Research Paper Engineering

Design of 500 mm Bore Pneumatic Double Acting Linear Actuator with Rear End Cover Integrated Rear Trunnion Mounting

A.S. Jadhav	M.Tech Scholar B.V.D.U.C.O.E. Pune
V. J. Shinde	Associate. Prof. B.V.D.U.C.O.E. Pune.
V. C. Bakshi	Deputy Manager- Design, Schrader Duncan Ltd, Pune.

ABSTRACT

In pneumatic fluid power systems for automation of engineering control systems, the actuator is one of the important components. The pneumatic actuator converts the pressurized air energy into mechanical work i.e. thrust and motion. The paper details the design of 500 mm bore pneumatic double actuator at 10 bar compressed air pressure. The rear trunnion mounting for the actuator is integrated in the rear end cover, resulting in reduction of number of parts and weight reduction. The material selection and calculations required for the sizing of components is done. The modelling and assembly of the components is done in software Solidworks 15. The FEA static structural analysis of the major components is done in software ANSYS 14. The results are discussed for the safe design.

KEYWORDS pneumatic actuator, cover integrated mounting, static structural analysis.

INTRODUCTION

The pneumatic actuators are the devices that use compressed air to do the useful mechanical work. The pressure vessels are used to generate high pressure inside the actuator than ambient pressure, creating a pressure differential. The air pressure is set by pressure regulators and supply or control of air is done by direction control valves. The thrust, generated as the output from the actuators, is utilized in various machines and mechanisms that perform the desired work. The pneumatic actuator's selection is to be done considering various requirements of the application. Pneumatic actuators are used with connection-options like forks, rod eye end bearings, and mountings with multiple choice.

The review of contemporary context of pneumatic and other control systems like hydraulic, electromechanical is done by Hazem et.al. The application based performance benefits like lower specific weight of actuators, higher working temperature and nuclear environment, higher power rate, reduction in storage and return lines for the fluid i.e. air being clean source and can be exhausted in air. Also, the review of suitability of pneumatics for better control system and a wide application in pneumatics in industry is done. The basic mathematical model of pneumatic actuator is done considering the three basic inputs, thermodynamics for mass flow, fluid dynamics for air pressure, temperature and volume and dynamics of motion for determination of dynamic load [1].

The methodology of design for an actuator used in the application of special purpose CNC machine hydraulic clamping system to automate the clamping of workpiece, which is earlier manually clamped by machine operator is also studied by KeYanga et.al. The design parameters like basic working pressure on plunger, inner diameter of actuator, cylinder wall thickness of the actuator, plunger bar stress intensity, etc. are observed. [2]

The industrial product development for valve actuator of organization Regin AB is thoroughly reviewed with process of industrial product development from marketing stage to final product validation. The different commercial as well as technical aspects are studied. Jenny Anderson studied the feature based comparative analysis from the perspective of an industrial designer. [3]

The pressure vessel design is always concerned with the practical environmental conditions, as even if the extreme conditions, above design parameters, last for a minor period, may lead to failure, the design methodology of the pressure vessel as per ASME codes is described by B.S. Thakar et.al. The classification of pressure vessels and different norms of the pressure vessel and benefits of using this standards are understood. [4]

For the laminated composite material hydraulic actuator static condition i.e. applying pressure to inner surface of the cylinder, when the piston arrives at the end if stroke and cylinder is filled with fluid, the analysis of stress and deflections in the co-ordinate axes is showed and modal analysis is done to obtain natural frequency and vibration shapes are obtained by applying the boundary conditions.[5]

A pneumatic actuator basically consists of a cylindrical tube or barrel, and the ends covers of this barrel holds the air pressure inside and are given air supply connections in form of ports, the piston inside the barrel moves to and fro with respect to air supply from the ports in either directions. Piston is guided and given the desired stroke with piston rod. The sealing system is important criteria for actuators as the output of the actuator is dependent on it.

METHODOLOGY

The optimized design of 500 mm actuator with the integration of rear trunnion mounting in the rear end cover is done at 10 bar maximum pressure, starts with considerations of operating environmental conditions and material selection. The calculations for sizing the components is to be considered with safety margin, for durable design and avoiding any sort of failure mode like internal cracking of barrel or braking of end covers, or buckling of piston rod, etc.

