Finite Element Analysis of Hydraulic Actuator by using CAE tools

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Abstract: A hydraulic system is a fluid power system that is commonly used in industries due to its ability to sustain high pressure. So, here in this paper double acting hydraulic actuator is designed based on the force acting on it and the stroke length. Analysis of different parts of actuators is done by using CAE tools checking for the equivalent stresses and deformation. Two different materials are also used to check proper material to be used.

Keywords--- Hydraulic actuator, double acting, analysis

I. INTRODUCTION:
Hydraulic actuators are the end results of Pascal’s law. Hydraulic actuator is the device which converts hydraulic energy into mechanical energy it consists of cylinder that transforms the flow of pressurized fluid into a push or pull of piston rod. In double acting actuators the fluid pressure can be exerted from both sides. Hydraulic actuators are rugged and suited for high force applications.

II. METHDOLOGY:
The load of 21832 N is to be sustained by actuator but if fails suddenly, so taking into account sudden loading conditions it should be designed for 43664 N (21832 *2). So by using CATIA V5 models are made, and every component is analyzed separately using two different materials that are low carbon steel and E335 steel. Low carbon steel is used as it is lighter in weight with good yield strength, tensile strength, corrosion resistance, ductility whereas E335 steel is practically used in industries as a material. So here we find out which one is better.

III. DESIGN PROCEDURE

Let A be the full area of the piston and a be the cross sectional area of the piston rod. Since the design is a double acting double ended hydraulic cylinder, pressure is acts on both sides of the rod, hence the area which the pressure is acting on is given by (A-a). The force produced is given in the equation below.

The following assumptions were taken into the consideration of the design of the cylinder, piston, piston rod and seals in the hydraulic cylinder.

Working fluid is mineral oil
Available pressured Pa =200 bar = 200*10^5 Pa
Atmospheric pressure = 1.0135 *10^5 Pa
Stroke length= 1135 mm =1.135 m
Cylinder output force = 43664 N
Factor of safety =3
End fixing factor = K = 0.7
Properties of materials used

<table>
<thead>
<tr>
<th>Material</th>
<th>Low carbon steel</th>
<th>E355 Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ultimate tensile stress</td>
<td>430 MPa</td>
<td>540 - 620 MPa</td>
</tr>
<tr>
<td>Yield tensile stress</td>
<td>215 MPa</td>
<td>290 - 450 MPa</td>
</tr>
<tr>
<td>Young’ Modulus (E)</td>
<td>210 GPa</td>
<td>190 GPa</td>
</tr>
</tbody>
</table>

A. Design of piston rod
The rod is more likely to fail by buckling under the compressive load. In this case, the rod behaves like a column and is subjected to buckling. Therefore Euler’s formula in the equation below for long column can be used to obtain the piston rod diameter

\[
P = \frac{\pi^2 EI}{L^2 K^2}
\]

Where: 
- \( P \) = Buckling load (N)
- \( L \) = the column length (m)
- \( I \) = Moment of inertia (m^4)
- \( E \) = Young's Modulus of Elasticity for the column material (Pa)
- \( K \) = the end fixing factor =0.7
- \( E \) = Young’s modulus of the material used in this design calculation is 120 GPa

\[
P = \frac{\pi^2 (120*10^9)I}{(1.135^2*0.72)}
\]

\[
I = 39.894*10^{-9} m^4
\]

\[
d = 15 mm
\]

from Baym Hydraulics Corporation catalog of metric rod wipers and piston seals the nearest standard rod seal diameter is 20 mm.

B. Design of the piston

\[
A-a = \pi (D^2-d^2)/4
\]

\[
200*10^5 = (43664*384)/(\pi(D^2-0.02^2))
\]

\[
D = 0.0934 m = 93.48 mm
\]

from Baym Hydraulics Corporation catalog of metric rod wipers and piston seals, the nearest standard rod seal diameter is 100 mm.
2. Length of piston = D to 1.5D = 100mm

3. Thickness of piston head
\[ t = 0.43D \left( \frac{P}{\sigma} \right)^{1/2} \]
\[ = 0.43 \times 100 \times \sqrt{\left( \frac{200 \times 10^5}{55 \times 10^6} \right)}^{1/2} \]
\[ = 0.43 \times 100 \times \left( \frac{200}{55} \right)^{1/2} \]
\[ = 0.43 \times 100 \times \left( \frac{200}{55} \right)^{1/2} \]
\[ = 25.92 = 26 \text{mm} \]

4. Radial thickness
\[ t_r = \frac{D}{2} \left( \frac{3P}{\sigma} \right)^{1/2} \]
\[ = 0.1 \left( \frac{3 \times 0.03090}{(430/3)} \right)^{1/2} \]
\[ = 2.5 \text{mm} \]

**C. Design of the cylinder**

Let OD = outside diameter of the cylinder.

