresses (42).
e) Window washers

It was recommended that teflon® washers be used in order to avoid the breakage of the glass windows due to tightening by distributing the load on the glass uniformly, and to seal the windows from leaking. However, it is possible to use different types of materials for the washers, but they should have the same or better properties at a temperature around 400°F (204°C). The type of teflon that was used (38) was poly-tetrafluoroethylene (PTFE) whose mechanical properties are listed in appendix C together with appropriate calculations. The following teflon properties data was used:

\[ E = 9,000 \text{ lb./sq. in. (6.2 } \times 10^7 \text{ N./sq. m.)} \text{ at } 400°F (204°C) \]
\[ \nu = 0.38 \]
\[ a = 1.51 \times 10^{-4} \text{ in./in.(F) (2.72 } \times 10^{-4} \text{ m./m.(C))} \]

f) Headers

Since there is no high pressure exerted on the headers, material other than stainless steel could be used. Brass is a suitable substitute since it is less expensive and easy to machine. An asbestos gasket (or similar material) was used as a seal between the die and the

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headers. See Figure 6 for details.

2.3 Basic design of some parts and accessories of the die

a) Supporting bolts and nuts considerations

According to design limitations, the length of the die was limited to 7.25 in. (18.4 cm.). The number of bolts which support the two halves of the die were limited to ten, 5/8 in. diameter bolts. This number allowed the use of a socket wrench to tighten and loosen the bolts. The bolt size was calculated in appendix A taking into considerations that the student who assembles the die can not apply more than 200 lb.-ft. (271 N.-m.) torque on the bolt using a manual torque wrench.

Since the mating or contacting surfaces of the die need to be under compressive stress during the time of operation to avoid the leakage, it was very important to apply a proper initial tensile preload $F_i$. This initial preload should be applied to the bolts during assembly by means of a torque wrench. During the extrusion process, the preload should be sufficient to prevent leakage due to internal pressure. The maximum design pressure inside the die should not exceed the value of pressure, which will be calculated
in the following chapters. That pressure will be distributed over the surface of the channel that has an effective area of 18 sq. in. (116 sq. cm.). The external force \( P \) on the bolted connection due to the pressure alone can be calculated by multiplying the pressure by the area and then dividing by ten number of bolts. The recommended initial preload which is around 19,200 lb. (89,000 N.) (47), can be calculated from the torque recommended above (see appendix A for details). Simply, by adding the magnitudes of both the applied preload \( F_i \), and the load portion \( P_b \) of the external load \( P \) taken by the bolt, the recommended total load \( F_b \) which is applied to the bolt would be calculated. Then the tensile stress in the bolt can be calculated by dividing the magnitude of \( F_b \) by the cross sectional area of the bolt. The calculated tensile stress in the bolt 79,700 lb./sq. in. \( (55 \times 10^7 \text{ N./sq. m.}) \) should not exceed the effective stress of the bolt given in article (b) section 2.2.

To assure that the two parts of the die will not separate, there should be a resultant compressive load \( F_m \) remaining between them after applying the internal pressure. The magnitude of the resultant load \( F_m \) can be calculated by subtracting the preload \( F_i \) from that portion of the external load taken by the parts \( P_m \). Therefore, the correct size of the bolts can be checked (see calculations in appendix A).
Ten bolts of 5.5 in. (14 cm.) in length, 5/8"-18-UNF-2A, and nuts of the size, 5/8"-18-UNF-2B, AISI 410 stainless steel were used. The die bolt holes should have a diameter of 21/64 in. = 0.65625 in. (1.667 cm.).

b) Heat transfer considerations and cooling system design

The die was divided into two regions according to temperature distribution. The first region starts at the die entrance and extends to the end of the shaping section where a constant temperature is maintained that does not exceed 400°F (204°C). To maintain this temperature, a specially designed heater was used to fit and surround those parts of the die that should be heated. The temperature is controlled by transducers and temperature control gages. The second region covers the whole land section at the end of the die including the cooling section. The temperature distribution in this region was assumed to drop linearly in the direction of the flow only to simplify the analysis. Besides, the amount of heat that should be removed from the polymer is so small that it does not exceed 1,428 Btu/hr. (100 Cal./sec.) (48). However, rough assumptions and calculations were sufficient to design the cooling system taking into consideration the symmetry of cooling and the distribution of tubes inside the die to circulate water or
any other coolant. Experimentally, it is possible to control the temperature distribution in the second region, by one of the transducers which is fixed at the beginning of the land section and connected to a temperature gage. At the same time the flow rate of the water or coolant passing through the cooling section can be adjusted. The temperature measured by the transducer at the second region should be lower than the melting temperature of the extruded polymer. Each half of the die has an independent cooling system but both are identical to each other. Figures 1-6 and appendix B give the location, dimension and calculations of the cooling system and temperature distribution along with the assumptions made. Therefore, temperature values at certain distances from the shaping section can be calculated and used in the analysis of the thermal stresses in the die which will be discussed in the following chapters.

c) Function and size design of the headers

Because the headers will not take any high pressure, they can simply be designed to satisfy the need for circulating the cooling water into and out of the die. A schematic diagram of one header is shown in Figure 6. Each header consists of two separated reservoirs. The closest header to the outlet of the polymer should be used as an inlet and the other as an outlet. The inlet reservoir
must be connected to the supply water and the outlet to the drain. The location of the two headers should be fixed at the end of the two split parts of the die as shown in the Figures and Photographs 1-6. Twelve screws of the size (1/8-13UNC-A) were needed to support each header to the die. An asbestos gasket was used to seal the contact surfaces.

