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Design, Analysis & Reliability study of Hydro-pneumatic suspension system

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ABSTRACT

Hydro-pneumatic suspension system(HSS) is designed for taking longitudinal loads under shock in lowering and raising of a containerized object weighing 31 tons and taking thrust load during releasing of the object. Finite element analysis is carried out for HSS system to verify stress level developed in system for various loads. The analytical results are validated against FEM results. Design level Failure Mode and Effect Analysis is carried to find component level criticality observing Risk Priority Number (RPN) & identifying type of failures through systematic methodology of DFMEA. In this paper critical component is designed,analysed & FMEA is carried out. Numerical and analytical stress-strength based reliability is estimated by using normal distribution.

Keywords— HSS (Hydro-pneumatic suspension system), FEM, DFMEA (Design Failure Mode and Effect Analysis), Reliability

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I. INTRODUCTION

Today, suspension is necessary part of automobile world. Requirements of suspension are to minimize accelerations on isolated side, Equalize Variations of Vertical Wheel Forces [1].

Hydro-pneumatic suspension is adaptive i.e. semi-active type of suspension as it can only change the viscous damping coefficient of the shock absorber. Hydro-pneumatic suspension system is combination of hydraulic and gaseous system to achieve desired output suspension effect. Following things happens in it-

a) Combined control of springing and damping elements is considered as effective in terms of decreasing load of parts mounted on it.

b) Spring force is achieved by compressed nitrogen as it is six times more flexible than steel springs.

c) Damping force is generated due to pressure loss in damping valve.

d) Gas absorbs excessive force, whereas fluid in hydraulics directly transfers the force

The simplest hydro-pneumatic suspension system consists of only three components a hydraulic cylinder, a hydro-pneumatic accumulator, which is directly mounted on the cylinder and, of course, the hydraulic fluid. In case cylinder and accumulator need to be separated for example due to design space reasons additional oil lines and fittings are necessary to provide the hydraulic connection [1].

After adjusting the hydraulic pressure to the required level (by adding or releasing hydraulic fluid) this system now already provides the suspension function. When displacing the piston rod, the fluid volume in the accumulator is changed and there that with the pressure. This causes a change of the force at the piston rod which, in combination with the change of the position, defines the spring rate c. To allow for additional damping, a flow resistor is placed between cylinder and accumulator. It converts part of the kinetic energy of the hydraulic fluid into heat (viscous friction). This provides the desired damping in combination with the (undesirable) boundary friction caused by the cylinder sealing and guiding elements. This so called suspension unit consisting of cylinder, accumulator, flow resistor and hydraulic fluid already provides the suspension function and could replace the typical combination of mechanical spring and damper. Basic system is shown below,

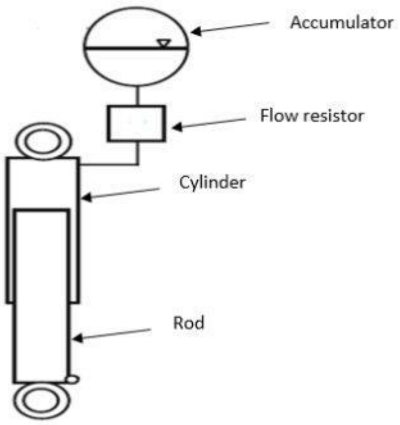


Fig. 1 Basic Hydro-pneumatic suspension system

II. Modified Hydro-Pneumatic Suspension System

Hydro-pneumatic suspension system is designed to support heavy launcher system. Initial conditions of system are as given below,

Initial pressure in cavities = 18 Mpa Total load acting on system = 31 tons No. of suspensions used = 3

Load on single suspension = 10.33 tons Total movement = 65 to -115 mm

Gas used = Nitrogen

Liquid used = Natural oil

Accumulator volume = 2650 cm^3 Temperature range = -10^0 to 40^0c

Pictorial view of system is given below in which small dotted lines shows Nitrogen gas and large dotted lines shows oil.

