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Design and Testing of Hydraulic Actuator in Typical Areospace Vehicle

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Abstract: A hydraulic actuator is a mechanical actuator that is used to impart a unidirectional force through a unidirectional stroke. The detailed study of various control actuations and design of typical hydraulic actuation system is need as it plays a major role in the working of aerospace vehicle. This study involves the design of hydraulic actuation system for the flex nozzle of a solid motor as part of thrust vector control actuation system. This actuation system is going to control the vehicle in pitch and yaw direction during powered phase. The design, fabrication and testing of the hydraulic actuator are considered in this work. After designing the hydraulic actuator according to the specifications, testing is done manually, to check the performance of the vehicle by using different techniques. The actuator which is designed, fabricated and tested successfully is ready to be fitted in the aerospace vehicle.
Keywords: Hydraulic Actuator, Aerospace Vehicle, Design, Piston, Eye end, Testing.

I. INTRODUCTION

Aircraft actuators are devices that can transmit and redirect one form of motion energy to another. The various types of transmitting elements are employed to drive the mechanical links to a desired orientation. They are Hydraulic, Pneumatic, Electrical AC and DC Motors. Hydraulic system uses ‘oil’ under pressure and use electro-hydraulic valve. Hydraulic actuators are used in converting the energy of the working fluid into mechanical energy related to the reciprocating motion. The pressure of the working fluid acts on the piston and creates a force causing the piston assembly to move. As a result, the piston rod can perform useful work. Hydraulic actuators are executive elements in power hydraulic systems. These structures have several advantages, which include the possibility of obtaining large working forces and low operating speeds. Hydraulic power may be reconverted to mechanical power by means of the aircraft.

A. *Hydraulic Actuator*

An actuator where in hydraulic energy is used to impart motion is called hydraulic actuator. Actuator produces physical changes such as linear and angular displacements. They also modulate the rate and power associated with these changes. A fluid power hydraulic cylinder is a linear actuator which is most useful and effective in converting fluid energy to an output force in linear direction for performing desired work. Hydraulic cylinders are broadly classified according to function performed and construction.

B. *Main Components of Actuator*

1) Actuator consists of a servo jack and a servo valve. Servo valve gives the required flow to move the linear actuator at specified velocity. Jack consists of piston& cylinder and a linear potentiometer to give position feedback. Servo valve (flow control valve) plays a major role in electro-hydraulic actuation system. It consists of a torque motor stage and a hydraulic amplifier stage. Output flow rate is proportional to the input current.
2) LVDT: An inductive sensor (LVDT) is integrated in the hydraulic cylinder. It provides information about the exact stroke and supplies the signal to the machine control to regulate the oscillation stroke.

Fig.1: Hydraulic actuator

Fig. 2: LVDT

II. WORKING OF HYDRAULIC ACTUATION SYSTEM

A typical Aerospace vehicle hydraulic actuation system consists of thrust vector control (TVC) system and aerodynamic control (ADC) system. TVC system consists of Boot strap hydraulic reservoir, a pressure relief valve, a non-return valve, a pump motor package, 4 actuators, 11 hose assemblies (both pressure and return) stainless steel pipe assemblies, hydraulic connectors and TVC linkage systems. ADC system consists of 4 actuators, 10 hose assemblies, an accumulator, 2 QC/DC nipples, SS pipe assemblies, ADC linkage system, and a charging valve.

Hydraulic actuators achieve the gimballing of the engine in TVC and movement of the control surface in ADC. Each phase of the control scheme i.e., TVC and ADC is provided with a set of actuators. In the TVC phase, each engine is mounted with two actuators in mutually perpendicular directions (planes) for pitch and yaw. In ADC phase, each of the four control surfaces is connected with an actuator. Total number of actuators in the vehicle is eight. The oil is stored in a bootstrap hydraulic reservoir, which supplies oil to the suction of the pump. The reservoir is self-pressurized piston type. It takes system pressure to develop a suction pressure. A variable delivery axial piston type pump driven by a DC compound motor, pumps the oil from reservoir at a rated system pressure. The electrical supply to the motor is taken from a battery. High-pressure fluid is pumped through a nonreturn valve and a high-pressure filter. A high-pressure filter relief valve is provided in the pressure line. The outlet of the relief valve is connected to the return line. Two accumulators are connected in line, up-stream to the actuators.

