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Research on Stiffness of Hydraulic Pad in Hydraulic Overload Protection System

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Abstract. The hydraulic overload protection system is widely utilized with the advantages of high precision control, suitable for multi-point press and automatic reset. The stiffness of the hydraulic pad is a key factor affecting the corresponding performance. In contrast, the stiffness value of the hydraulic pad and the effects of the pipe length, the thickness and the diameter to the hydraulic overload protection system dynamic response characteristics acquisition have not been studied systematically. In this paper, the methods of theoretical analysis, finite elements, system modeling and numerical simulation were utilized to obtain the change rule of hydraulic pad stiffness and the affecting factor in the dynamic response of the hydraulic overload protection system. The results indicated that the method to obtain the stiffness of hydraulic pad was effective and the hydraulic pad stiffness gradually increased as the pressure increased. The pipe would reduce the stiffness of the hydraulic pad. As the pipe length increased, in the same pressure range, the stiffness of the hydraulic pad was reduced. In addition, the peak level of the dynamic response pressure in the pipeline was affected by the pipe length, the pipe diameter and the pipe thickness. The pipeline diameter had the highest effect on the peak pressure, whereas the pipe thickness had the least effect.

Keywords: Stiffness, Hydraulic Pad, Hydraulic Overload Protection System

INTRODUCTION

The forging processing technology is widely utilized in automobiles, aviation, instrumentation, light industrial machinery and other fields，due to the corresponding high efficiency, good quality, light weight and low cost [1,2]. Additionally, with the application of large-scale parts and the practical utilization of the high-sized vertical continuous automatic press, the large-scale has currently become an important development direction of the press machine [3-7]. In contrast, the occurrence frequency of the overload phenomenon becomes increased along with the large-scale. The overload will cause damage to the press, such as to the connecting rod thread damage, the screw bending, the fracture of crankshaft bending and even the deformation and body fracture. These hazards will highly reduce the product efficiency and increase the economic losses [8,9]. Therefore, effective measures must be taken to prevent the occurrence of overload or to protect the press from damage in the event of an overload.

At present, the overload protection device is mainly of three types: the mechanical, the hydraulic and the gas-liquid. The hydraulic overload protection system is widely utilized with the advantages of high precision control, suitable for the multi-point press and the automatic reset [10]. In contrast, the design theory of the hydraulic overload protection device installed in the mechanical press is not perfect. The hydraulic system is simply transplanted from the common working principle of the hydraulic system, where no further systematic study of the mechanical presses hydraulic overload protection device characteristics, often cause the corresponding work stability, overall structure high cost, as well as the inconvenience in the utilization and repair [11,12].

The overload protection equipment of the press is generally installed in the sliding block of the press, which is mainly utilized for the overload protection during the work of the press, which constitutes an unloading device during the work duration when the load exceeds the load rate of 100% ~ 110% [13,14]. The overload value is derived from experience and it is impossible for precise control to be achieved.

The hydraulic overload protection system mainly includes the hydraulic pad, the hydraulic pipes, the unloading valves and the pressurized systems. The principle of this sort of hydraulic overload safety device is: between the slide and the driving device a connector exists. The connector is composed of three parts, a ring, a cylinder and a piston, as presented in **FIGURE. 1**. The driving devices are connected to the piston, whereas the slide is connected to the cylinder frame, in order for the driving devices and the slide to move relatively, if no liquid exists in the cylinder. The ring is utilized to restrict the upward movement of the piston. The cylinder volume below the piston is full of high-pressure liquids. The cylinder body is connected to the pipeline and the unloading valve. The liquid inside the cylinder and the pipeline is called hydraulic pad. If the liquid pressure in the cylinder becomes higher than the preset pressure value of the unloading valve, the hydraulic system will perform an action to unload. Consequently, the driving devices and the slide can move relatively to avoid the machine press and the die damages, by the liquid discharge through the unloading valve [15-17].