2.4 The types of stress that are taken into consideration in design

There are different regions in the die that should be studied. However, the type of stress which should be taken into consideration must be chosen according to the problems existing in that region. Since the method used in this thesis is the finite element method, there is no choice in using stresses other than the stresses which are already given by the program. Otherwise, thousands of calculations would have to be done by hand, because the analysis can not be modified as can the commercial program ELAS. The ELAS program can provide the following output: the six components of stress $\sigma_x$, $\sigma_y$, $\sigma_z$, $\tau_{xy}$, $\tau_{yz}$ and $\tau_{zx}$ in global and local coordinates, the three stress invariants $I_1$, $I_2$, and $I_3$, and the effective stress ($S_e$) which is the octahedral shearing-stress multiplied by $(3/\sqrt{2})$. Therefore, the appropriate failure theory used in the die design was
the octahedral shearing-stress theory which is equivalent to the maximum distortion energy theory. In general, this theory was used to check for weak points in various sections of the die that might fail.

With regard to leakage the stress in a direction perpendicular to the leakage surface was of most interest. If the value of this stress is negative (compression), there would be no leakage and vice versa. Specifically, these stresses should be investigated in sections surrounding the glass windows and at the surface of contact between the two halves of the die.
CHAPTER III

Stress Analysis Investigation

Before starting with the description and assumption of each solution program, it is convenient to give a brief description of the ELAS program.

3.1a) ELAS program (34-36)

ELAS is a compact computer program for solving linear equilibrium, structural mechanics problems. This program (52) is useful as an instructional program for students because its storage is of limited size and therefore the running time is also limited. There are 18 different types of elements for use in one-, two- or three-dimensional structures. These elements have the following geometry: line segment, triangle, quadrilateral, tetrahedron, torus and conical segment. They can be used for the following structural types: planar and space truss or frame, gridwork frame, plane stress and strain, plate bending, general solid, general shell, solid of revolution and shell of revolution. In addition, the program can handle
concentrated loads, pressures, variations in the following: material, cross sectional areas, torional constants, moments of inertia, local-global angles, thicknesses, deflection boundary conditions, and temperature gradients. The output gives displacements, forces and stresses at each node. The program can relabel nodes to give the minimum bandwidth to reduce needed storage. The computed stresses at the mesh nodes are calculated from best-fit strain tensors.

3.1b) The checking program (37)

Another computer program was used to check the results obtained from the ELAS program. A brief description of this program is included. This program can solve only two types of structural problems namely the solid of revolution and the plane strain problem. The element types used are the two-dimensional triangular and rectangular elements. The output of the program gives the displacements at the nodes and the stress components at the centroid of each element. This program is used mainly to solve pile (soil mechanics) problems. There is a listing copy of the program available on tape in the Chemical Engineering Department (38).

3.2 Sections and programs used with their description and assumptions.

Since there are different problems in different regions of the die, one section is not enough to solve all of these
problems. Therefore, different sections of the two- and three-dimensions type were taken to solve the mechanical problems of the die. The first two problems of the die were concerned with leakage from the interface of the two split halves of the die, and the amount of pressure allowed inside the die. These problems were solved by three dimensional analyses. The second major problem concerned the breakage of the glass window since it is the weakest region in the die. This problem can be checked by studying section A-A of Figure 1 as a solid of revolution in one case, and as a plane strain or stress in the other taking into account a temperature increase of 400°F. The third problem was to check whether the area around the teflon washer was leaking or not under pressure and temperature by determining the stress. This also was solved by using the same section A-A but the effective stress was considered. The forth problem studied the effect of the temperature gradient and pressure across the length of the die at section C-C of Figure 1. The last problem was typical to all sections and cases, and involved checking all the spots for failure by the octahedral shearing-stress theory. This chapter covers the description and assumptions made for all critical sections. All results are shown by graphic contour plots in the next chapter.
a) Three-dimensional mesh section

A three-dimensional section of the die was selected that would cover one bolt hole, with a thickness equal to the distance between two bolts (1.4375 in.), and a length equal to half the width of the die (2.75 in.) as shown by the shaded area in the top view of Figure 1. Two identical meshes were chosen to solve the problem, but one with a bolt and the other without. The circular hole of the bolt was replaced by a square hole because a circular hole would need more nodes and elements which the ELAS computer program cannot handle. Figures 7 and 8 show the mesh for each program with a shaded area on which the pressures are distributed, and an arrow which shows the direction of that pressure. Both programs were solved by the ELAS computer program.

a.1) Three-dimensional mesh with bolt.

The mesh for this program, shown in Figure 7, has 104 nodes and 48 elements. Since, there is symmetry in the x-y plane passing through the center of the hole, the nodes on both sides of the plane have equal displacements in x, y and z directions. All nodes at distance 2.75 in. (center of the die) should have zero displacements in x-direction. This program has been used to check the design amount of pressure which should be applied inside the die to control the leakage throughout the contact surfaces. The first point that might leak would be around the bolts. Therefore,
Figure 7. The 3-Dimensional Mesh with Bolt.
all the nodes that are at the surface of contact \((y=0)\),
between the nearest edge of the hole \((x=1.0408\text{ in.})\) to the
reservoir and the edge \(x=0\) should be fixed in \(y\)-direction to
make sure that these points are under compression. If any
of these points has a tensile stress in the \(y\)-direction,
displacement at those points should be released to allow
free movement in that direction. However, any node that is
on the surface of contact which is not fixed in the \(y\)-
direction, and has a negative \(y\) displacement should be fixed
in that direction. This should be a realistic assumption
since it is impossible for the nodes at \(y=0\) to have negative
displacement in the \(y\)-direction due to the support it has
from the other half of the die. Therefore, there are a
total of 168 deflection boundary conditions.