Accumulators supply demand flow over the pump flow and dampen the pressure surges. The accumulators are charged with ultra-pure nitrogen. From each accumulator the supply is distributed to four actuators. Electro-hydraulic actuators convert fluid pressure into motion in response to a signal. Return oil from all the eight actuators are passed through two low-pressure filters and fed-back to the reservoir. Two quick connect couplings are provided in the pressure and return line for interfacing external rig.

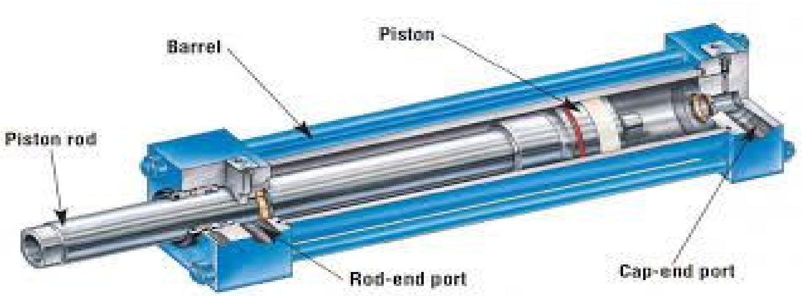

Fig 3: Schematic picture of hydraulic actuator

III. DESIGN OF ACTUATOR COMPONENTS

A. Design Calculations of Actuator Components

15-5 precipitation hardened stainless steel material is selected for the fabrication of components. Its main advantage is its high strength with good machinability. It is high corrosive resistant material with high hardness.

15-5 PH stainless Steel properties

Density	: 7.8 g/cc
Tensile strength, ultimate	: 1438Mpa
Tensile strength, yield	: 1385 Mpa
Elongation at break	: 9.4%
Thermal conductivity	: 17.9W/m-K
Hardness, Rockwell c	: 46

Inputs:

Fluid pressure	: 210bar or 2.1 kgf/mm^2
Proof test pressure	: 3.15 kgf/mm^2
Peak load on the actuator	: 5500 Kgf
Material of the components	: 15-5PH steel
Yield strength of the material	: 100 kgf/mm^2

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B. Design of the Piston

The piston is the moving part in the actuator which will move the load. An LVDT is housed inside the piston to get the position feedback. So, the inside the piston is taken as 20 mm to accommodate LVDT. The probable failure of the piston and piston rod during operation can be due to

1) Tensile and compressive stresses induced in the piston rod due to fluid pressure.
2) Shear stress induced at piston.
3) External pressure acting on the piston rod.
4) Buckling.

And the probable failure of the piston and piston rod during proof test can be due to

a) External pressure acting on the piston rod.

Direct stresses induced in the piston rod:

ID of the piston rod	: 20 mm
OD of the piston rod	: 28 mm
Cross sectional Area of the piston rod	: ∏/4(D^2-20^2)
Load on the piston rod	: 10000/ [∏/4 (1-20)]
Allowable stress of the material	:100 kgf/ mm²

By equating safe stress to the actual stress, we can get the diameter D

σ =p/A 100=10000[∏/4(D^2-20^2)]

where, D=21.87 m.

So, by considering a minimum thickness of 4 mm & nearest seal available the OD of piston rod is taken as 28 m

The piston is subjected to shear load. The thickness of the flange can be found as follows. The allowable shear stress of the material is 50 kgf/mm²

Shear load on the flange: 5500Kgf

Arca: ∏Dt

Shear stress: 10000/28

So, by equating safe stress to the actual stress, we can get the thickness of piston, t.

50=10000/x28xt

∴ t = 2.27 mm

But the piston has to be provided with appropriate sealing. Therefore, a width of 16 mm is selected for the piston to accommodate a composite seal. The gland dimensions are selected based on seal chosen and as per MIL-G-5514.

C. Design of Body

The body is shown in the fig. Though it is having some extra material for mounting of the servo valve it is considered as a cylinder for the analysis purpose. The inside diameter of the body is chosen as follows. The working pressure is selected as 210bar. The load pressure is taken as 200bar assuming that about 10bar is back pressure.