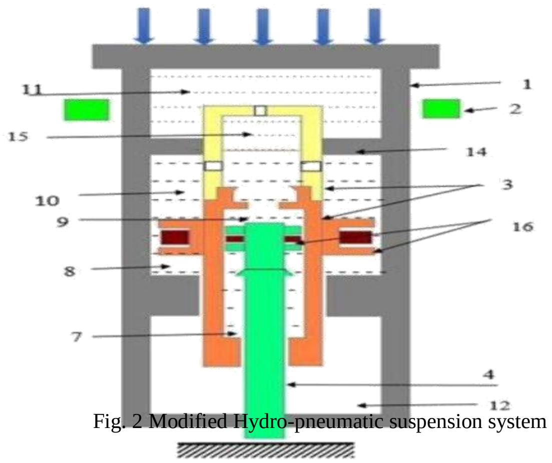


Fig. 2 Modified Hydro-pneumatic suspension system

Nomenclature-1-Casing 2-rest

3-main rod 4-tracking rod

7-second hydraulic cavity of tracking rod

8-first hydraulic cavity of main rod

9-first hydraulic cavity of tracking rod

10-second hydraulic cavity of the main rod

11-main gas cavity

12-air cavity open to the atmosphere

14-journal box

15-intermediate gas cavity

16-reversing valves

III. Spring Characteristics

A. Force Vs Displacement

The most important method to describe the behavior of a spring is the force displacement curve for compression and rebound. It has been mentioned that this curve is basically linear for a regular mechanical coil spring. Here curve is exponential and not straight means spring rate increases at the end. It is advantageous for large load variations.

The following simple relationship is obtained [2][4]:

?? ___________ (1)

?

= ????$_?$?×?$_?$?$^{?.? \times ?}$ Where,

S = area of rod

?$_?$= displacement $^?$?=$_?$ initial pressure

= initial volume

Force Vs Displacement graph is shown below,

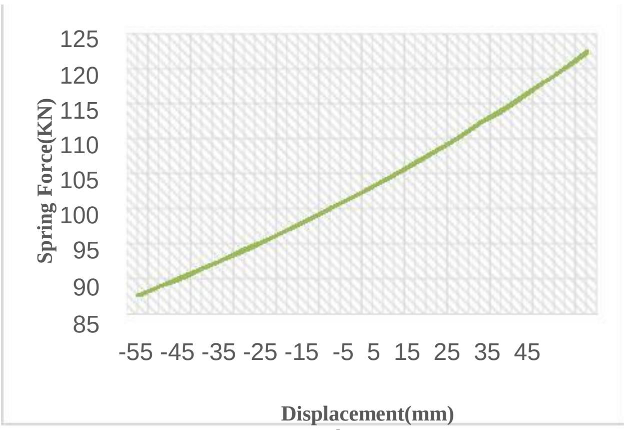


Fig. 3 Force Vs Displacement

B. Spring rate Vs Spring force

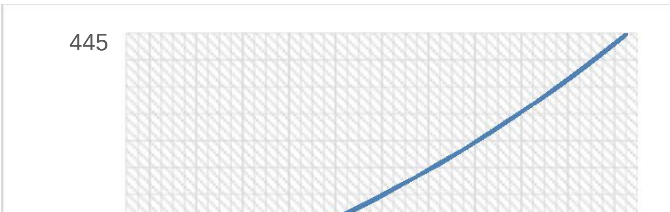
In case a hydro-pneumatic suspension is subjected to a wide range of static spring loads, another important characteristic curve needs to be considered: the dependency of the spring rate on this static spring load. Spring rate is calculated as follows,

Spring rate(c) = ? ________(2)

Where,

F = force applied (KN) $_?$n$_?$= polytropic index

= height of column of gas (mm) S = Displacement (mm)



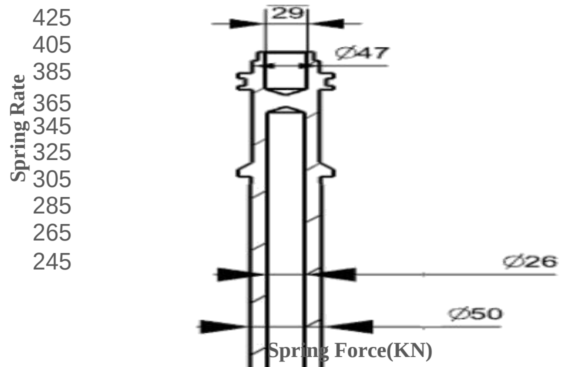


Fig. 4 Spring rate Vs Spring Force

Here the oil volume which is changed during the levelling process so here it is the gas mass which remains constant at all times. Yet this gas mass changes its volume after a load change; a higher load means a smaller gas volume and therefore a higher spring rate

C. Frequency Vs Spring load

It is possible to calculate the natural frequency *f* for the non preloaded hydro-pneumatic suspension [1]:

f = ? ~~?×?×?~~ (3)
~~??~~ ? ?? × ??