From the calculations of appendix A, the preload \(F_i\)
can be distributed on the cross sectional area of the bolt
and used as a pressure boundary condition which is equal to
78,000 lb./sq.in. The pressure applied inside the reservoir
was assumed first as 5,000 lb./sq.in., then raised to 6,000
and finally to 7,000 lb./sq.in. to show the effect of
stresses on the surface of contact due to different pressure
intensities.
Figure 8. The 3-Dimensional Mesh without Bolt.
a.2) Three-dimensional mesh without bolt.

The mesh used in this analysis was identical to the previous program except that the nodes and elements of the bolt were missing as shown in Figure 8. The number of nodes were 92 and the number of elements 42. The assumptions for the deflection boundary conditions (DBC) were taken from the discussion of the previous program, but with 150 DBC because of the missing bolt. The pressure used on the bolt in the previous program can be replaced by an equivalent compression force distributed on the area under the head of the bolt, which is (29,270 lb./sq.in.). There is no change in the assumptions made in the previous program of the pressure inside the reservoir. The purpose of this program is to compare these results with the results of the previous program, and show that both programs give reasonable results.

b) Solid of revolution mesh

The geometry for the solid of revolution was chosen to represent section A-A of Figure 1, because the glass window and the structure around it are in a cylindrical shape. To study the effect of forces on the glass window, section A-A was divided at the center line of the glass window into two programs. Figures 9 and 10 show the finite element mesh for both programs, which are identical in the number of nodes (115 nodes) and also in the number of elements (88
Figure 9. The 2-Dimensional Mesh for the Small Diameter Solid of Revolution.
Figure 10. The 2-Dimensional Mesh for the Large Diameter Solid of Revolution.
elements). The major difference between the two programs is that the inside diameter for the first program equals the shortest distance from the center of the glass window to the reservoir side (0.75 in.), and the diameter of the other program equals the distance from the center of the glass window to the other side of the reservoir (1.75 in.).

The deflection boundary conditions in both programs were the same in that all nodes in the x-direction at the center of the glass window (x=0) were fixed, as were the nodes at the surface of contact (y=0) in the y-direction. The 6,000 lb./sq.in. pressure which was distributed on the surface of the reservoir and the lower part of the glass is shown as a shaded area. The pressure which was substituted for the bolt force was assumed to be equivalent to the bolt force divided by the bolt area, and that would be 9150 lb./sq.in. Two program runs were taken for each of the above defined programs. One of these runs used pressure only, and the other included the effect of temperature change to check the effect of both pressure and temperature on the structure and the sealing points. The ELAS computer program was used to solve all the above mentioned programs. However, the programs with pressure only also were solved by the alternate computer program too.
c) Two-dimensional mesh of section A-A

To be sure that the results from the solid of revolution programs were reasonable, a section A-A was taken into consideration with elements and nodes limited to 74 and 105 respectively. Nevertheless, both pressure conditions were identical to those for the solid of revolution problems. However, the deflection boundary conditions for this program were assumed differently, where the three nodes at the surface of contact and closest to the window were fixed, and only two nodes on the other side were fixed because the third node had a tensile stress under the applied pressures. Because the teflon washers caused the section to have a weak connection between the two steel sides of the section, additional deflection boundary conditions had to be taken into consideration. Four nodes were fixed in the x-direction, two on each side of the steel corners (nodes 40, 51, 61 and 63) of Figure 11.

Section A-A of Figure 1 was solved as a plane strain problem because of the conditions of loading on the die. This assumption did not work for the case with a temperature increase, because the assumption of zero strain in the z-direction causes very high stress components in that direction about (100,000 lb./sq.in.) which is unreasonable. On the other hand when the same temperature problem was solved as a plane stress problem reasonable stresses were
Figure 11. The 2-Dimensional Mesh for Section A-A (Fig. 1) as a Plane Strain and Plane Stress Problem.
obtained. Therefore it is adviseable to solve this problem both as a plane strain and as a plane stress case under the same pressure condition. Then, the results can be compared, to see if they correspond to each other. If they compare favorably then the problem can be solved completely as a plane stress case. This size of mesh can be handled by the ELAS computer program for both temperature and pressure, and it can be handled by the alternate computer program too, but with pressure only.

d) Two-dimensional fine mesh of section A-A

Figure 12 shows a finer mesh especially at the glass window region of section A-A with 224 nodes and 176 elements. Using exactly the same boundary conditions discussed in article (c) this program was checked for more accurate results. This program can not be solved by the ELAS computer program because of its size, but can be solved as a plane strain case by using the alternate computer program. One more note about this program which should be mentioned is that its mesh is the combination of the two solid of revolution programs discussed in article 3.2b.

e) Two-dimensional mesh of section C-C

In order to have a good idea about the stresses in the die due to the temperature gradient caused by the cooling system, a program was selected for section C-C in Figure 1.
Figure 12. The 2-Dimensional Fine Mesh for Section A-A (Fig.1) as Plane Strain Problem.
Figure 13. The 2-Dimensional Mesh for Section C-C (Fig.1) as Plane Strain & Plane Stress Problem.
The mesh is shown in Figure 13 with 126 nodes and 102 elements. A compressive pressure of 6,000 lb./sq.in. was distributed along the surface of the channel, and is shown by a shaded area. Section C-C was solved as a plane strain case first but the same problems mentioned in section A-A of article (c) were found. Therefore, the problem was solved as a plane strain case and then as a plane stress case, and the results were compared to see if they were reasonable. These cases were solved with pressure only by the ELAS computer program and the alternate computer program. All nodes at a distance \( y=2 \) in. should be fixed in the \( y \)-direction to compensate for the effect of the bolt supporting force. Other deflection boundary conditions were the fixing of the first four nodes at \( x=0 \) (nodes 1, 2, 3 and 4 of Figure 13) in the \( x \)-direction, due to the effect of the adapter.

