Design and Fabrication of a Pneumatic Punching Machine

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Abstract: Pneumatics is a branch of engineering that involves the study and application of pressurized gas to produce mechanical motion. It focuses on the properties of air and gases and their use in various engineering applications. Pneumatic systems are power transmission systems that use compressed air as the working medium. Their principle of operation is similar to hydraulic systems, which use pressurized oil for power transmission. Pneumatic systems are extensively used in industries where compressed air or inert gases are employed. The reason for this is that a centralized and electrically powered compressor, which powers motors, cylinders, and other pneumatic components via solenoid valves, is often a suitable and efficient power source. Pneumatic systems are increasingly preferred due to their high efficiency, low maintenance costs, and safety. Pneumatic sensing systems, also known as air sensing control systems, are frequently used to control industrial processes and correspond to primary sensing units. These systems are reliable and functional for controlling synthetic processes. One such application is in punching machines, which are widely used in manufacturing. This work will focus on the different components of a punching machine and their assembly, as well as the relevant diagrams.

Keywords: *Pneumatic; Punching; Control Valves; Cylinders; Circuit; Time Frame.*

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I. INTRODUCTION

In 2008, the Pneumatic Center at Huazhong University of Science and Technology was one of the most active research centers in China regarding fluid power transmission and control. The center specializes in component development and control technologies related to fluid power. It focuses on integrating electronics, computer technology, and pneumatics. The research achievements from this center have played a vital role in the success of several national key engineering projects. The center has developed various pneumatic technologies, including gas temperature control, pressure and vacuum servo control, leak-testing, pneumatic muscle platforms, and high-pressure pneumatic valves. Additionally, practical examples and research projects in these fields will be presented.

In pneumatic cylinders, air at a certain pressure is typically exhausted into the atmosphere after the work process. It is significant that this air energy can be recovered and reused. In 2005, Yunxu SHI studied the constitution of an exhausted air reclaiming system for pneumatic cylinders. The study investigated the effect of the system on the cylinder's velocity characteristics and tested different control switch points. Experimental results showed that attaching a reclaiming device would not negatively affect the velocity stability if the switch point was properly controlled.

Most manufacturing industries have adopted automation to increase productivity and overcome the shortage of skilled labor. Pneumatic systems have gained significant importance over the past few decades due to their accuracy and cost-effectiveness. The ease of operating pneumatic systems has led to the design and fabrication of this unit as a project. This unit is intended to be operated by semi-skilled operators. Pneumatic punch tools have the advantage of working under low pressure; even a pressure of 10 bar is sufficient for operation. Pressurized air travels through tubes to the cylinder, forcing the piston to move and transmitting the power through a linkage to the punch. The workpiece is then shaped to the required dimensions and can be collected through the clearance provided in the die. The die is designed to accommodate various shapes, enabling the production of a wide range of products. Different types of

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punches can be used based on the material being processed, which allows for flexibility in the types of products produced.

In today's world, advancements in manufacturing and machining processes have reduced production time, leading to increased productivity, which has a significant impact on mass and batch production. This project work involves the design of a pneumatically controlled small-scale punching machine to perform piercing operations on thin sheets of various materials. The main aim of this project is to reduce the required punching force.

In recent times, some of the control systems have been largely replaced by electrical control systems due to the lower size and lower cost of the electrical factors. On the other hand, pneumatic servo systems are still used in artificial processes where compressed air is the only available energy source or where cost, safety and other considerations overshadow the advantages of ultramodern digital control. Glickman [1], illustrated a complete description of such types of these Valve Controlled Actuators "VCA". The study of these VCAs and their basic components have been discussed in some figured publications [2-4]. This area of engineering is concerned with the study and operation of pressurized gas to induce both linear and rotary mechanical motions. As figured in references [5-9], compressed air is used in fixed installations similar to manufacturers as the contraction of atmospheric air allows for a sustainable force. Clean, dry, and reliable supply of the used compressible fluid must be ensured. This is done by removing any percentage of humidity by using any mechanical or chemical method. In addition, a small quantity of oil is used in the compressor as a lubricant. It also helps in prevention of corrosion of mechanical components. With plants connected with pneumatic systems, one does not need to worry about poisonous leaks as the gas is generally just air. Lower or standalone systems may use other compressed feasts that pose an asphyxiation trouble, similar to nitrogen, which is constantly appertained to as "OFN" when supplied in cylinders. All compressed gasses other than air are a source of a choking hazard. For example, nitrogen which is not an asphyxiation hazard, is an extreme fire hazard. For this fact, it is not used in pneumatically operated outfits. Compressed carbon dioxide is used in some mobile pneumatics tools. This is similar to bouncy drink bottles and fire extinguishers which are readily available and the phase change between liquid and gas allows a lower volume of compressed gas to be pulled from a lighter vessel than would be possible with compressed air. Carbon dioxide is asphyxiated and can also pose a frostbite hazard if not vented duly. Both pneumatics and hydraulics are operations of fluid power. Pneumatics uses compressible fluids like gasses similar to air, while hydraulics uses fairly incompressible fluids similar to mineral or Synthetic oils.