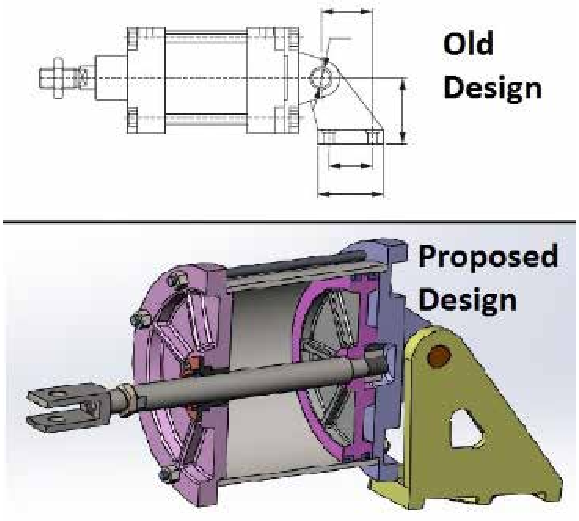

Figure 1: Mounting Integration Design Concept for Pneumatic Linear actuator

After the study of requirement and working conditions for this actuators, as shown in Fig- 1, the decision for integrating the rear trunnion mounting into rear end cover is taken. The integration process will result in reduction number of parts consequently resulting in savings of material cost and assembly time. The CAD models of actuator components and the assembly of the actuators are created using software Solidworks 2015. The static structural analysis of major components of the actuators is done using the analysis software ANSYS 14, by applying the required boundary conditions. The results for stress and deformations obtained for various components are discussed. Manufacturing of the actuator is done according to the manufacturing drawing. The assembly of the actuator is tested for leakage at maximum compressed air pressure of 10 bar and validation of the design for performance and dimensions is done.

MATERIAL SELECTION

The 500 mm bore actuator is a large bore and heavy duty actuator. Hence, as per pneumatic industrystandards, the material selection is done. The tube or barrel is made up of mild steel as it offers suitable tensile strength and can be easily machined. The end covers, piston, rear trunnion mounting and hinge bracket is made up of grey cast iron FG 260 due to the benefits it offers like easily shaped with low tooling cost, good vibration damping ability, and steady mechanical properties for higher operating temperature of actuators. Piston rod is made up of hard chrome plated steel EN8 as it corrosion resistant, tie rods are made up of stainless steel. The seals are made up of Nitrile butadiene rubber, it is best suited for the application working temperature requirements.

THEORETICAL CALCULATIONS

The cylinder tube or barrel is the pressure vessel, the design of the tube is done based on ASME codes, [6].

The total calculated thickness of tube,

$t_{T} = t + CA$

where, CA = Corrosion allowance, (mm)

t = minimum thickness required for the tube, (mm), t = [(di * Pd)/(2*Sa*Un-1.2Pd)],

Pd = design pressure, (MPa)

Un = Joint efficiency of tube (1)

Sa = Allowable stress for material (Mpa)

The value of calculated t_T is 5. 21 mm but the market available tube of 10 mm thickness is used. The end covers of the actuator are designed according to the case of flat plates with respect to the uniformly distributed pressure over q from r_0 to a. [7],Where, H = The value of minimum thickness of plate (mm), q = uniformly distributed pressure (MPa), r_0 = radius of start of pressure distribution (mm), a = radius of outer support provided to end cover (mm).

The calculated value of end covers minimum thickness is 30 mm, but considering the air inlet port size of 25 mm, thickness of the end covers is taken as 50 mm.

The piston of actuator is designed with respect to guided and fixed of annular plate with uniformly distributed pressure q over the portion from r_0 to a. [7]

Where,

H = Value of minimum thickness of plate (mm)
q = Uniformly distributed pressure (MPa)
r_0 = Radius of start of pressure distribution (mm)
a = Outer radius of piston (mm)

The minimum thickness of piston required is 55 mm, considering the grooves for seals, the piston thickness is taken as 77 mm.