The maximum working stress (\( \sigma_m \)) is given as
\[ \sigma_m = \frac{\text{Tensile stress of material}}{\text{FOS}} \]
\[ = \frac{430}{3} \]
\[ = 143.3 \times 10^6 \text{Pa} \]

\[ \text{OD}^2 = D^2 \left( \frac{\sigma_m + P}{\sigma_m - P} \right) \]
\[ = 0.1 \left( \frac{143.3 \times 10^6 + 200 \times 10^5}{143.3 \times 10^6 - 200 \times 10^5} \right) \]

\[ \text{OD} = 115 \text{mm} \]

**D. Cylinder Tube thickness**

The wall thickness required for the cylinder can be calculated from the formula in equation
\[ t = (\text{OD} - d)/2 = (115 - 100)/2 = 7.5 \text{mm} \]

**E. Bursting stress**

The bursting stress can be referred to as the amounts of hoop stress and longitudinal (axial) stress that are produced in the wall of the cylinder when subjected to internal and external pressures that may cause the material which the cylinder is made from to fail. This happens if the hoop stress exceeds the tensile strength of the material.

The hoop stress (\( \sigma_H \)) of a cylinder can be determined from the Barlow formula as shown in the equation below.
\[
\sigma_H = \frac{P((d_o^2 + d_i^2))/ (d_o^2 - d_i^2))}{(d_o^2 - d_i^2)\}
\]

Where,
\[ P = \text{oil pressure, 200bar = 200 \times 10^5 \text{Pa}} \]
\[ d_o = \text{outer diameter of cylinder = 55mm} \]
\[ d_i = \text{inner diameter of cylinder,} \]
\[ = 200 \times 10^5 \times \left( \frac{(115^2 + 100^2)}{(115^2 + 100^2)} \right) \]
\[ = 144.031 \text{MPa} \]

Also the longitudinal stress is given by:
\[ \sigma_L = \frac{(P_1 R_1^2 - P_2 R_2^2)}{(R_1^2 - R_2^2)} \]

Where, \( P_1 = \text{Internal pressure (200 \times 10^5 \text{pa)}}, \)
\[ R_1 = \text{Internal radius} \]
\[ R_2 = \text{External radius} \]
\[ = \left( \frac{(200 \times 10^5 \times (50^2 + 3^2)) - (1.0135 \times 10^5 \times (57.5^2 + 10^2))}{(57.5^2 + 10^2) - (50^2 + 3^2)} \right) \]
\[ = 61.599 \times 10^6 \text{MPa} \]

**IV. RESULT AND DISCUSSION:**

**A. Low carbon steel:**

![Fig. 1 Stresses in piston rod](image1)

![Fig. 2 Deformation in piston rod](image2)

![Fig. 3 Stresses in cylinder](image3)

![Fig. 4 Deformation in cylinder](image4)
Table 2. Low Carbon Steel

<table>
<thead>
<tr>
<th></th>
<th>Rod</th>
<th>Cylinder</th>
<th>Piston</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stresses (MPa)</td>
<td>153.29</td>
<td>179.4</td>
<td>240.33</td>
</tr>
<tr>
<td>Total deformation (mm)</td>
<td>0.75084</td>
<td>0.034873</td>
<td>0.10226</td>
</tr>
</tbody>
</table>

B. E355 Steel:

Fig. 5 Stresses in piston

Fig. 6 Deformation in piston

Fig. 7 Stresses in piston rod

Fig. 8 Deformation in piston rod

Fig. 9 Stresses in cylinder

Fig. 10 Deformation in cylinder

Fig. 11 Stresses in piston
V. CONCLUSION

Here, design and analysis of different parts of actuator i.e., piston rod, piston, cylinder is done using CATIA V5 and ANSYS. From the above obtained value, we can conclude that the stresses developed in low carbon steel are more than E355 steel and the total deformation of E355 is more than low carbon steel. Hence from obtained data it is beneficial to use E355 steel.

VI. REFERENCES