II. CIRCUIT DESIGN OF CASE STUDY

According to the required function of the machine, the circuit has three control positions.

- *Neutral Position*

In this position, all control valves are in a neutral position therefore the punching cylinder C1 is actuated in "extension mode and punching position" while the cylinder C2 is retracted as shown in the following Fig. (1).

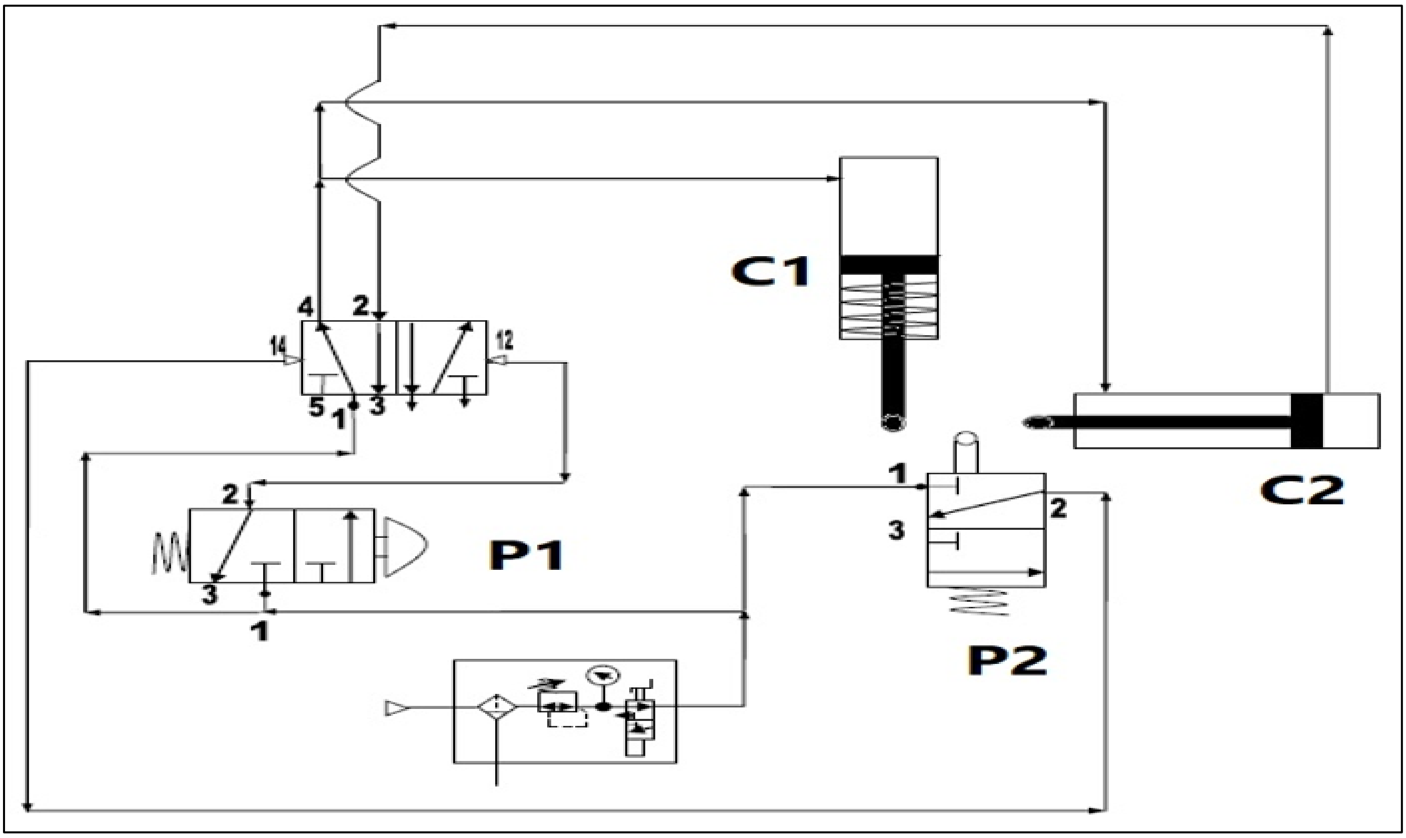

Fig 1 Neutral Position

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➢ *First Actuated Position*

In this position, the 3/2 push button DCV P1 is operated manually to actuate the 5/2 DCV, while the roller 3/2 DCV P2 is in the neutral position. The punching cylinder C1 is neutralized in "Retract mode and ready to punch", while cylinder C2 is actuated in "Extension mode" to actuate the roller 3/2 DCV P2 by its full extension, as it is declared in the shown Fig. (2).