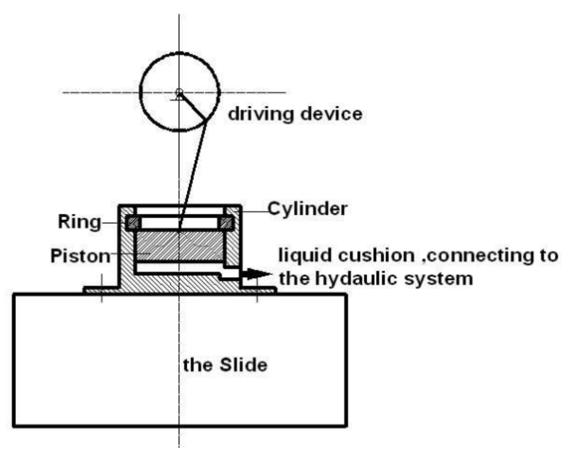

FIGURE. 1 Basic Components of Machine Press

In contrast, in the hydraulic overload protection system, the hydraulic pad stiffness is an important factor that affects the control accuracy and the molding quality [18].When the temperature is certain, the fluid volume will decrease as the pressure increases, representing the compressibility of the fluid [19]. In general, the theory of incompressible fluid can be utilized to solve the liquid balance and movement problems, whereas when the fluid compressibility critical phenomena of the water hammer and the hydraulic impact, the fluid compressibility must be considered. At present, when the hydraulic overload protection system is designed, the liquid is often utilized as an incompressible fluid, causing the early unloading phenomenon impossible to be explained scientifically.

Under the high pressure of the liquid and the piston deformation, the cylinder and pipeline affect the hydraulic pad stiffness. The rigidity characteristic change affects not only the punching precision, whereas it also affects the unloading control accuracy [20].

During the stamping process，the hydraulic pad will transfer the force from the piston to the slide. The optimum condition is that the working force is equal to the connector acting force [1], which is determined by the stiffness of the hydraulic pad. In contrast, the stiffness of the hydraulic pad is affected by many factors, constituting the of the hydraulic pad stiffness difficult to be calculated. At present, the matching problem of the stiffness of the connector and the hydraulic pad is determined through experimentation, which causes inconvenience in the practical work.

In summary, the current research on the hydraulic pad stiffness has not yet produced a systematic theory. The calculation formula utilized mainly depends on experience, whereas the liquid is often treated as an incompressible fluid in the computational formula, which makes it difficult to display the precise unloading and the control precision is poor. Due to the lack in theory, leading to the phenomenon of early unloading cannot produce a reasonable explanation. Therefore, in this paper, a method through theoretical analysis, simulation and numerical calculation to solve the aforementioned problem was proposed. In addition, the hydraulic pad stiffness effects on the pressure-transfer in the hydraulic overload protection systems were also analyzed.

COMPRESSED VOLUME OF LIQUID IN RIGID CONTANINER

Principle of Compression

To discuss the deformation of the liquid under certain pressure, with the assumption that the liquid container is rigid, the initial pressure of the liquid is P_0, whereas consequently the pressure increases to P. According to the liquid compressibility definition, the liquid volume is reduced. The volume compression rate κ of the liquid is [15]:

$$\kappa = -\frac{1}{V} \cdot \frac{dV}{dp} \tag{1}$$

where, V is the initial volume of the liquid under the initial pressure, it can be regarded as constant V_0. It can be written as:

$$\kappa \cdot dp = -\frac{dV}{V_0} \tag{2}$$

In addition, the liquid volume compression rate is the reciprocal of the bulk modulus:

$$\kappa = \frac{1}{K} \tag{3}$$

where, K is the bulk modulus of the liquid. According to the previous experiments, when the oil temperature was 50℃, the bulk modulus of elasticity was:

$$K = K_0(1 - e^{-0.25p-0.2}) + 1.85p \tag{4}$$

where, K_0 is 1500MPa.

When the liquid pressure increases from P_0 to P, the liquid volume will be reduced from V_0 to V. Consequently, the following equation can be concluded:

$$V_0 \int_{P_0}^{P} \kappa dp = -\int_{V_0}^{V} dV \tag{5}$$

Through Eqs. (3) and (4) substitution into Eq. (5):

$$V_0 \int_{P_0}^{P} \frac{1}{K_0(1-e^{-0.25p-0.2})+1.85p} dp = -\int_{V_0}^{V} dV = V_0 - V = \Delta V_l \tag{6}$$

where, ΔV_l is the compressed volume of the liquid in a rigid container. The integral results could be obtained by MATLAB [21].

