FABRICATION OF PNEUMATIC IMPACT DEVICE

Rajneesh Kumar, Shivangi Dixit,Aditya Veer Gautam, Anurag Kumar
Faculty of Engineering and Technology, Rama University
rajkum2288@gmail.com

ABSTRACT

A Pneumatic Impact Machine offers the functionality of larger, more expensive machines such as Fly presses, Kick presses, Drop Stamps, and Crank presses at a fraction of the cost. It serves as a powerful unit that can handle a vast array of press work applications. Many tasks benefit from its impact-based operation, which can lead to cleaner cuts and better grain structure in forging due to the rapid application of energy. Furthermore, a key benefit of the Pneumatic Impact Machine is its smaller size compared to the previously mentioned equipment.

The design and working principle of an Impact Machine are straightforward. At the heart of the machine lies the Impact Cylinder, this is notable for its simplicity and reliability, ensuring longevity. This cylinder mainly consists of a single moving part, the piston rod assembly. It is connected to a basic control circuit for ease of operation and can be powered using standard factory or garage air supplies. Typically, the machine is compact enough to fit on most wooden workbenches. For specialized applications, the Impact Machine can be equipped with pneumatic, PLC, or electronic interlocking systems, along with automatic feeding and ejection mechanisms.

DOI Number: 10.48047/nq.2021.19.5.NQ21092 **NeuroQuantology 2021; 19(5): 404-409**

Background

Impact Machines differ significantly from traditional Fly Presses and Arbor Presses. In an Impact Machine, the force applied to the work piece is not reliant on the pressure exerted by the operator when pulling the lever. With a mechanical advantage ratio exceeding 175, the machine delivers consistent pressure on the work piece regardless of how gently the lever is pulled. This ensures immediate shaping of the material to the desired specifications and a uniform finish. These machines are exceptionally suited for a variety of tasks including riveting, numbering, bending, flaring, embossing, punching, shearing, stamping, button-fixing, eyeleting, corner-fitting, stud-fitting, hot-stamping, and assembling.

This machine operates on the principles of a pneumatic system. It is ideal for a range of applications such as marking, riveting, and caulking, numbering, dinking, and shearing, where consistent and uniform impact is necessary across different materials. In such machines, air serves as the power source. This is similar to how air powers tools like drills, sanders, and grinders in settings such as garages and auto body shops.