When solving the problem due only to temperature change a plane stress case must be assumed in order to use the ELAS computer program. For the deflection boundary conditions, only five deflection boundary conditions were used consisting of the four nodes at \( x=0 \), which are fixed in the \( x \)-direction, and any node at \( y=2 \) that is fixed in the \( y \)-direction. Since the deflection boundary conditions used in either the case of pressure alone or the case of temperature alone were different, a combined program can not be solved
without superposing all node deflections and using them as deflection boundary conditions. However, the combined program needs 252 DBC in order to be solved by the ELAS computer program as a plane stress case.
CHAPTER IV

Results

This chapter presents the solutions to the problems discussed in chapter III. All the results are given in the form of contour line graphs, which represent the output of the computer programs taken from formulated computer data files. These data files were used to supply the computer programs with data for plotting stress contours as shown in this chapter. Each graph has a constant differential increment from one curve to an adjacent curve, as specified in one of the data files. The magnitude of the increment depends on the maximum and minimum values of stress, and the density of the curves.

Each one of the following articles covers the results corresponding to a specific mesh. All figures were outlined with a hidden line to show the location of teflon washers, glass window and stainless steel parts.

a) Three-dimensional section

In both three-dimensional programs described in the
previous chapter, four sectional planes were taken into consideration. Figures 7 and 8 show these sectional planes by numbers inside rectangles which are drawn in the direction of that plane. Figures 14 to 18 show the \( \sigma_y \) stress contour lines acting on plane (1) for the cases with and without the bolt, respectively. Figures 14, 16 and 18 show the results from Figure 7, while the results from Figure 8 are shown in Figures 15, 17 and 19. Figures 14-15, 16-17 and 18-19 are the results for polymer pressures of 5,000, 6,000 and 7,000 lb./sq. in., respectively.

Figures 20 and 22 show the \( \sigma_y \) stress contour lines acting on plane (2) for the cases with and without the bolt, respectively. Figures 21 and 23 present the effective stress contour lines for plane (3) for the cases with and without the bolt, respectively. The effective stress for plane (4) for the cases with and without the bolt are shown in Figures 24 and 25, respectively. The results of Figures 20 to 25 are chosen for a program run with a 6,000 lb./sq.in. pressure.

b) Solid of revolution section A-A

Two sizes for the solid of revolution problems were discussed in chapter III, therefore, two sets of results are presented, one for the smaller diameter (Figure 9), and the other for the larger diameter (Figure 10).
Figure 14. $\sigma_y$ Stress Contour Lines for Plane 1 of Figure 7 (5,000 lb./sq.in. Pressure)

Figure 15. $\sigma_y$ Stress Contour Lines for Plane 1 of Figure 8 (5,000 lb./sq.in. Pressure)
Figure 16. $\sigma_y$ Stress Contour Lines for Plane 1 of Figure 7 (6,000 lb./sq.in Pressure)

Figure 17. $\sigma_y$ Stress Contour Lines for Plane 1 of Figure 8 (6,000 lb./sq.in Pressure)
Figure 18. $\sigma_y$ Stress Contour Lines for Plane 1 of Figure 7 (7,000 lb./sq.in Pressure)

Figure 19. $\sigma_y$ Stress Contour Lines for Plane 1 of Figure 8 (7,000 lb./sq.in Pressure)
Figure 20. Effective Stress Contour Lines for Plane 2 of Figure 7.

Figure 21. Effective Stress Contour Lines for Plane 3 of Figure 7.
Figure 22. Effective Stress Contour Lines for Plane 2 of Figure 8.

Figure 23. Effective Stress Contour Lines for Plane 3 of Figure 8.
Figure 24. Effective Stress Contour Lines for plane4 of figure 7.

Figure 25. Effective Stress Contour Lines for plane4 of figure 8.
b.1) Small diameter solid of revolution

Figures 26 and 27 show the effective stress contours under pressure only given by the ELAS and the alternate computer programs, respectively. Figures 28 and 29 give the effective stress contours and the $\sigma_y$ stress contours, respectively, under pressure and temperature which were solved by the ELAS program.

b.2) Large diameter solid of revolution

Figures 30 and 31 show the effective stress contours under pressure only given by the ELAS and the alternate computer programs, respectively while Figures 32 and 33 give the effective stress contours and the $\sigma_y$ stress contours, respectively, under pressure and temperature which were solved by the ELAS program.

c) Two-dimentional plane of section A-A

The problem of Figure 11 was solved first as a plane strain case under pressure only. Figures 34 and 35 show the effective stress given by the ELAS and by the alternate computer programs respectively. However, the same problem also was solved as a plane stress case using the same conditions but by the ELAS computer program, in which the effective stress results are shown in Figure 36. The effective thermal stress solved as a plane stress case is shown in Figure 37. Figure 38 shows the effective stress
Figure 26. The Effective Stress Contour Lines for Figure 9 under Pressure Only (ELAS).
Figure 27. The Effective Stress Contour Lines for Figure 9 under Pressure Only (Checking).
Figure 30. The Effective Stress Contour Lines for Figure 10 under Pressure Only (ELAS).
Figure 34. The Effective Stress Contour Lines for Figure 11 as Plane Strain under Pressure Only (ELAS).
Figure 35. The Effective Stress Contour Lines for Figure 11 as Plane Strain under Pressure Only (Checking).
Figure 36. The Effective Stress Contour Lines for Figure 11 as Plane Stress under Pressure Only (ELAS).
Figure 37. The Effective Stress Contour Lines for Figure 11 as Plane Stress under Temp. Only (ELAS).
Figure 38. The Effective Stress Contour Lines for Figure 11 as Plane Stress under Pressure and Temp. (ELAS).
Figure 40. The Effective Stress Contour Lines for Figure 12 as Plane Strain under Pressure Only (Checking).
for the plane stress case with the combination of both temperature and pressure effects. Finally, Figure 39 shows the $q_y$ stress plotted from the output of the plane stress program under both temperature and pressure. Another result can be added to this article and that is the effective stress of Figure 12 which was solved by the checking computer program, and under pressure only as shown in Figure 40.