Initial liquid volume

The liquid cavity of the slide hydraulic protection system consists of three connected parts: the cylinder liquid chamber, the pipeline and unloading valve chamber. As the cavity shape of the cylinder liquid chamber and the unloading valve chamber are irregular, it is difficult to directly calculate the liquid volume in the cavity. In this paper, the volume of liquid in the cavity was calculated through modeling software.

The model and volume of the liquid in the connector and the unloading valve are presented in **FIGURE. 2** and **FIGURE. 3**. It could be obtained that: $V_c = 3315466mm^3$, $V_u = 133645mm^3$. Where, V_c is the liquid volume in the connector and V_u is the liquid volume in the unloading valve.

The liquid volume in the pipe can be calculated through simple calculations:

$$V_p = \frac{\pi}{4} d^2 l \tag{7}$$

where, d is the pipe diameter, d=89mm and l is the pipe length.

The initial liquid volume is:

$$V_0 = V_c + V_u + V_p \tag{8}$$

FIGURE. 2 Model and volume of liquid in connector

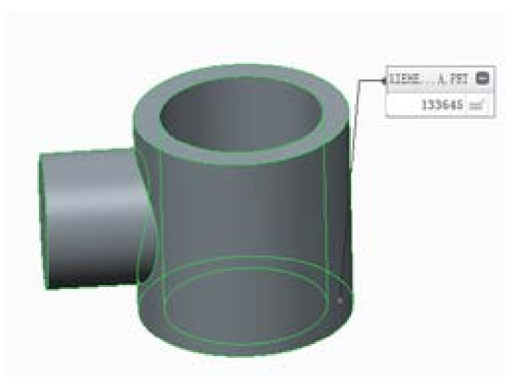

FIGURE. 3 Model and volume of liquid unloading valve

Cylinder deformation

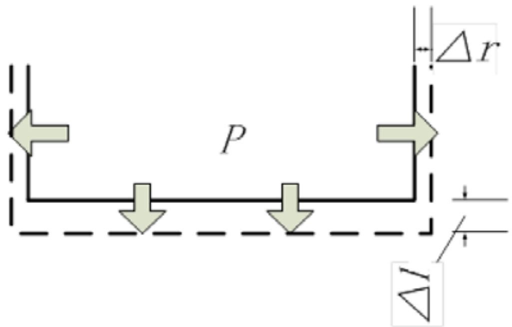

FIGURE. 4 Cylinder deformation illustration

The cylinder deformation illustration is presented in **FIGURE. 4**. Where, Δr is the cylinder increase in the radial direction under a load, Δl is the increase of the cylinder in the axial direction under a load. The Δr and Δl could be obtained by the ABAQUS software. The container volume increment can be expressed as:

$$\Delta V = \pi \cdot (r + \Delta r)^2 \cdot (l + \Delta l) - \pi \cdot r^2 \cdot l \tag{9}$$

The changes in the cylinder values under 5MPa are listed in **TABLE 1**.

TABLE 1. Container volume increment at 5MPa

Δr /mm	Δl /mm	ΔV /mm^3
1.251E-3	0.0644	7930.57

Pressure interface movement

As presented in **FIGURE. 5**, the force F is applied to the upper surface of the liquid and the surface moved by x, due to both liquid compression and container deformation.

$$x = \frac{\Delta V_l + \Delta V}{S} = \frac{\Delta V_l + \Delta V}{\frac{1}{4}\pi d^2} = 4 \cdot \frac{\Delta V_l + \Delta V}{\pi d^2} \tag{10}$$

where, x is the movement of the pressure interface. S is the area of the liquid pressure interface and d is the diameter of the pressure interface.