The maximum load: 10000Kgf

Back pressure of the fluid: 2Kgf/mm²

Area, A: 10000/2

: 5000mm²

Effective area: $\pi/4(d^2 - d1^2)$

Where, d is inside diameter of the body and d, is the outer diameter of the piston rod

d1=28mm

Equating the effective area to area A we can get the 11) of the body

d=84.57mm

The ID is selected as 84.57 as per MIL-C-5514 to accommodate the composite seal selected. Now we can find the OD by considering it as a pressure vessel subjected to internal pressure. As the proof pressure of 3.15Kgf/mm' is applied to cylinder calculation is done using proof pressure.

Thickness t= (p.d) (2f1)

=3.15 84.57/2 100

=1.3319mm

A minimum thickness of 5mm is considered from machinability point of view and the OD is taken as 94.57mm.

Now the Body is a cylinder with a wall thickness of 5mm and inner dia. of 84.57mm.

t/D=5/94.57 =0.05287

t/D< 0.07

The body should be considered as a thin pressure vessel.

Internal pressure:3.15Kgf/mm²

ID of the Body: 84.57mm

OD of the body: 94.57mm

The hoop stress induced in the body is fh= PD/2t

Longitudinal stress induced is fi = PD/4t

And radial stress ft = P

Where P is internal pressure &D is ID of the body

fh = (3.15x84.57) (2x5)

=26.639 Kgf/mm

Fi = (3.15x84.57)/(4x5)

= 13.3197 Kg/mm²

&fr = 3.15Kgf/mm²

According to von mises stress theory the equivalent stress is 28.96 Kg/mm

So, the factor of safety is

100/28.96 = 3.453

D. Design of Eye End

This Eye end is making two functions.

1. Acting as a head for hydraulic cylinder.
2. Taking the direct load.

With the case (1) we can find the thickness of the flange with empirical formulae.

T =d(K.P/ft)

D =PCD of bolts

=95mm

P =Pressure

= 3.15Kgf/mm²

Ft =100Kg/mm²

And for this particular mounting position

K=0.162 (bolted joint.)

∴ tc = 95√ (0.162×3.15/100) =6.8 mm

The thickness of the flange is selected as 8mm.

& Stress induced in the flange is 71.9Kgf/mm²

Factor of safety is 1.39

∴ It is safe.

Direct load of 10000 kg is acting on the cover in return stroke. The ID of the extended portion is selected as 28mm to facilitate the free movement of the piston rod, and the OD can be found as follows.

Cross sectional area of the Eye end = π /4(D-28)

Direct stress acting on the Eye end =10000/ π /4(D-28')

Allowable Stress of the material = 100Kgf/mm³

∴100 =10000/x/4(D-28)

∴D =30.189 mm

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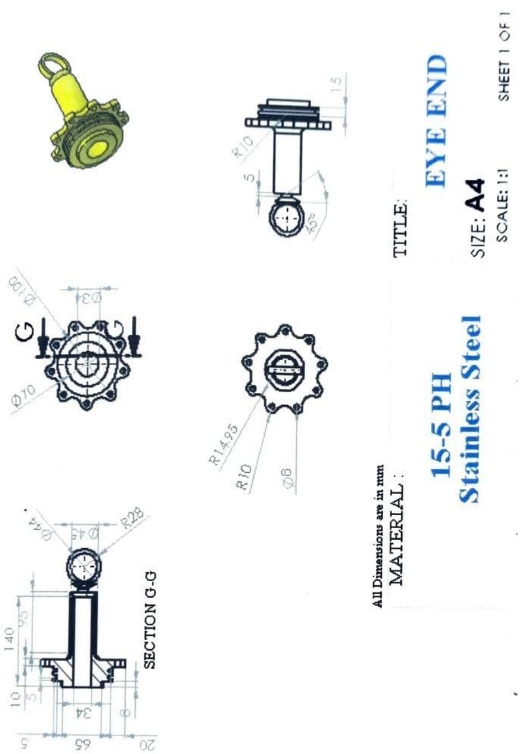
So, a minimum thickness of 5mm is considered and OD is taken as 40mm.