Where,

n = Polytrophic index F = force applied (KN)

?g = acceleration due to gravity (m/s^2) ??= initial pressure (KN/m^2)

= initial volume (m^3)

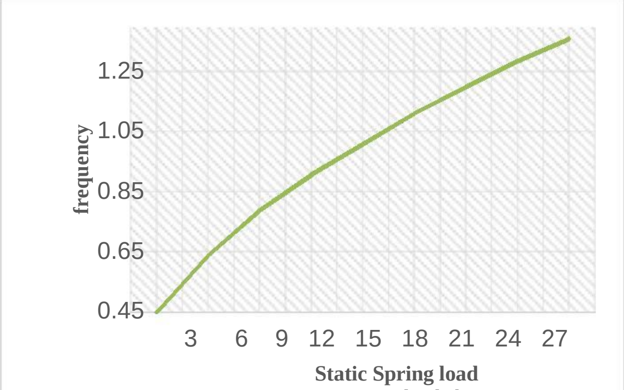


Fig. 5 Frequency Vs Static spring load characteristics

For good protection of the isolated side from the input side, the lowest possible natural frequency and therefore also the lowest possible spring rate needs to be aimed. the natural frequency of a hydro-pneumatic system will more or less increase with increasing loads, depending on the system layout.

IV. DAMPING CHARACTERISTIC

Main purpose of flow resistors(reversing valves) is generating pressure loss and increase damping force. When damping during rebound and compression phase is of different level we use this type of flow resistors. Usually rebound damping is usually chosen to be higher than compression damping, often with a ratio of about 2:1 3:1. This is achieved by different flow paths for the respective flow directions.

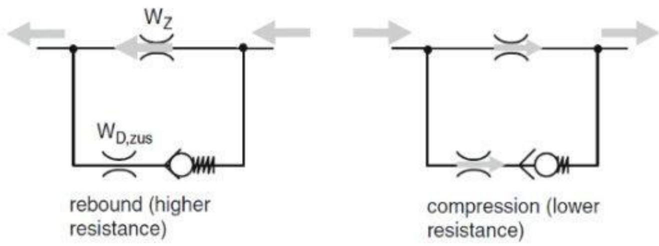


Fig. 6hydraulic circuit for a direction depending resistor

Area of orifice is given by,

?? ?.?.
? (4)
?? = ?.?? Where,
area of displacement of hydraulic cavity
?? = density of fluid used
coefficient of resistant
diameter? of orifice is calculated from value of
Therefore ?

Flow resistors ?for compression stroke and rebound stroke are different.Here flow resistor for compression stroke is ?

attached? to outer rod and flow resistor for rebound stroke is

attached to inner tracking rod. Coefficient of resistance is taken based on history.

1) Area of orifice on main rod in forward stroke,
? =9007088 mm^2
Kg/ m^3
= 10.22 mm.

2) Area of orifice on main rod in reverse stroke,
? =9002243 mm^2
Kg/ m^3
A?= 1.1mm.

3) Area of orifice on tracking rod In forward stroke,
? =9001963 mm^2
Kg/ m^3
A?= 4.28 mm.

4) Area of orifice on tracking rod in reverse stroke,
? =9001963 mm^2
Kg/ m^3
= 1 mm.

V. Design & Analysis of Tracking Rod

A. Design of Tracking Rod

Type thick cylinder
Material: High strength corrosion resistant
steel s_T = 1300 mpa

If we built it with solid material it will results into high component weight with additional increase in unsprang mass. Therefore it is designed hollow. We use Lame s equations to design it. Tracking rod looks like this,

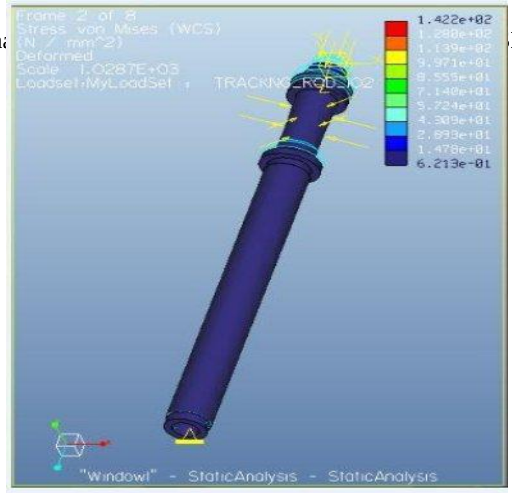


Fig. 7 Cross-section of tracking rod

Under external pressure and absence of internal pressure.
Let,

= internal pressure in the cylinder = 0

??= external pressure in the cylinder

??= internal radius of the cylinder = 29/2 = 14.5 mm

r?= external radius of the cylinder = 50/2 = 25 mm

r?= mean radius of the cylinder = ????? = 19.75 mm

By using Lame s equation-

? ?

∴Stress in circumferential direction is

???.??????.???? ???.???(?????) (5)

And Stress?in tangential? direction??? is

?? = ? ?? ??? ?− ? ? (?? ???) ?