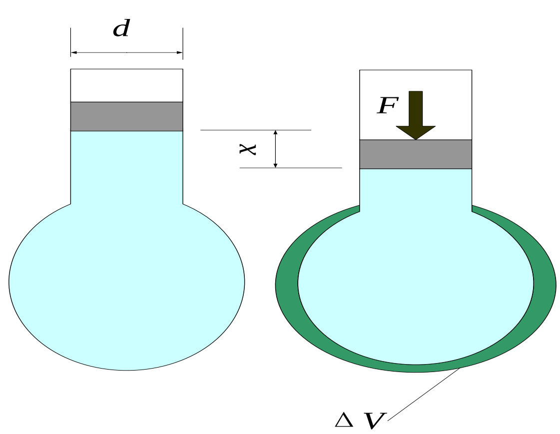

FIGURE. 5 Schematic diagram of pressure interface movement

Stiffness of hydraulic pad at different pressure intervals

TABLE 2. Volumes of liquid compression at different pressures

Pressure/MPa	ΔV_l /mm^3
2.1	1.32e+04
4.1	2.06e+04
6.1	2.66e+04
8.1	3.19e+04
10.1	3.69e+04
12.1	4.16e+04
14.1	4.63e+04
16.1	5.09e+04
18.1	5.55e+04
20.1	6.00e+04
22.1	6.45e+04
24.1	6.90e+04

With the assumption that the initial pressure of the liquid P_0 is 0.1 MPa, the volumes of liquid compression with different pressures were calculated and the results are presented in **TABLE 2**. The hydraulic pad stiffness was calculated at every 2 MPa, based on the data in **TABLE 2**. The results are listed in **TABLE 3**.

TABLE 3. Stiffness of hydraulic pad at different pressures

pressure intervals /(MPa)	stiffness/(KN/mm)
2.1-4.1	4032.98
4.1-6.1	5047.03
6.1-8.1	5652.31
8.1-10.1	6012.86
10.1-12.1	6349.33
12.1-14.1	6432.65
14.1-16.1	6526.87
16.1-18.1	6593.60
18.1-20.1	6638.25
20.1-22.1	6701.78
22.1-24.1	6641.84

It could be observed from **TABLE 3** that as the pressure increased, the hydraulic pad stiffness gradually increased. In addition, the stiffness values of the hydraulic pad in the pressure zone of 2.1 MPa to 4.1 MPa and the stiffness in the zone of 22.1 MPa to 24.1 MPa differed significantly. Therefore, it was not feasible to assign a specific stiffness value to the hydraulic pad, whereas a significantly reasonable way was to mark the stiffness curve.

PIPE VOLUME EFFECT ON HYDRAULIC PAD STIFFNESS

The diameter of the pipe was constant, d = 89mm, the pipe material was the 20 steel, the elastic modulus was 206GPa, the Poisson's ratio was v = 0.3, whereas the length of the pipeline varied. The pressure was 5MPa.

TABLE 4. Volume increase of different pipe lengths at 5MPa

pipe length/m	ΔV /mm^3	V/mm^3
0.5	2328313	5777423.816
1	4656626	8105736.631
1.5	6984938	10434049.45
2	9313251	12762362.26

TABLE 4 presents the volume increase of different pipe lengths at 5MPa. According to Eqs. (1) to (6), the liquid compression volume could be calculated. Consequently, the pressure interface movement could be obtained from Eq. (10). The results are presented in **FIGURE. 6**. As the pipe length increased, in the same pressure range, the pressure interface displacement increased, meaning that the stiffness of the hydraulic pad was reduced. Therefore, the pipe would reduce the stiffness of the hydraulic pad.

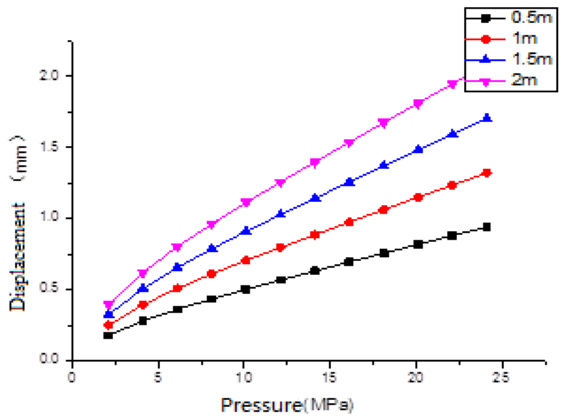

FIGURE. 6 Displacement curve of hydraulic pressure interface versus pressure

PIPELINE DIMENSION PARAMETER EFFECT ON HYDRAULIC PAD STIFFNESS

In the experiments of hydraulic pad stiffness, it was apparent that the stiffness and pressure peak of the hydraulic pad were affected by the dimension parameters of the pipeline[22]. To study the dynamic effect of the pipe length, the pipe diameter and pipe thickness on the maximum pressure peak, the AMESim was utilized to build the simulation model, as presented in **FIGURE. 7**.