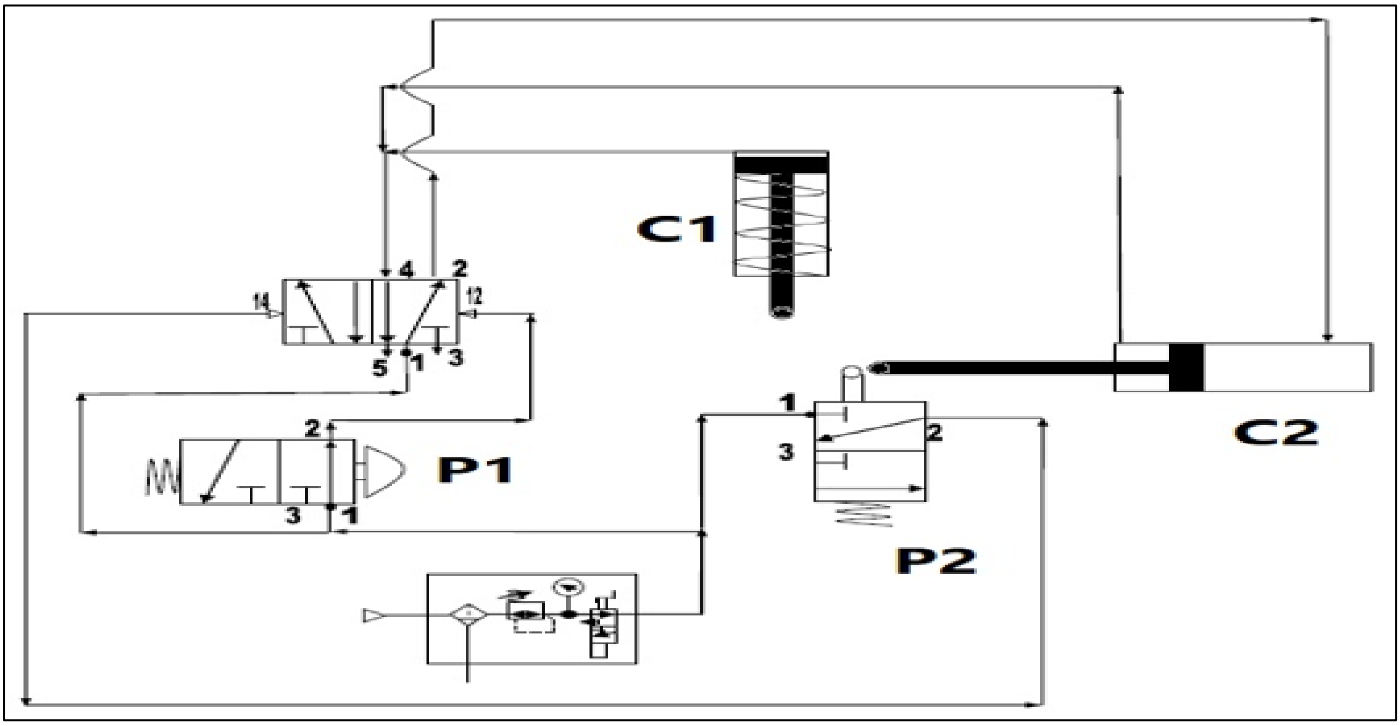

Fig 2 First Actuated Position

➢ *Second Actuated Position*

In this position, cylinder C2 actuates roller 3/2 DCV P2 at the end of its extension movement. The 5/2 DCV has neutralized again so that cylinder C1 is in "extension mode and in punch position" and cylinder C2 can be retracted. By the end, the machine returns to its initial neutral position as illustrated in Fig. (3).