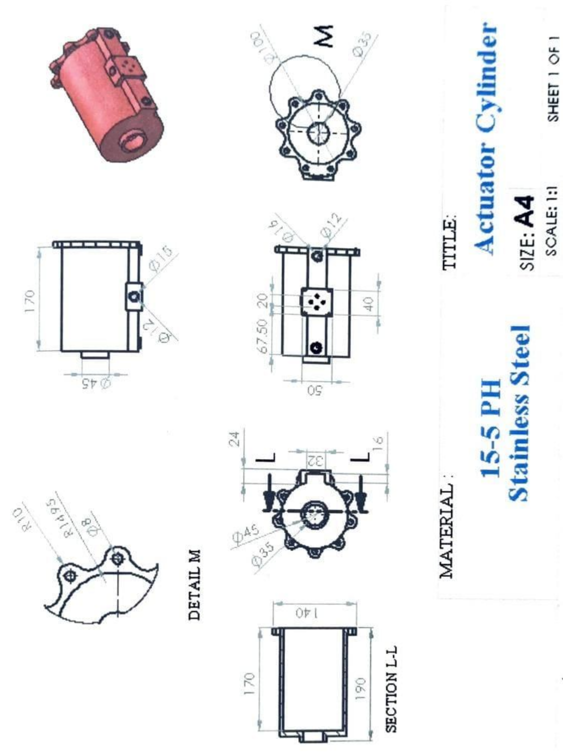
∴The stress induced is :10000//4(40-28)

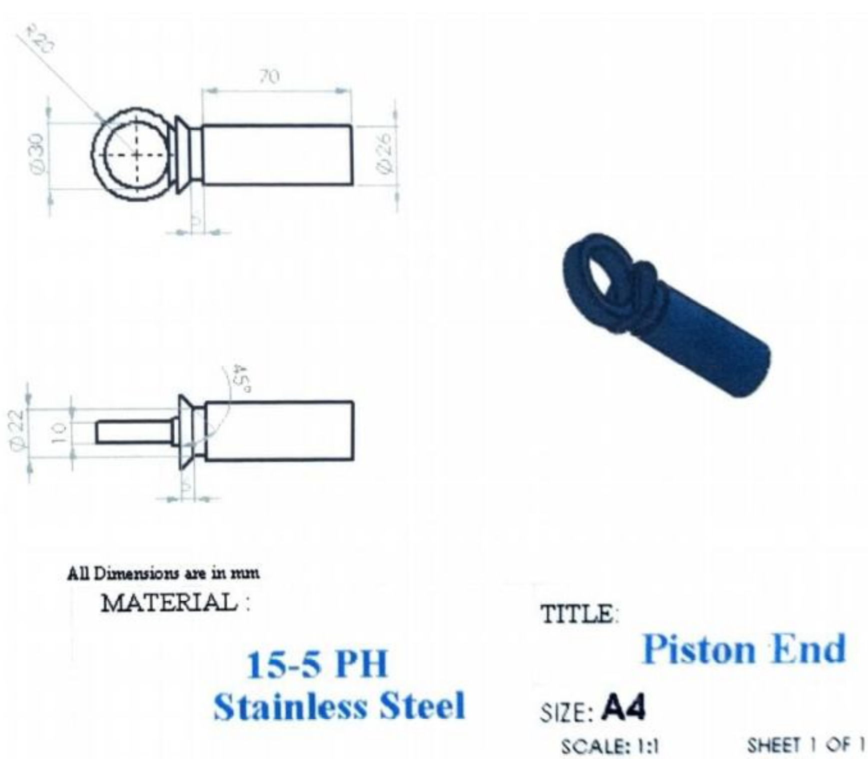
: 15.611 Kgf/mm^2

The Factor of safety is 6.4

∴The minimum factor of safety of the component is 2.







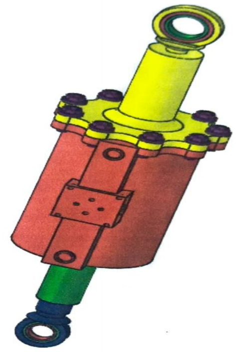


Fig: Hydraulic Actuator

IV. TESTING AND RESULTS

A. Testing Of Actuator

The following tests are performed to ascertain the performance of all the units. All the units shall be subjected to the following tests prior to installation in the system.