In the model, the working load of the connector was 2500KN, the hydraulic pad piston area was 1195cm2, the maximum working pressure was 20.9MPa and the pre-pressure of the hydraulic pad was selected as 70% of the maximum working pressure, whereas the unloading pressure of the unloading valve was 110% of the maximum working pressure of the hydraulic pad. All parameters were set according to the actual situation and design dimensions.

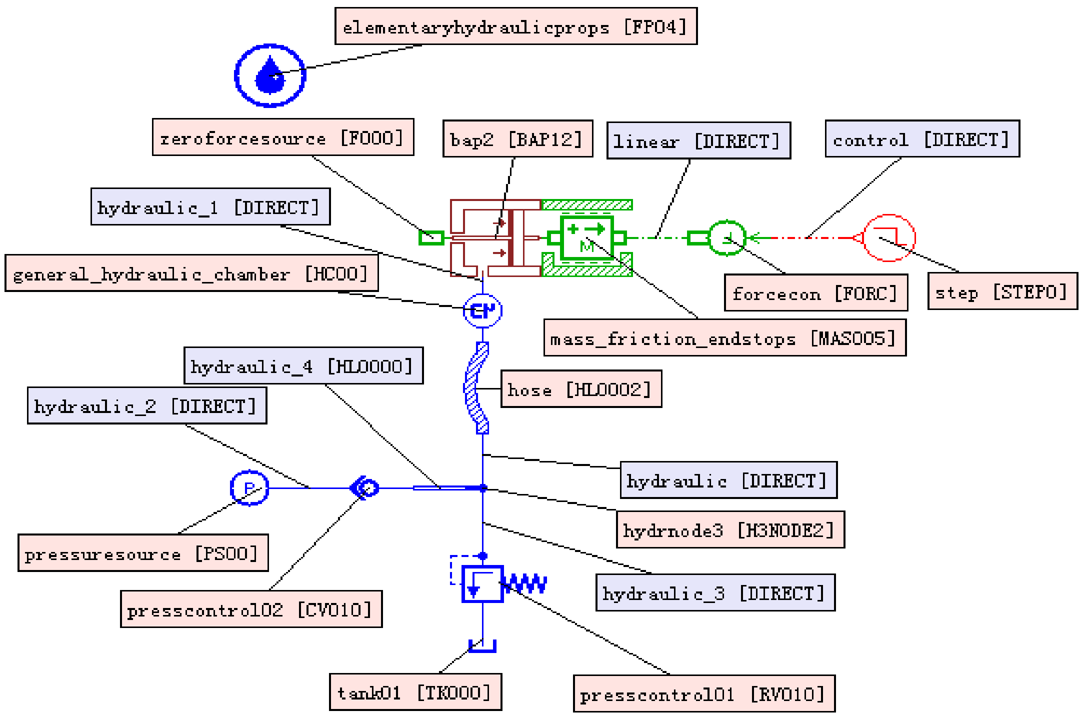

FIGURE. 7 Simulation analysis model

Under the step input signal, the simulation model system would have a dynamic response[23]. As presented in **FIGURE 8**, the maximum pressure inside the hydraulic pad was 23.96 MPa. As presented in **FIGURE. 9**, the maximum unloading pressure of the unloading valve was 23.43 MPa. It was apparent that the pipe could decrease the peak value of the hydraulic pressure. In contrast, the max pressure values of the hydraulic pad and the unloading valve were higher than the unloading pressure of the unloading valve which was set at 23.03 MPa. The occurrence of this phenomenon was due to the liquid compressibility and the result of the coupling between the liquid and the solid. The liquid could be regarded as a hydraulic spring, which under the step load, would cause pressure fluctuations. The result was that pressure relief would be caused in a short time.