The piston rod is designed for minimum area of rod and buckling considerations for extended stroke of actuator, [8]

The minimum piston rod diameter is found using Euler's equation for critical load on buckling of column given by

$$Pcr = \{[(n * \pi^2 * E * A)] / [(l/k)^2]\}$$

The design of tie rods and hinge mounting is done on the basis of threaded joint, [8].

MODELLING AND ASSEMBLY OF ACTUATOR FOR COMPONENTS

The models are made after the calculations and selection of size and dimensions for the components are finalized. The modelling of the actuator components is done in Software Solidworks 2015. The components of the actuator includes tube, front end covers, rear end cover integrated with rear trunnion mounting, piston, piston rod, gland, tie rods, etc.

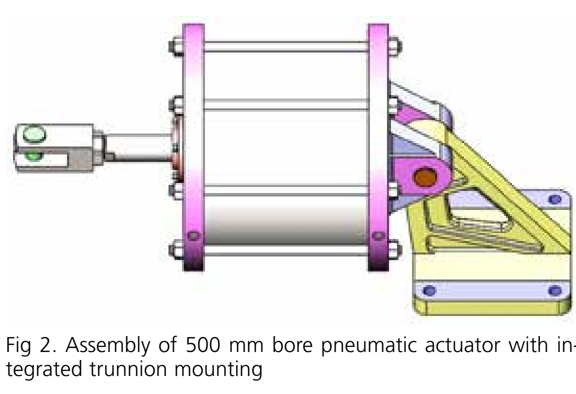

Fig 2. Assembly of 500 mm bore pneumatic actuator with integrated trunnion mounting

The fig 2 shows the 500 mm bore actuator assembly with integrated mounting and fork accessories.

STATIC STRUCTURAL ANALYSIS OF THE MAJOR COMPONENTS OF THE ACTUATOR

The static structural analysis of the major individual components is done applying boundary conditions and the load to analyze the resultant stress and deformation in the component, after applying the load. The static structural analysis of the component is done in software ANSYS 14.

 ISSN - 2250-1991

TUBE

Figure

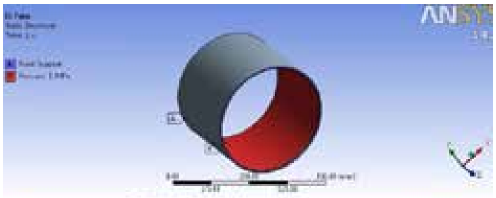

Figure 3 a - Tube boundary conditions

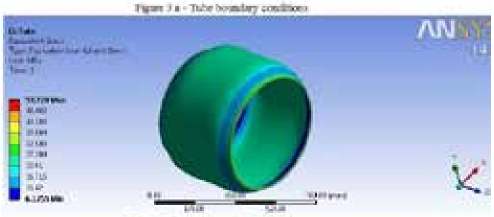

Figure 3 b - Tube equivalent stress

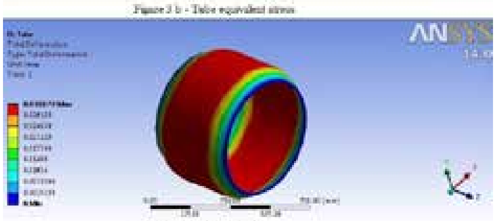

Figure 3 c - Tube deformation

3: Structural analysis of tube

Fig. 3 a shows that tube is fixed at the ends on the end covers, so, the edges of tube are restricted by the fixed support and pressure of 10 bar i.e. 1 MPa is applied to inner surface of the tube. Fig 3 b shows the equivalent stresses generated in the tube. For the mild steel material tube, the equivalent (Von-mises) stress generated is in the range of 6.125 to 53.778 MPa, which less than the allowable stress of 73 MPa and the maximum stress is at the edge of tube which is fixed. According to fig. 3 c, the deformation of the tube is zero at the fixed ends and gradually rises to maximum deformation of 0.031679 mm.