\textbf{d) Two-dimensional plane of section C-C}

The problem of Figure 13 was solved first as a plane strain case under pressure only. Figures 41 and 42 show the effective stresses given by the ELAS and the alternate computer programs, respectively. However, the same problem also was solved as a plane stress case using the same conditions but by the ELAS computer program. The effective stress is shown in Figure 43. The effective thermal stress for the plane stress case is shown in Figure 44. Figure 45 shows the effective stress for the plane stress case with the combination of both temperature and pressure effects.
Figure 41. The Effective Stress Contour Lines for Figure 13 as Plane Strain Under Pressure Only (ELAS).
Figure 42. The Effective Stress Contour Lines for Figure 13 as Plane Strain under Pressure Only (Checking).
Figure 43. The Effective Stress Contour Lines for Figure 13 as Plane Stress under Pressure Only (ELAS).
Figure 44. The Effective Stress Contour Lines for Figure 13 as Plane Stress under Temp. Only (ELAS).
Figure 45. The Effective Stress Contour Lines for Figure 13 as Plane Stress under Pressure and Temp. (ELAS).
CHAPTER V

Discussion and Conclusion

Discussion

It can be seen from the results of Figures 14 to 19 that the uniaxial stresses perpendicular to the surface of contact and around the hole of the bolt varies with the pressure applied inside the channel. The Figures also show that the assumed type of supporting load on the section does not affect the results farther away from loading which are almost identical to each other. A limiting pressure of 6,000 lb./sq.in. is recommended for this die, while the results show that at 5,000 lb./sq.in., the compression stress is high, at 7,000 lb./sq.in. the die leaks at the edge of the bolt hole. At 6,000 lb./sq.in. the results show a reasonable compressive stress (around 700 lb./sq.in. pressure). Therefore, the 6,000 lb./sq.in. pressure might be taken as the design pressure and it should not be exceeded in order to prevent the die from leaking. The die was operated by several graduate students including the
author and it was found that the die leaked in the region of the holes after exceeding the 6,000 lb/sq.in. This discovery supports the conclusions predicted by the finite element analysis. Nevertheless, Figures 20 and 22 show that the results on the surface of the three-dimensional section excluding the area under the bolt head are not reasonable which should be around zero because all the stresses perpendicular to a free surface should equal zero. The low number of nodes and elements in the y-direction might be the reason for the inaccuracy of the results. The results of a weak section that was considered (plane 4 Figures 7 and 8), are shown in Figures 24 and 25. These results show a similarity to each other. Another interesting section is (plane 3 Figures 7 and 8) where the results are shown in Figures 21 and 23. These results can be compared to the results of section A-A to be presented later. Figures 21 and 23 show identical results except for the top locations of each graph, the difference found being due to the different types of load distributions.

The results for the smaller diameter solid of revolution section shown in Figures 26 and 27 are almost equal although, each graph is a solution for a different computer program. The slight difference in the results is due to the output given by each program, since the ELAS computer program supplies the stresses at the nodes (115
nodes), while the alternate computer program supplies the stresses at the center of the elements (88 elements). Investigating Figure 28, the glass window and the region around it show that the stresses are higher compared to that of Figure 23 due to the differential thermal expansion between the glass window, teflon washers and the stainless steel for the case of pressure and temperature. Nevertheless, the results are within the limits of the design stresses. In Figure 29 the \( \sigma_y \) stresses are plotted to give an idea whether or not there is tensile stress around the teflon washers. According to the results, the two washers are under an average compressive stress of 3500 lb./sq.in. which guarantees sealing at that location in the die.

The results for the larger diameter solid of revolution under pressure for both computer programs, are shown in Figures 30 and 31 which show approximately equal results. The conclusions made for the small diameter section also are true for this section, except that from Figure 33 an average compressive stress of 3000 lb./sq.in. is exerted on the teflon washers.

The results for section A-A was obtained from an ELAS computer program solution for a plane strain problem. Comparable results were obtained by the alternate computer
program as shown in Figures 34, 35 and 40. Since more nodes and elements covered the glass window and the region around it in the mesh corresponding to Figure 12 than in case of Figure 11, the results should be more accurate than those shown in Figure 35. The results for the same body treated as a plane stress case are shown in Figure 36. It is clear that there is insignificant difference between their results. However, the results of plane stress show effective stress values higher than that for plane strain due to the compressive stress components in the z-direction.

A plane stress solution was used to include the effect of temperature change, and will be considered in this section. From Figure 37, the results of the temperature effect alone show that the effective stresses are low. Upon superposing results due to pressure one can see the resultant effective stresses in Figure 38 and that these are within design limits. Figure 39 gives the results for the y stresses and shows that parts of the teflon washers on the left side of the structure, are subjected to an average compressive stress of about 3000 lb./sq.in., while portions of the washer on the right side indicate a compressive stress of 2400 lb./sq.in.

If results from Figures 26 to 40 are compared to each other, a final decision regarding failure of teflon seal or
the structure can be checked. To make the right decision the results for the smaller diameter solid of revolution should be compared with the results for the left side of section (see Figures 26-29 and 34-40), and the results for the larger diameter solid of revolution (see figures 30-33 and 34-40) should be compared with the right side of the same section. In general, the results indicate that the teflon washers do not leak, and they have an average compressive stress of approximately 2800 lb./sq.in. The effective stresses over the glass window and on the right side of section A-A share the highest stresses with the effective stresses under the glass window and on the left side of section A-A. The reason for the unsymmetrical stress distribution can be related to the unsymmetrical structure since the right side of section A-A behaves as a cantilever due to its distance from the center of the glass window to the edge of the reservoir (1.75 in.), while the left side behaves as a rigid body (0.75 in.). The results also show that the teflon washers tend to isolate the glass window from the rest of the structure.