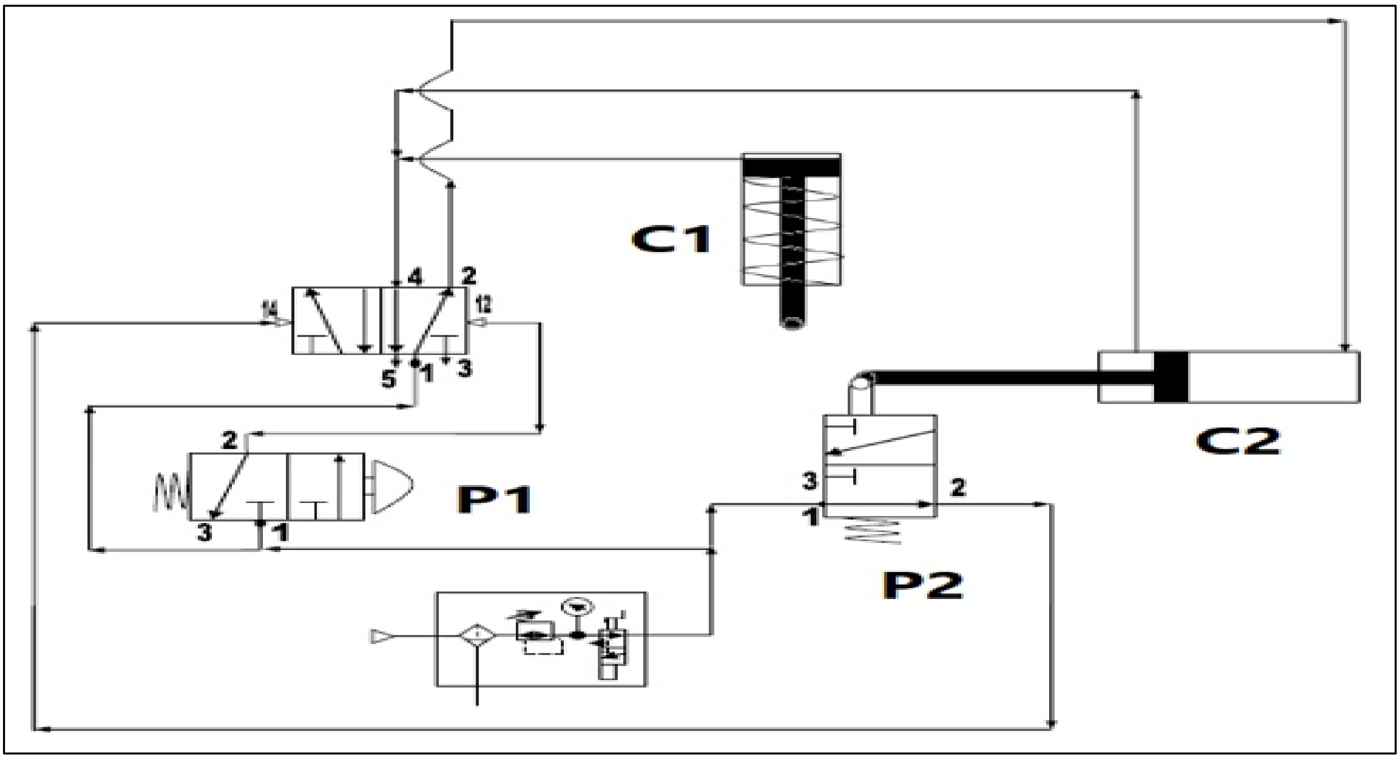

Fig 3 Second Actuated Position

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In all cases, the relation between the motions of the cylinders can represented in the time frame Fig. (4).

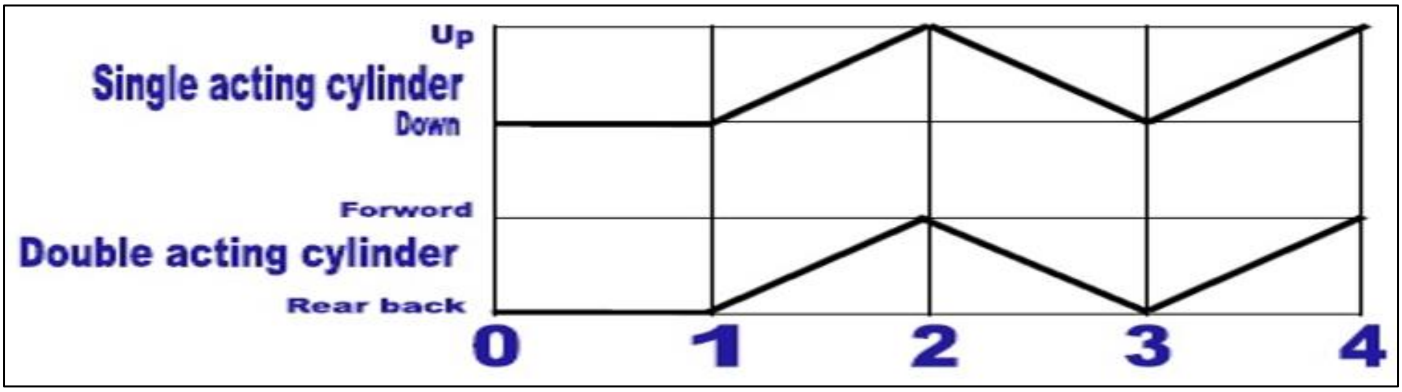

Fig 4 Cylinders Movements Time Frame

III. MODELLING OF SPOOL VALVE-CONTROLLED ACTUATORS

In the VCA Fig. (5), the maximum displacement of the spool of the DCV relative to the valve body is limited by mechanical position limiters, normally within ±1 mm. If greater deflection is required, the control rod should be moved continuously by applying the required force until the spool reaches the desired position. During this period, the control valve is fully open, the spool is continuously displaced and the piston rod follows this displacement. It is therefore necessary to investigate the behavior of the system under these conditions. In the illustration shown, a symmetrical pneumatic cylinder is controlled by an ideal and zero-lapping 4/3 DCV. The flow characteristics of this system are analyzed in steady-state operation, taking into account the following assumptions:

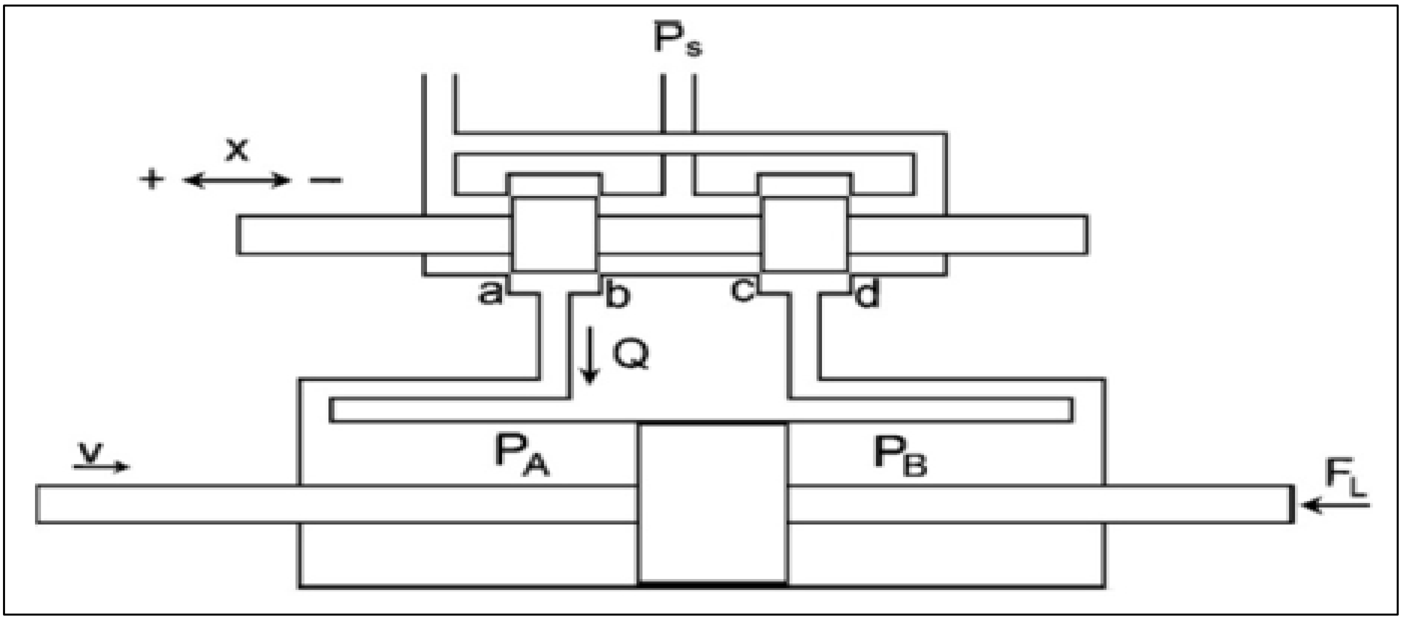

Fig 5 Valve Controlled Actuator, VCA

- The DCV is a matched symmetric valve. This means,
- $A_a(x) = A_c(x),\ A_b(x) = A_d(x),\ A_a(-x) = A_b(x)$ and $A_c(-x) = A_d(x)$.
- The pneumatic cylinder is ideal; no friction and no leakage.
- The throttling areas of the valve ports are linearly proportional to the spool displacement. (A_x= ωx).
- The spool valve is of the zero-lapping type, with no radial clearance leakage.
- The return line pressure is null: P_{atm} = 0.

The flow rates through the valve restrictions are calculated using the following equations:

$$Q_a = C_d A_a(x)\sqrt{2(P_A - P_{atm})/\rho} \tag{1}$$

$$Q_b = C_d A_b(x)\sqrt{2(P_s - P_A)/\rho} \tag{2}$$

$$Q_c = C_d A_c(x)\sqrt{2(P_s - P_B)/\rho} \tag{3}$$

$$Q_d = C_d A_d(x)\sqrt{2(P_B - P_{atm})/\rho} \tag{4}$$

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In the steady state, for positive spool displacement, the flow rates Q_b and Q_d are equal due to the symmetry of the cylinder, and the areas A_b and A_d are also equal for matched valves.

$$C_d A_b(x)\sqrt{2(P_s - P_A)/\rho} = C_d A_d(x)\sqrt{2(P_B - P_t)/\rho} \quad (5)$$

Then;

$$P_s = P_A + P_B \quad (6)$$

If the load pressure is defined as,

$$P_L = P_A - P_B \quad (7)$$

Then

$$P_A = (P_s + P_L)/2 \quad (8)$$

$$P_B = (P_s - P_L)/2 \quad (9)$$

Also, if the inlet and outlet flow is defined as,

$$Q = Q_b - Q_a = Q_c - Q_d \quad (10)$$

If one then, insert P_A and P_B from equations (8) and (9), the flow results from the following equation:

$$Q = C_d A_b(x)\sqrt{(P_s - P_L)/\rho} - C_d A_a(x)\sqrt{(P_s + P_L)/\rho} \quad (11)$$