???.??????.?? ?? ???.???(?????) (6)

?? = ? ??????? ?+ ? ??(?????₂?) ?

= force /area = 250000/4261 = 53 Mpa

??Hoop stress ( ) = -2.31×

= -2.31×53

=∴ -122.43 Mpa ?? ??

Longitudinal stress (- 0.69×

= - 0.69 ×53

=∴ -36.57 Mpa ??) = ??

∴ Safety factor (h) = ???????.?? > 10

B. Static Analysis Of Tracking Rod

Figure number 8 shows FEA result of tracking rod. Von mises stress is shown by red colour which is not visible in figure. As factor of safety provided is in range of 5-7 so there will be less very chances of failure.

Fig. 8 Von mises stress calculated by software

VI. Failure mode & effect analysis of tracking rod

The Failure Modes and Effects Analysis (FMEA), also known as Failure Modes, Effects, and Criticality Analysis (FMECA), is a systematic method by which potential failures of a product or process design are identified, analyzed and documented. Once identified, the effects of these failures on performance and safety are recognized, and appropriate actions are taken to eliminate or minimize the effects of these failures. An FMEA is a crucial reliability tool that helps avoid costs incurred from product failure and liability.

The FMEA process is an on-going, bottom-up approach typically utilised in three areas of product realization and use, namely design, manufacturing and service. A design FMEA examines potential product failures and the effects of these failures to the end user.

Table 1 FMEA of Tracking Rod

Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	SEVi	Classification	Potential Cause(s) of Failure	OCCi	Current Design Controls	DETi	RPNi
Pushes main rod upward and after some time locks the system by touching his extended part to Bearing box	Failure due to high pressure	System fail	9		Insufficient structural margin	1	Sufficient margin is considered	1	9
					Insufficient fatigue margin	2	Design simulation & Testing	3	54
					Wrong material selection	1	Suitable material considered	1	9
					Suddenly applied load	4	Design & Testing	2	72
					Pressure transients	5	Testing	5	225
	Thread failure(Tearing/shearing)	Instability	5		Wrong thread design	2	Already proven design	4	40
					Overloading	4	Design & testing	2	40
					Insufficient torque	3	Proper precaution	3	45
					Improper assembly	2	Assign procedure	4	40
	Corrosion	Reduce life	3		Fluid Contamination	3	Quality fluid	3	27
					Oil/Coating Incompatibility	2	Fluid quality and purity	2	12
					Coating failure	7	Qualification of coating process & Testing	6	126
	Effect of temperature increased due to compression of fluids	Rod failure	8		Continuous operation	7	Operational life	4	224
					contamination	3	Quality fluid	3	72
					Poor thermal design	2	simulation Design & Testing	3	48
	Wear and tear	Leakage, improper working	3		Misalignment	4	Precautionary measure	3	36
					Inadequate lubrication	3	Check points	3	27

The **failure mode** that describes the way in which a design fails to perform as intended or according to specification;

The**effect** or the impact on the customer resulting from the failure mode; and

The**cause**(s) or means by which an element of the design resulted in a failure mode.

As tracking rod is critical part so it is necessary to do FMEA of it. RPN number is calculated which shows criticality of causes. Table number 1 shows result of FMEA of tracking rod.

VII. Reliability Based On Normal Distribution

Let,

= mean value of strength of material.

??= mean value of calculated stress value.

??= standard deviation of strength value of material.

??= standard deviation of calculated stress value.

??= reciprocal of mean value of stress

? = reciprocal of mean value of strength

Safety factor (SF) =

? ? ??? ???????????????????????? ??

? ??? ??????? ????????? ?????? = ??

As much data is not available with us that s why standard deviation used is of 10% of mean value. Here stress and strength associated with an item are normally distributed.

Reliability(R) = ?? (7)

Where, ∫?? √?? ? ?? ????

z = standard normal variate = - ????? (8)

????????

As tracking rod is of high strength which is near about 1400 Mpa so reliability is coming up to 99.99. In fact we can say that for having reliability of more than 99% material whose strength is in range of 1300-1400 Mpa is selected. As maximum stress for tracking rod is coming as 122.43 Mpa so aim is fulfilled.

VIII. CONCLUSIONS

- This work discusses design and failure analysis of hydro pneumatic suspension system.
- Spring and damping characteristics are calculated.
- Analysis result shows that von mises stress of tracking rod is 142 Mpa and analytical result gives value of 123 mpa which are very close.
- A design FMEA of one of the important part i.e. tracking rod is presented which shows that RPN for causes like pressure transient, continuous operation, coating failure are critical.
- Also strength based reliability is estimated by calculating standard normal variate and is coming 99.99.

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