The results obtained from both computer programs for section C-C are almost identical as shown in Figures 41 and 42. The same problems for section A-A arise in this section when one attempts to include the temperature change, and again the same reasons can be used to explain the
conditions. The results for both solutions, for the plane strain and plane stress cases are shown in Figures 41 and 43, respectively. An insignificant difference in stress distribution of about 600 lb./sq.in. can be noticed. However, the results show that there is a stress concentration located around the shaping section which is low compared to the average stress across the section. The effective thermal stresses shown in Figure 44 indicate high stresses (around 2400 lb./sq.in.) in the land section due to the cooling system but the values tend to drop when going upstream along the reservoir section. Superposing the results for both temperature and pressure, as shown in Figure 45, one can detect a maximum effective stress of about 8100 lb./sq.in. in the land section.

When summing the forces for each output for the ELAS computer program, a magnitude of almost zero will result which shows that the stresses meet the equilibrium conditions.

Conclusions

The manner in which the bolt force is applied for the three dimensional program shows no effect on the results at a distance of approximately 2 in. (as used in this analysis) from the point of loading. Also the results show that a
maximum internal pressure of 6,000 lb./sq.in. should avoid leakage from the die. The maximum stress obtained from the various solutions is about 20,000 lb./sq.in. (compression) and occurs on the right top edge of the glass window in section A-A. Nevertheless, it is less than the design effective stress which is 26,700 lb./sq.in. The remaining sections of the glass window show relatively low effective stresses well within the design limits.

From the results of section C-C one can conclude that a study should be performed of the area around the shaping section when the die is subjected to a constant temperature along its length, while a study should be performed of the land section in the case of internal cooling.

A computer program of the size of ELAS is barely sufficient to solve for the stresses in a complex structure such as this die even after making simplifying assumptions for the boundary conditions. It would be extremely helpful to use a large size, three-dimensional finite element program for determining stresses in this type of structure.
CHAPTER VI

Recommendations

1) Using symmetrical structures would help a lot to simplify the solution and reduce the time and cost needed to solve the problem by the finite element method.

2) Strain gages might be used on the body of the bolts, and on the outer surface of the die to check the deflections, forces and stresses taken by the bolts and die structure.

3) The die might be used for higher temperatures if the present teflon washers were replaced by ones of similar specifications but higher stiffness. Calculations should be made to check the effect of the substitute material on the die due to increased temperatures.

4) Using computer graphic subroutines together with the analysis can help in more rapidly understanding the output and in decreasing the number of errors occurring in the
input. If the computer program has its own graphic subroutines, it is better to make them available rather than using other computer graphic programs. The main graphics the computer program should have is the following:
   a- Contour lines for the stresses.
   b- The original mesh.
   c- The mesh after deformation.

5) While assembling the die these points must be noticed:
   a- Clean the bolts and nuts before using them.
   b- Tighten slightly the window plugs after heating the die to the required temperature.
   c- Adjust the die to the adapter while tightening the screws and bolts.
   d- Center the glass window before tightening.
   e- Never untie the transducers and window plugs until the die is cool.
   f- When disassembling the die, the stainless steel AISI 410 nuts should be released slowly until they are loose due to the fine clearance between bolt and nut threads.

6) If future experiments on the die prove to be useless due to leakage from the die, a modification can be done on the die by reducing the size of the channel thus decreasing the force on the bolts and allowing higher internal pressure to be used.
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Calculations of the Bolted Connection

The size limitation of the die in width, thickness and length caused each bolt to be limited to 5/8 in. dia., and withstand high stresses pressure. To calculate the initial preload \( F_i \), this equation can be used (47):

\[
T = 0.2 F_i D
\]

where,
- \( T \) is the torque required to produce the preload = 200 lb.-ft.
- \( F_i \) is the preload on bolt.
- \( D \) is the nominal diameter = 5/8 in.

so,

\[
F_i = \frac{(200)(12)}{(0.2)(0.625)} = 19,200 \text{ lb.}
\]

The total external load \( P \) can be calculated after choosing the right design pressure. Then,

\[
P = Pr A
\]

where,
- \( Pr \) is the design pressure = 6,000 lb./sq. in.
- \( A \) is the total effective area of the channel = 18 sq.in

so,

\[
P = (6,000)(18)/10 \text{ bolts} = 10,800 \text{ lb.}
\]

To calculate the load portions which are taken by the bolt and by the die, these symbols should be defined;

- \( P_b \) is portion of \( P \) taken by the bolt.
- \( P_m \) is portion of \( P \) taken by the die.
- \( F_b \) is resultant load taken by the bolt.
- \( F_m \) is resultant load taken by the die.
- \( K_b \) is stiffness of the bolt.
- \( K_m \) is stiffness of the die.

Because the bolt and the die are of the same material, then; \( K_m = 8K_b \) (47). Upon substituting the magnitudes of \( P \) and \( F_i \) in these equations we get,

\[
F_b = P_b + F_i = \frac{(K_b)P}{K_b + K_m} + F_i = 20,400 \text{ lb.}
\]

and
Fm = \( \frac{(Km)P}{Kb+Km} \) -Fi = -9,600 lb.

To calculate the uniaxial tensile stress \( S \) in the bolt;

\[
S = \frac{Fb}{Ab}
\]

where, \( Ab \) is the tensile stress area (47) =0.256 sq. in.

so,

\[
S = \frac{20,400}{0.256} = 79,700 \text{ lb./sq.in.}
\]
APPENDIX B

Calculations of the temperature distribution across the die.