In the case of an ideal valve, for positive spool displacement,

$A_c = A_a = 0$, $A_b = A_d = \omega x$ and $Q = Q_b = Q_d$

Then,

$$Q = C_d \omega x \sqrt{(P_s - P_L)/\rho} \quad (12)$$

The maximum flow rate Q_{max} is obtained at x = x_{max} and $P_L = 0$,

$$Q_{max} = C_d \omega x_{max}\sqrt{P_s/\rho} \quad (13)$$

The flow equation is written in dimensionless form by defining the following dimensionless parameters:

$\bar{Q} = Q / Q_{max}$, $\bar{x} = x / x_{max}$ and $\bar{P} = P_L / P_s$

Then,

$$\bar{Q} = \bar{x}\sqrt{1 - \bar{P}} \quad (14)$$

The dimensionless piston velocity is given by $\bar{v}$ = v /v_{max},

Where:

v = Q / A_p and $v_{max} = Q_{max} / A_p$

Then,

$$\bar{v} = v / v_{max} = \bar{Q} = \bar{x}\sqrt{1 - \bar{P}} \quad (15)$$

With a negative piston displacement (see figure below), the piston moves in the same direction as the load force. The flow rate of the valve is determined by the following relationship:

$$\bar{Q} = \bar{x}\sqrt{1 + \bar{P}} \quad (16)$$

In this case, the maximum opening is $\bar{x} = 1$

$$\bar{Q} = \sqrt{1 + \bar{P}} \quad (17)$$

The steady-state flow characteristics of the valve-controlled actuator are plotted in the illustrated Fig. (6).

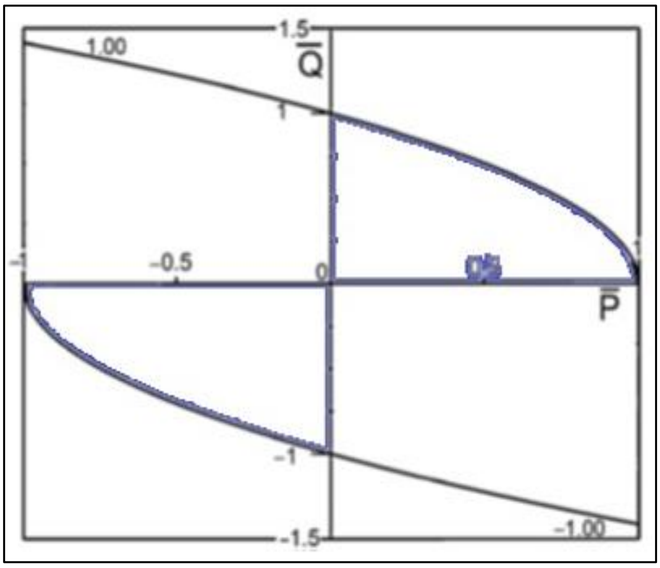

Fig 6 Steady State Flow Characteristics of VCA

The performance characteristic of the valve-controlled actuators describes the relationship between the output power, the spool movement, and the load pressure. The output power can be obtained by the following equation:

$$N = F_r v = P_L A_P \frac{Q}{A_P} P_L Q \quad (18)$$

Or

$$N = C_d \omega x P_L \sqrt{(P_s - P_L)/\rho} \quad (19)$$

The output power is zero when either x or P_L becomes zero or $P_L = P_s$. The maximum output power is reached at x = x_{max} and load pressure P_L in the range of $0 < P_L < P_s$. Thus the maximum power is driven as:

$$\frac{\delta N}{\delta P_L} = 0 \quad (20)$$

$$C_d \omega x\sqrt{(P_s - P_L)/\rho} - \frac{1}{2\rho} C_d \omega x P_L \left\{\frac{1}{\rho}(P_s - P_L)\right\}^{\frac{-1}{2}} = 0 \quad (21)$$

ISSN No:-2456-2165 https://doi.org/10.38124/ijisrt/25oct642

Then,

$P_L = \frac{2}{3} P_s$ and $\bar{P} = \frac{2}{3}$

$$N_{max} = \frac{2}{3\sqrt{3}} C_d \omega x_{max} P_s \sqrt{\frac{1}{\rho} P_s} \quad (22)$$

If the dimensionless power $\bar{N}$ is described as $\bar{N}$ = N / N_{max} then,

$$\bar{N} = \frac{3\sqrt{3}}{2} \bar{x} \bar{P} \sqrt{1 - \bar{P}} \quad (23)$$

In this case, since the maximum opening is $\bar{x} = 1$

$$\bar{N} = \frac{3\sqrt{3}}{2} \bar{P} \sqrt{1 - \bar{P}} \quad (24)$$

The power characteristics of a valve-controlled actuator are plotted as shown in Fig. (7).