1) Examination of the product
2) Operation and leakage test
3) Maximum displacement test:
4) Null off set test and leakage measurement test
5) Threshold test
6) Gain and linearity test
7) Frequency response test

B. Results

TABLE I

Testing of Hydraulic Actuator by using following tests parameters and limits

SI. NO	Parameter	Test Limit	Test Results	Remarks
1.	Cycling	100 cycles	Done	No Remarks
2.	Leak Test	Zero leak	No	At 210 KSC Piston at both extreme positions
3.	Gain	$1.06 \pm 10\%$	1.0382	No Remarks
4.	Null Shift	<750 mV	311 mV	No Remarks
5.	Linearity	< 10%	2.0625	No Remarks
6.	Polarity	+ve signal Piston moves in	OK	No Remarks
7.	Threshold	< 60 mV	30.00 mV	No Remarks
8.	Frequency Response at -90^0 Phase Shift	12-28 Hz	19 Hz	No Remarks
9.	Electrical Inspection • Coil resistance • Insulation Resistance	$500 \pm 50\Omega$ $>50\ M\Omega$		No Remarks
10.	Null Leakage	<0.9 lpm		No Remarks

TABLE II
Spot Frequencies During Frequency Response Test:

Frequency (Hz)	Phase (Deg.) Specification	Phase (Deg.) Observed	Gain(dB) Specifications	Gain(dB) Observed
1	≥ -11	-929	-0.5 to 2.5	0.50
3	≥ -28	-25.12	-1.15 to 2.5	-0.00
5	≥ -44	-38.98	-2.5 to 2.75	-0.89
10	-48 to -85	-64.67	-5.9 to 2.7	-3.74
15	-65 to -115	-80.23	-9.3 to 0.6	-6.38
20	-75 to -132	-92.15	-12.6 to -3.0	-8.67
25	-85 to -145	-101.01	-16 to -5.5	-10.55
28	-90 to -150	-105.81	-18 to -6.5	-11.46

V. CONCLUSIONS

This project aims at study of the actuation system, its types and working of the hydraulic actuator in an aerospace vehicle. The selection of the actuator is accomplished according to the requirements of the vehicle. The design and fabrication of the hydraulic actuator is completed by using solid woks and different methods of manufacturing. To know the performance of the actuator tests like, 100 cycles test, leakage test, linearity test, null shift, gain, polarity, threshold test and frequency response test are conducted. In all the above-mentioned tests the actuator exhibited satisfactory results. Therefore, the design, fabrication and testing of hydraulic actuator is performed successfully.

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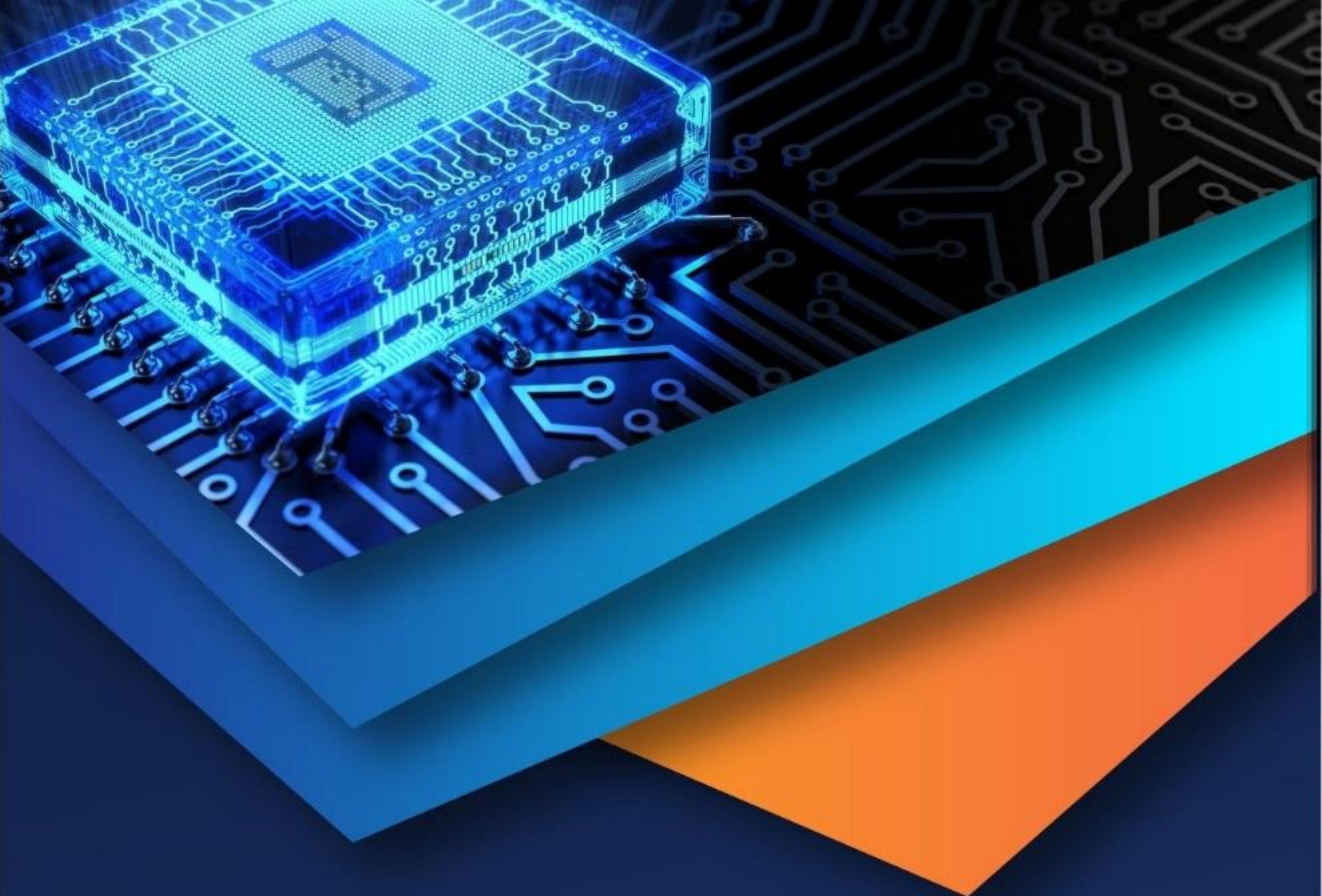
REFERENCES