To simplify the heat transfer problem, some assumptions must be made. Using the figure above will help to explain the problem. The assumptions are:

1) The land section is divided into two regions having linear temperature drops. The first region is (a) to (b) and the second is (b) to (c).
2) There is a temperature gradient in the direction of flow from (a) to (c) only.
3) There is flow of heat from the polymer to the coolant (perpendicular to the polymer flow) in the region (b) to (c).
4) The coolant is water supplied at a temperature $T = 70^\circ F$.
5) The mass flow rate is assumed to be 1 gal/min. = 515 lb./hr.

The data that are used in calculations are:

- Temperature at the end of the shaping section is $T_a = 400^\circ F$.
- The rate of heat flow from the polymer $Q = 1,428$ Btu./hr.
- Specific heat of water (49) $C_p = 1$ Btu./lbm(F) @70°F
- Thermal conductivity of steel (50) $K_s = 25.867$ Btu/sq.ft. hr F @212°F
- Thermal conductivity of water (50) $K_w = 0.347$ Btu/sq.ft. hrF @65°F.
- Nusselt No. (49) $Nu = 4.36$
- the heat transfer coefficient \( H_v \) @ 65°F can be calculated from this equation (49);

\[
H_v = \frac{\text{Nu} \cdot K_w}{D} = \frac{(4.36)(0.347)(12)}{0.25} = 72.62 \text{ Btu./sq.ft.hr.F}
\]

- The cross sectional area at (b) is;

\[
A_b = \frac{(2.5)(2)}{(12)(12)} = 0.035 \text{ sq. ft.}
\]

- Overall coefficient of heat transfer \( U \);

\[
U = \frac{1}{1 + \frac{1}{1 + \frac{1}{H_v \cdot K_s}}} = \frac{1}{1 + \frac{1}{1 + \frac{1}{(1.5625)}}} = 53.18 \text{ Btu./ft.hr.F}
\]

Calculations,

\[ Q_1 = m \cdot C_p \cdot (T_2 - T_1) \text{ --------- (1)} \]
\[ 1,428 = (515)(1)(T_2 - 70) \]
\[ T_2 = 73°F \]

The average temperature at the center of the vertical tube \( T_{ce} \) is,

\[
T_{ce} = \frac{T_2 + T_3}{2} = \frac{73 + T_3}{2}
\]

\[ Q_2 = m \cdot C_p \cdot (T_3 - T_2) \text{ --------- (2)} \]
\[ Q_2 = (515)(1)(T_3 - 73) \text{ --------- (2a)} \]

but,

\[ Q_2 = U \cdot A_b \cdot (T_a - T_b) \text{ --------- (3)} \]
\[ Q_2 = (53.18)(0.035)(400 - \frac{T_3 + 73}{2}) \]
\[ Q_2 = (0.93)(727 - T_3) \text{ --------- (3a)} \]

solving the two equations 2a & 3a, \( T_3 \) can be calculated;

\[ T_3 = 74.2°F \]

so, the amount of heat transferred from point (a) to (b) can be found by substituting \( T_3 \) into equation 3a,
\[ Q_2 = 607 \text{ Btu./hr.} \]

Now the temperature distribution at certain distances \(X\) from point \((a)\) to \((b)\) can be found by substituting different values of \((X)\) in equation 4a,

\[
Q_2 = \frac{K_s}{X} Ab (T_a - T) \quad (4)
\]

\[
607 = \frac{(25.867)}{X} (12)(0.035)(400 - T)
\]

So, \( T = 400 - (55.87 \times X) \quad (4a) \)
APPENDIX C

Mechanical Properties Calculation of Teflon

To calculate the mechanical properties of teflon at 400°F (204°C) certain equations and methods may be followed.

1) The modulus of elasticity $E$

The modulus of elasticity of teflon nonlinearly decreases with increasing temperature. Using the two equations from Ref. (51) the magnitude of $E$ can be extrapolated.

$$E = 4210(%C) - 163,500 \quad @ 23°C$$
$$E = 1270(%C) - 44,500 \quad @ 100°C$$

where, %C is the percentage of crystallinity and is equal to 60% (53). Then after substituting in the equations above we get,

$$E = 89,100 \text{ lb./sq.in.} \quad @ 23°C$$
$$E = 31,700 \text{ lb./sq.in.} \quad @ 100°C$$

Plotting the log values of these two results, the magnitude of the $E$ @ 204°C can be found by extending the line as shown in the figure below. Then,

$$E = 9,000 \text{ lb./sq.in.} \quad @ 204°C (400°F)$$

2) Poisson's ratio $\nu$ (38).

Poisson's ratio for polymers is classified between the Poisson's ratio for solids and liquids depending on the crystallinity content. Therefore, using the following equation with the assumption that the liquid Poisson's ratio is around (0.5) and the solid's around (0.3) one finds
\[ \nu_{\text{ef}} = (\text{non-crystal content})(\nu_1 - \nu_s) + \nu_s \]
\[ = (0.4)(0.5-0.3)+0.3 = 0.38 \]

which is a reasonable value.

3) **Coefficient of thermal expansion** $\alpha$.