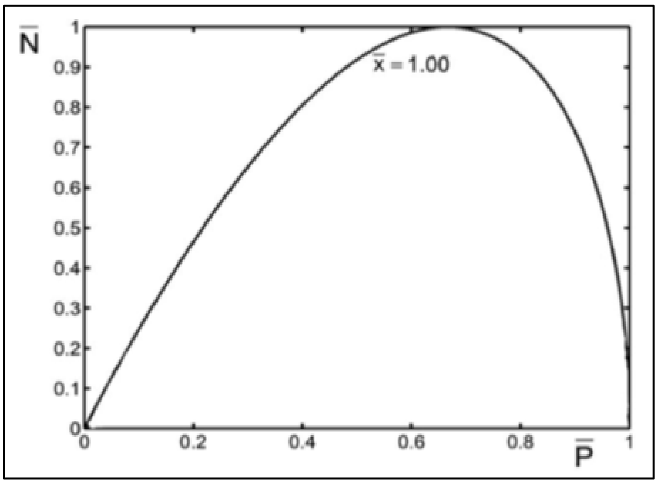

Fig 7 Steady State Power Characteristics of VCA

IV. FABRICATION OF PUNCHING MACHINE

The start of production process always begins with preliminary calculations of the most important design parameters, such as the following.

According to Stacy, C. [7], the maximum operating load pressure of this machine is considered 10 bar.

$$V = \frac{L}{T} = \frac{0.1}{0.1} = 1 \quad m/s$$

$$A_P = \frac{\pi\, d^2}{4} = \frac{\pi \times 0.025^2}{4} = 0.00049 \quad m^2$$

$$Q = A_P \times V = 0.00049 \times 1 = 0.00049\, m^3/s$$

$$F_{max} = P_{max} \times A = 10 \times 10^5 \times 0.00049 = 490\, N$$

Since the permissible shear stress $[\tau] = 61.32 \quad N/mm^2$ for bolt and the force acting on each bolt is given by; $\frac{490}{4} = 122.5 \quad N$

To ensure that the force acting on the bolt is safe or not,

$$F_s = (2n_2 + n_1) \frac{\pi d^2 S_s}{4} [\tau]$$

Where $n_1 = 4$ and $n_2 = 0$

$$F_s = \pi \times 10^2 \times 61.32 = 19264.25 \quad N$$

Then

$$F_{max} < F_s$$

So, the force acting on the bolt is safe. The reduced load is given by:

$$F_r = W_L + W_P + W_d$$

$$F_r = 9.81 + 7.142 + 4.905 = 21.857\, N$$

$$P_L = \frac{F_r}{A_P} = \frac{21.857}{0.00049} = 44606.122 \quad N/m^2$$

Then

$$\bar{P} = \frac{44606.122}{6 \times 10^5} = 0.074$$

$$\bar{N} = \frac{3\sqrt{3}}{2} \bar{P} \sqrt{1 - \bar{P}}$$

Then

$$\bar{N} = \frac{3\sqrt{3}}{2} 0.074 \sqrt{1 - 0.074} = 0.185$$

$$\bar{Q} = \sqrt{1 + \bar{P}} = \sqrt{1 + 0.074} \sim 1$$

Finally, the relation between operating relative power and load pressure in the machine is illustrated in Fig. (8).

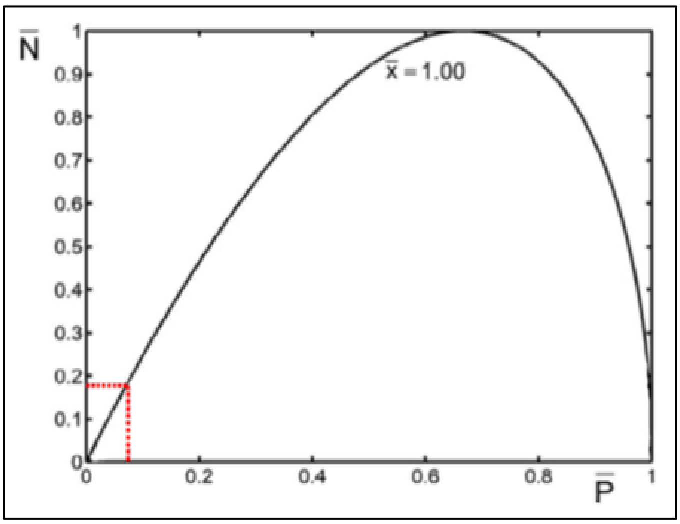

Fig 8 The Relation Between Relative Power and Pressure in Machine

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As soon as the calculation showed that it was feasible, production began. Firstly, the cutting, shaping, and planning work was carried out. The base was cut using hydraulic shears with a cutting capacity of 6 mm as the maximum thickness Fig. (9). Additional parts were also cut for the support of the vertical cylinder and the mounting of the vertical and horizontal cylinders. The final part was formed using a manual sheet metal bending machine with a maximum thickness of 1.5 mm.