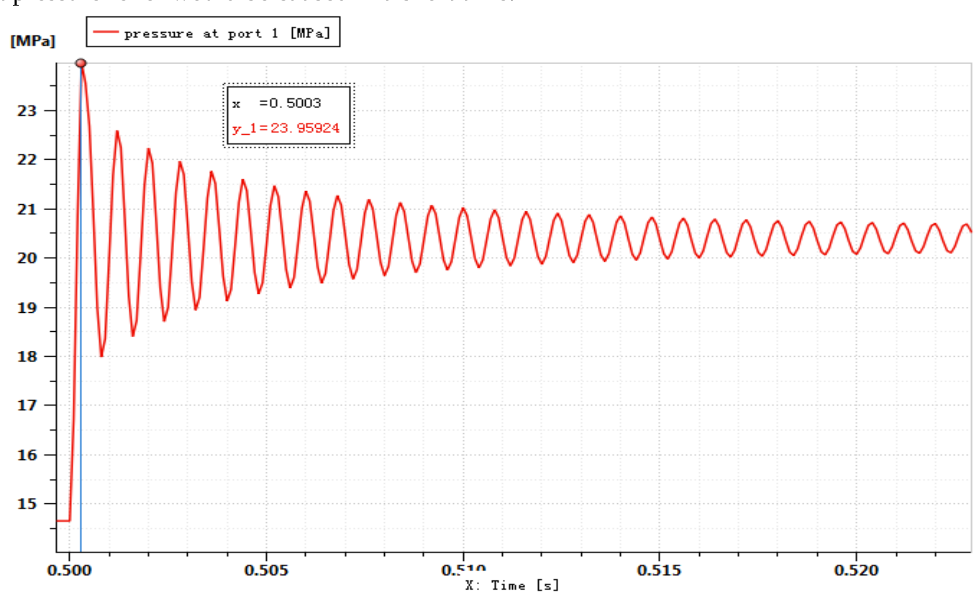

FIGURE. 8 Working pressure in hydraulic pad

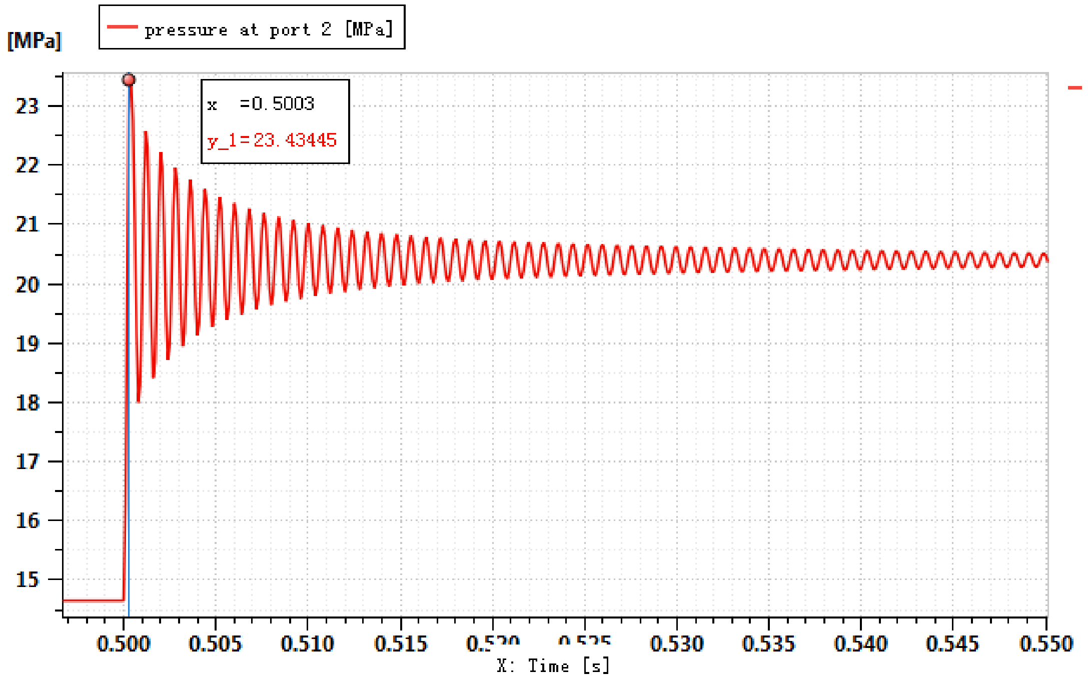

FIGURE. 9 Curve of unloading valve front port pressure

To analyze the dynamic effects of the pipe length, the pipe diameter and the pipe thickness on the maximum pressure peak, the Isight software [24] was utilized to build the Latin hypercube testing scheme and analyze the experimental results. The results are presented in **FIGURE. 10**, **FIGURE 11** and **FIGURE. 12**. **FIGURE. 10** presents the Pareto plot[24] for the max pressure, where it was apparent that the most significant parameter affecting the peak value of the pressure was the pipeline diameter, whereas consequently it was the pipe length, however, the pipe thickness effect was low. As presented in **FIGURE. 11** and **FIGURE. 12**, the higher the diameter, the shorter the length and the higher the maximum working pressure peak. On the contrary, the lower the diameter was, the increased the length and the lower the maximum pressure peak. In conclusion, to obtain the lower pressure, the pipe length should be increased and the diameter should be lower.