The average value of the coefficient of thermal expansion (51) for a temperature range from 25°C to 200°C is:

\[ \alpha = 151 \times 10^{-6} \text{ in./in.F (272 x 10^{-6} m./m.C).} \]
APPENDIX D

Computer Programs

FILE: IJD FORTRAN A-- OHIO UNIVERSITY COMPUTER SERVICES -- ATHENS, OHIO --

C...
C. BY I. A. ABUDEH, MECHANICAL ENGINEERING DEPARTMENT, OHIO UNIVERSITY--
C. THIS PROGRAM CAN BE USED TO PLOT 3-DIMENSION ISSUES ON Z-DISP.
C. SCREEN USING 'CALCOP SUBROUTINES'.
C...
DIMENSION X(500), Y(500), Z(500), IE(500, 4)
C...
READ NO. OF NODES (NWD), ELEMENTS (NEL), AND THE ANGLE (PSI) EQ. (O. TO
180) DEGREE BETWEEN Z-AXIS AND I-AXIS.
C...
READ (5, 1) NWD, NEL, PSI
1 FORMAT (2I5, PS. 4)
ZMAX=G.
XMAX=G.
T=0.
PSI=180.-PSI
C...
READ NODES DIMENSIONS.
C...
DO 10 I=1, NWD
READ (5, 2) N, X(M, I), Z(M, I)
2 FORMAT (2I5, E10. 4)
ZMAX=AMAX1(ZMAX, Z(I))
XMAX=AMAX1(XMAX, X(I))
ANC=COS(PSI*(3.141593/180.))
ANS=SNM(PSI*(3.141593/180.))
Y(I)=X(I)-Z(I)*ANC
Y(I)=Y(I)+Z(I)*ANS
IF(ZMAX.LE.T) GO TO 10
T=ZMAX
D=0.2-Z(I)*ANC
Y(I)=Y(I)+Z(I)*ANS
CONTINUE
10 CONTINUE
C...
READ ELEMENTS LABELING.
C...
DO 20 J=1, NEL
READ (5, 3) IE(I, J), I=1, 4
3 FORMAT (5I5)
CONTINUE
20 CALL PLOTS(6, 330, 7)
CALL FACTOR(1.5)
CALL SCALE(TX, 2., 8.5, 1.10.)
CALL SCALE(TY, 3., 6., 1.10.)
CALL PLOT(2.5, 1.5, 3)
CALL AXIS(I, 0.3, 0.1, 'I-AXIS (INS.)', -13.2, 0.0, 0.0, 10.0)
80 IF (PSI.GE.90.) GO TO 50
CALL AXIS(-2.0, 0.0, 'Z-AXIS (INS.)', 13.3, 0.90, 0.0, 1.10.)
CALL AXIS(XMAX, 2.0, 'Z-AXIS (INS.)', -13.5, PSI, 0.0, 1.10.)
GO TO 60
50 CALL AXIS(0.0, 0.0, 'Y-AXIS (INS.)', 13.2, 5.90, 0.0, 1.10.)
CALL AXIS(-0.2, 0.0, 'Z-AXIS (INS.)', 13.7, PSI, 0.0, 1.10.)
60 DO 70 I=1, NEL
DO 40 J=1, 4
J=IE(I, J)
XX(I)=X(J)
YY(I)=Y(J)
40 CONTINUE
XX(5)=XX(1)
YY(5)=YY(1)
XX(6)=0.
YY(6)=0.
XX(7)=1.
YY(7)=1.
CALL LINE(XX, YY, 5.1, 0.0)
70 CONTINUE
CALL PLOT(0.0, 0.999)
STOP
END
FILE: IBM
FORTRAN A
--- OHIO UNIVERSITY COMPUTER SERVICES - ATHENS, OHIO ---
C....
C....BY I. A. ABBUD, MECHANICAL ENGINEERING DEPARTMENT, OHIO UNIVERSITY.
C....THIS PROGRAM CAN BE USED TO PLOT TWO DIMENSION MESHES USING
C...."CALCOTE SUBROUTINES".
C....
DIMENSION X(500), Y(500), IE(500,4)
DIMENSION N(10), LL(500), XI(7), YY(7)
C....
READ NO. OF NODES (NND) AND ELEMENTS (WEL)
C....
READ(5, 1) NND, WEL
FORMAT(215)
C....
READ THE DIMENSIONS OF THE NODES.
C....
DO 20 I=1, NND
READ(5, 2) ND, I(ND), I(ND)
2 FORMAT (IS, 2E10.4)
10 CONTINUE
C....
READ ELEMENT LABELING
C....
DO 20 K=1, WEL
READ(5, 3) L, (IE(K, J), I=1, 4)
3 FORMAT (S15)
20 CONTINUE
CALL PLOTS(6, 832, 7)
CALL FACTOR(1.0)
CALL SCALE(XI, 3.5, 1, 10.)
CALL SCALE(YY, 3.5, 1, 10.)
CALL PLOT(1.0, 1.0, -3)
CALL AXIS(0.0, -0.1, 'X-AXIS (INS.)', -13, 3, 0.0, 1.0, 1.0)
CALL AXIS(-0.1, 0.0, 'Y-AXIS (INS.)', 13, 2.3, 90.0, 1.0, 1.0)
DO 5 J=1, WEL
L(L) = 0
5 CONTINUE
F=0,
DO 70 IM=1, WEL
X=IM,
Y=IM,
DO 30 I=1, 4
J=IE(IM, I)
XX(I) = XI(J)
YY(I) = YY(J)
U(I) = I
30 CONTINUE
IF (XX(I).LE.1.0 OR YY(I).LE.1.0) GO TO 50
DO 60 I=1, 4
N=I
60 CONTINUE
DO 7 F=1, N
IF (U(F).EQ.0 OR LL(F).GT.0) GO TO 60
7 CONTINUE
F=H(I)
CALL NUMBER(XX(I), .02, YY(I), .03, .04, F, 0., -1)
LL(W) = W
60 CONTINUE
50 CONTINUE
F=F*F+4.
IF (XX(I).LE.1.0 OR YY(I).LE.1.0) GO TO 40
CALL NUMBER(XX(I), .06, FF, 0., -1)
40 CONTINUE
XX(5) = XX(1)
YY(5) =YY(1)
XX(6) = 0.
YY(6) = 0.
II(7) = 1.
YY(7) = 1.
CALL LINE(XX, YY, 5, 1, 0, 0)
70 CONTINUE
CALL PLOT(0., 0., 999)
STOP
END