FRONT COVER

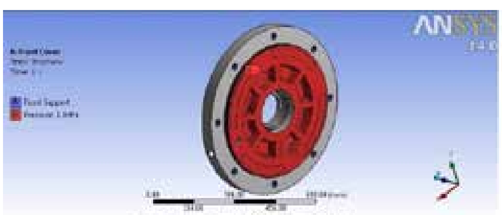

Figure 4 a - Front cover boundary conditions

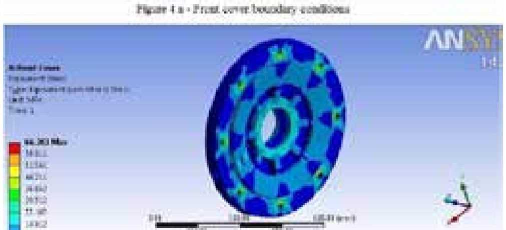

Figure 4 b - Front cover equivalent stress

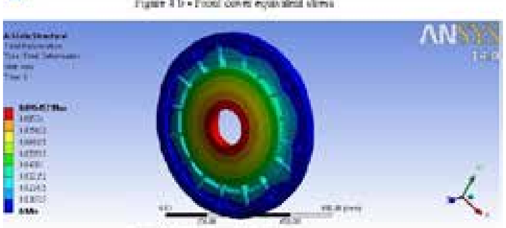

Figure 4 c - Front cover deformation

Figure –

4:Structural analysis of front cover

According to fig. 4 – a, the front end cover is fixed with rear end cover using tie rods through the 8 numbers of holes and the fix support for the holes is given and uniformly distributed pressure of 1 MPa is applied on the affecting surface. From fig-4 b, equivalent stresses (Von-mises) on the end cover are observed, which are within the allowable stress 91 MPa and maximum stress concentration 66.261 MPa is near the holes for tie rod mounting. Fig 4 c shows the total deformation of the front end cover. The maximum deformation of 0.0964 mm is observed at the inner edge of the cover.

REAR COVER INTEGRATED TRUNNION MOUNTING

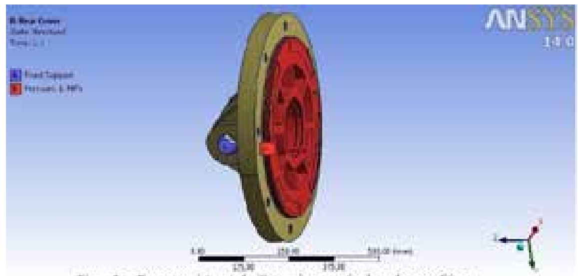

Figure 5 a - Rear cover integrated with trunnion mounting boundary conditions

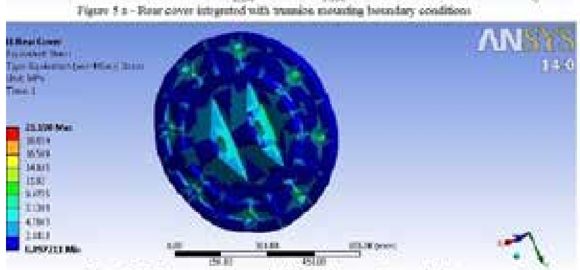

Figure 5 b- Rear cover integeted with trunnion mounting equivalent stress

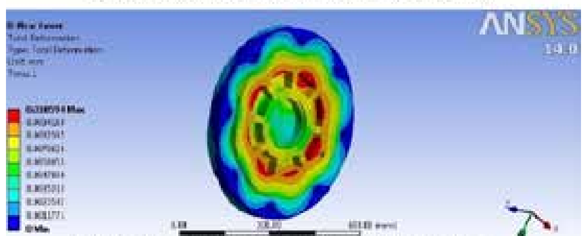

Figure 5 c - Rear cover integrated with rear trunnion mounting deformation

Figure 5: Structural analysis of rear cover integrated trunnion mounting

The boundary condition for rear cover integrated with rear trunnion mounting is shown in fig 5 a, similar to front end cover, the tie rod holes are fixed, additionally, the surfaces of holes for hinge pin are restricted in x direction. Fig 5 b shows the equivalent (Von-mises) stress concentration in the rear end cover integrated with rear trunnion mounting. The maximum 21.198 MPa stress concentration is observed at edges of hole and the stresses are within the allowable stress of 91 MPa and the total deformation in rear cover with integrated trunnion mounting is shown in fig 5 c, the maximum deformation of 0.01059 mm is observed across the rib area.