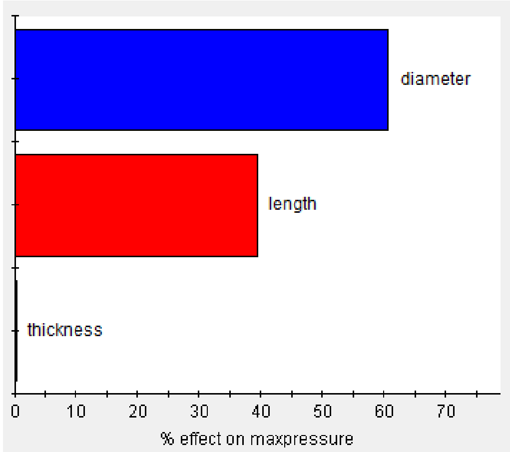

FIGURE. 10 Pareto plot for max pressure

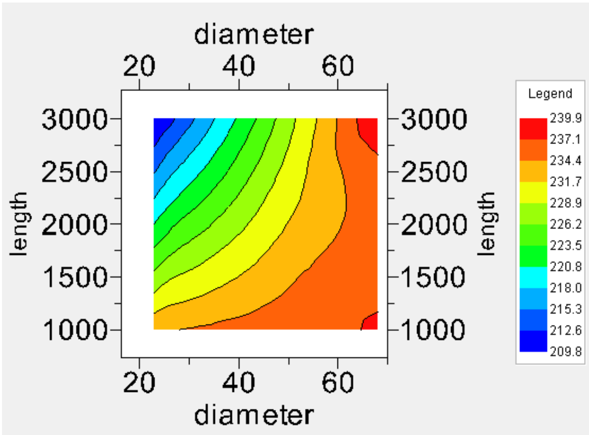

FIGURE. 11 Contour of max pressure with diameter and length

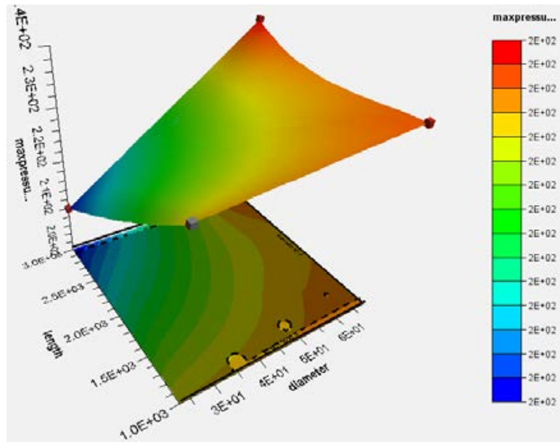

FIGURE. 12 Constraint overlay 3D of max pressure with diameter and length

CONCLUSION

In this paper, a method through theoretical analysis, simulation and numerical calculation to obtain the hydraulic pad stiffness was proposed and the pipe volume effect on the hydraulic pad stiffness was investigated. The following results were obtained:

(1) The method to obtain the hydraulic pad stiffness was effective;

(2) As the pressure increased, the hydraulic pad stiffness gradually increased;

(3) The pipe would reduce the hydraulic pad stiffness. As the pipe length increased, in the same pressure range, the hydraulic pad stiffness was reduced;

(4) The peak value of the pressure in the pipeline was affected by the pipe length, the pipe diameter and the pipe thickness. The most significant parameter was the pipeline diameter, whereas consequently it was the pipe length, however, the pipe thickness effect was low.

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