Fig 9 Hydraulic Power Shear Machine

The drilling and milling processes were then started as shown in Fig. (10). First of all, the drilling process is carried out using the new drilling machine to drill the required holes in the base, the bracket, and the vertical supports. It is necessary to attach the used DCVs and hold the needed cylinders. The milling process is done in series operation to make the surface finish of the individual parts. It was done using an updated milling machine.

Fig 10 Drilling and Milling Machines

The shaping machine is used finally. This machine aims to shape the rods into the desired square and L-shape. The bending of the vertical support was done by heating and hammering. Finally, the machine parts became ready. Finally, the assembly process is done by using the necessary bolts and nuts to get the required machine Fig. (11).

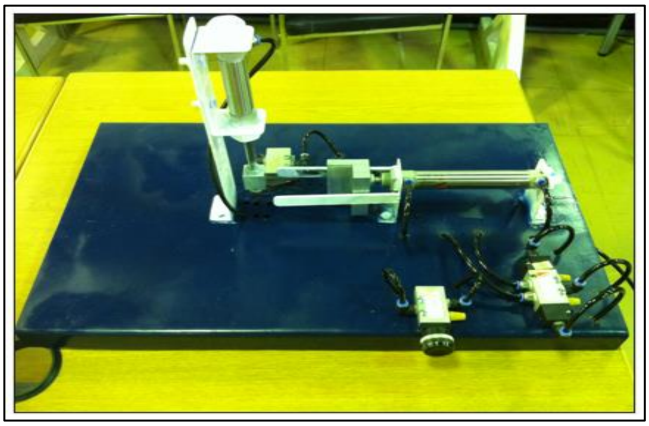

Fig 11 Pneumatic Punching Machine

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V. CONCLUSION

Pneumatic systems outperform hydraulic and mechanical systems in maintenance, cost, accuracy, and productivity. Based on calculations, the project model will be designed to operate at a maximum pressure of 10 bar. Pneumatic punching machines, which use compressed air for the punching process, offer a good balance of speed, cost efficiency, and ease of operation compared to other punching methods. They are particularly well-suited for high-speed die-cutting applications, especially in assembly lines, and can also be cost-effective for small operations. They offer faster cycle times, making them ideal for mass production and assembly lines. However, the choice between pneumatic and other methods, such as hydraulic punching, depends on the application and the level of precision and force required. Finally, pneumatic systems offer advantages such as a good power-to-weight ratio, low cost, easy maintenance, safety, versatility, high accuracy, repeatability, and clean, dry operation. This research paper highlights the advantages of using a pneumatic punching machine, including its cost-effectiveness in large-scale production and the use of compressed air as a power source. By comparing it to hydraulic punching machines, this study contributes to the understanding of the benefits and suitability of pneumatic systems for punching applications.

- *Nomenclature and Units*

Table 1 Nomenclature and Units

OFN	**Oxygen Free Nitrogen**	
DCV	Directional Control Valve	
L	Stroke Length of Cylinder	(m)
T	Duration Time for Extension	(s)
VCA	Valve Controlled Actuator	
Q	Flow rate	(m^3/s)
Q_a	Flow rate at port A	(m^3/s)
Q_b	Flow rate at port B	(m^3/s)
Q_c	Flow rate at port C	(m^3/s)
Q_d	Flow rate at port D	(m^3/s)
Q_{max}	Maximum flow rate	(m^3/s)
$\bar{Q}$	Relative flow rate	
C_d	Discharge coefficient	
A_P	Area of the piston	(m^2)
A_A	The area at Port A	(m^2)
A_B	The area at Port B	(m^2)
A_C	The area at Port C	(m^2)
A_d	The area at Port D	(m^2)
P	Applied system pressure	(N/m^2)
P_{atm}	Atmospheric pressure	(N/m^2)
P_a	Pressure at port A	(N/m^2)
P_b	Pressure at port B	(N/m^2)
P_L	Load pressure	(N/m^2)
P_S	System pressure	(N/m^2)
$\bar{P}$	Relative pressure	
x	Spool displacement	(m)
x_{max}	Maximum displacement	(m)
$\bar{x}$	Relative displacement	
v	Volume	(m^3)
v_{max}	Maximum volume	(m^3)
$\bar{v}$	Relative volume	
N	Power	(W)
N_{max}	Maximum power	(W)
$\bar{N}$	Relative power	
ρ	Density	(kg/m^3)
V	Velocity of the cylinder	(m/s)
F_{max}	Maximum load	(N)
F_r	Reduced load	(N)
F_s	Allowable load	(N)
$[\tau]$	Allowable shear stress	(N/m^2)
n_1	Number bolts under single shear	

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n_2	Number bolts under double shear	
W_L	Weight of the system load	(N)
W_P	Wight of the piston	(N)
W_d	Wight of the die (L – shape)	(N)
ω	Constant of proportionality	

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