HINGE MOUNTING BRACKET

Figure 6 a - Hinge mounting bracket boundary conditions

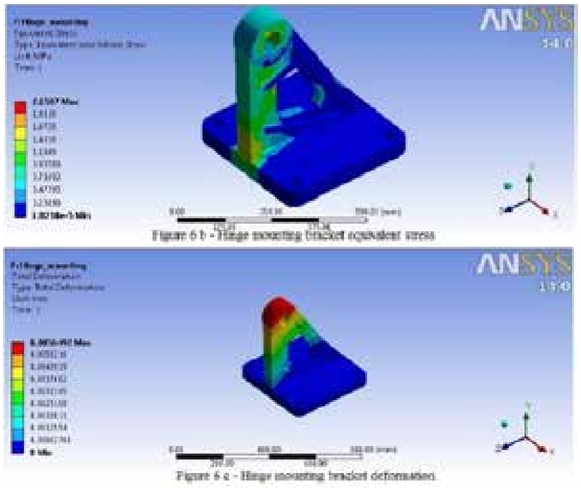

Figure 6: Structural analysis of hinge mounting bracket

Fig 6 a, shows hinge mounting bracket is fixed on a platform, and the mounting holes and surface is fixed and thrust of 196375 N is applied over the hinge pin area. The equivalent (Von-mises) stress in hinge mounting bracket is shown in fig 6 b, maximum stress of 142 MPa is observed near the bracket rib slot and it is within the allowable stress of 153 MPa. The total deformation of the hinge mounting bracket is observed as shown in fig 6 c, at the base support of the bracket, no deformation is seen and a very minute i.e. 0.6839 mm deformation is observed across hinge pin mounting area.'

RESULTS

Results of Structural analysis are shown in the table.

Table: 1 Results of structural analysis

Component	Observed maximum equivalent stress (MPa)	Allowable equivalent stress of material (MPa)	Observed Deformation (mm)
Tube	53	73	0.03167
Front End Cover	66.2610	91	0.09645
Rear end cover with integrated trunnion mounting	21.1980	91	0.0105
Hinge mounting bracket	142	152	0.06839

CONCLUSION

According to the results obtained from the structural analysis of the components, the Von miss stress generated within the components are less than the allowable stress limit from the materials and the maximum observed values of Von-mises stress are at fixed support, and near change in cross-sectional area of the component.

The deformation values are zero at the fixed supports and the negligible deformation values is observed in the components are in the range of few microns.

Hence, it is concluded that the design is safe.

REFERENCES

[1] Hazem I. Ali, SamsulBahari B Mohd Noor, S. M. | Bashi, M. H. Marhaban. "A Review of Pneumatic | Actuators (Modeling and Control)", 2009, | Australian Journal of Basic and Applied Sciences, | 3(2): 440-454, | [2] KeYanga, Shangjun Guan, Cunlong Wang. "The | design & calculation for hydraulic cylinder of | workpiece hydraulic clamping system of a special | CNC machine for guide disc." 2011. Procedia | Engineering 16 (2011) 418 – 422. | [3] Jenny Andersson , "Development of a valve | actuator for Regin AB." 2010. Master's Thesis in | the Master's program Industrial Design | Engineering, CHALMERS University Of | Technology Göteborg, Sweden. | [4] B.S.Thakkar, S.A.Thakkar, "DESIGN OF | PRESSURE VESSEL USING ASME CODE, | SECTION VIII, DIVISION." 2012. International | Journal of Advanced Engineering Research and | Studies. IJAERS/Vol. I/ Issue II/January-March, | 2012/228-234. | [5]N.Upendra, P.Moulali, K.Ajay Kumar Reddy. | International Journal of Engineering Research | (online),2347-5013(print), Volume No.3 Issue No: | Special 1, pp: 140-143, 2nd March 2014. | [6]ASME Code Section VIII, Division I. | [7] Warren C. Young Richard G. Budynas, Roark's | formulae for stress and strain. | | | | | [8] V. B .Bhandari, Design of Machine Elements. |