[1] Mr. S.R. Jadhav, “Design of hydraulic system”. International Journal of Trend in Research and Development, Volume 2(5), ISSN 2394-9333 www.ijtrd.com
[2] Salleh, S. & Rahmat, Mohd & Othman, Siti & A. Danapalasingam, Kumeresan. (2015). “Review on modeling and controller design of hydraulic actuator systems”, International Journal on Smart Sensing and Intelligent Systems. 8. 338-367.
[3] Md Rozali, 1N. Abdul Wahab, 1Zulfatman and Kamaruzaman “Modeling and Controller Design of an Electro-Hydraulic Actuator System” 1M.F. Rahmat, 2S. Jusuf American, Journal of Applied Sciences 7 (8): 1100-1108, 2010 ISSN 1546-9239 © 2010 Science Publications Corresponding Author: Kamaruzaman Jusuf, Faculty of Forestry, University Putra Malaysia, 43400 Serdang, Selangor, Malaysia 1100
[4] Mohd. Fuaad Rahmat, Sahazati md rozali, Dr. Norhaliza Abdul Wahab and Zulfatman Has. “Modeling and controller design of an Electro-Hydraulic actuator system”, American journal of applied sciences 7(8) DOI: 10.3844/ajassp.2010.1100.1108, August 2010.
[5] E. T. Raymond and C. C. Chenoweth, “Aircraft flight control actuation system design”, Society of Automotive Engineers, 1 September 1993.
[6] Maryia Sunil Joshi, A “Design and analysis of hydraulic cylinder using ductile cast iron”. International Research Journal of Engineering and Technology (IRJET), Volume: 08 Issue: 03, Mar 2021 www.irjet.net e-ISSN: 2395-0056 p-ISSN: 2395-0072
[7] Gadi Karthik, R Venkatesh, V Upendra and Neeraj Kumar, “Design and analysis of hydraulic actuator in a typical aerospace vehicle”, Journal for Research | Volume 04 | Issue 02 | April 2018 | ISSN: 2395-7549
[8] S R Majumdar, "Oil hydraulic systems-performance and maintenance", Published dec: 25, 2000
[9] Arthur Akers, Max Gassman, Richard Smith. "Hydraulic system power analysis", Journal for Research January 2006
[10] Keller, George R. "Hydraulic system analysis". Industrial Publishing Company: Cleveland, Ohio.



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