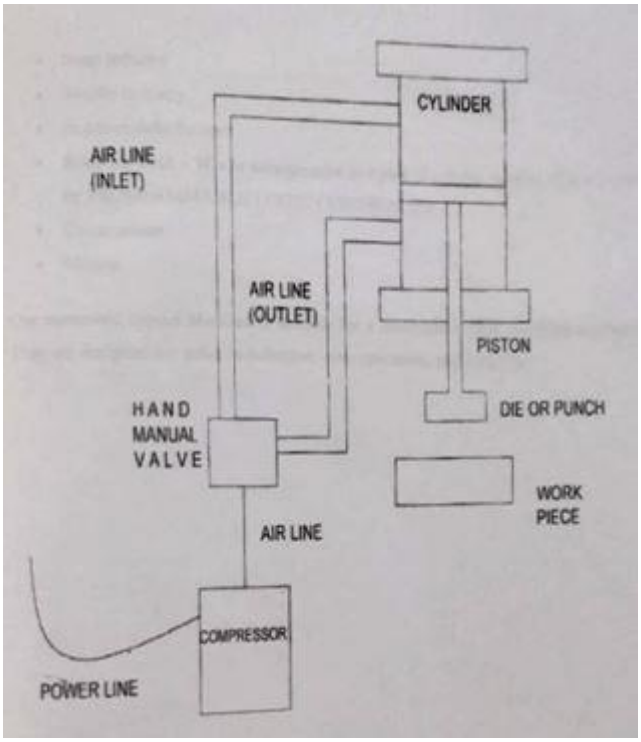

Fig: - Layout of Pneumatic impact machine

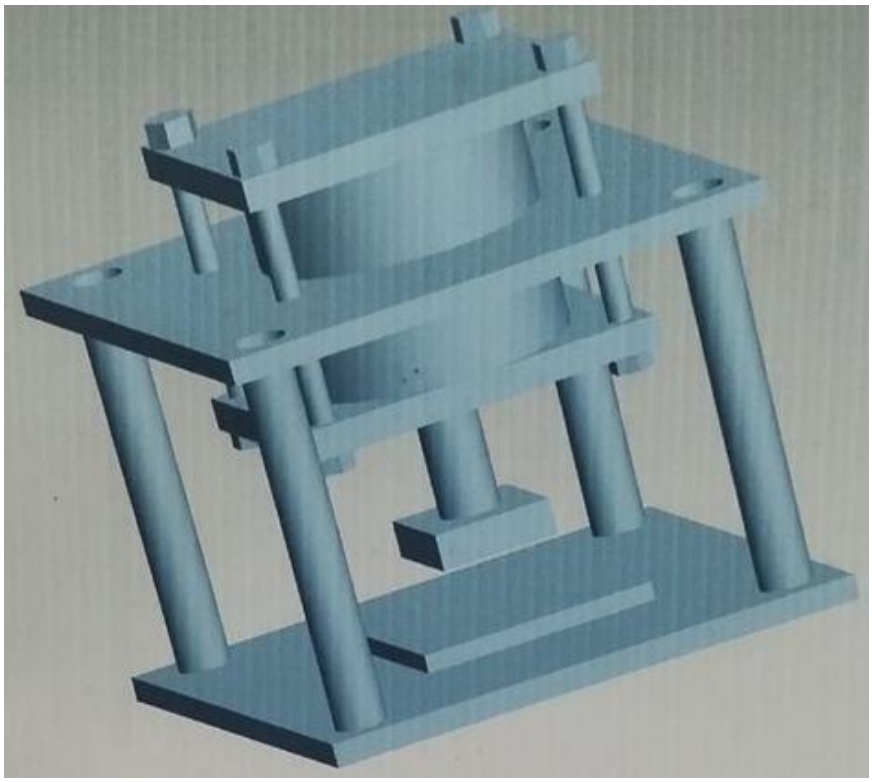

Fig: - CAD MODEL

DESCRIPTION OF MODEL

In our model we have used several parts

1- Compressor
2- Pneumatic cylinder
3- Pressure tube
4- Die
5- Work piece

Dimension of Model

1-	Plate dimension	200×178 mm
2-	Plate thickness	20 mm
3-	Plate drill diameter	13 mm
4-	Cylinder (shaft diameter)	16 mm
5-	Stroke length	60 mm
6-	Die	18.5 (big), 16.7 (small), 6 mm thread
7-	Distance between two plates	140 mm
8-	Height of pillar	140 mm
9-	Pillar	8 mm thread
10-	Pillar diameter	22.8 mm
11-	Pressure tube	8/10 mm

For making of die we have used the pantograph mechanism in which 1: 2 ratios should be taken.

DESIGN OF PRESSURE TUBE

Pressure developed by compressor =10 Kg/cm²

=.98 N/mm² = 1N/mm² (approx)

Calculation of mass flow in tube

Volume of cylinder = $(\pi/4)\, d^2 L$ here d = diameter of cylinder

$= 3.14 \times 50^2 \times 75$ L = length of cylinder

$= 147262.155\ mm^3$

Time taken by piston (upstroke + downstroke) = 2 sec

Total mass of air in cylinder = density of air × Volume of cylinder

$= 1 \times .000147262.155\ m^3$

Mass flow rate in cylinder $= .000147262.155\ m^3 / 2$

$= .00073631\ m^3/sec$

Because Mass flow rate in cylinder = Mass flow rate in Pressure tube

Hence Mass flow rate in Pressure tube = $.00073631\ m^3/sec$

Calculation of Velocity of flow in Tube

Hence $m° = A1V1 = A2V2$ here A1= area of pressure tube

A2 = area of Cylinder

V1 = Velocity of flow in pressure tube

V2 = Velocity of flow in Cylinder

$m° = A2V2$

$= \pi \times DL \times V2$

$V2 = .00073631 \times 10^6 / \pi \times 50 \times 75$

= .0062495 m/sec

A1V1 = A2V2

$\pi \times (d2^2 - d1^2) \times V1 = \pi \times DL \times V2$

$V1 = \pi \times 50 \times 75 \times 0.6249 / \pi \times 36$

= 2.6 m/sec

Hence

Pressure tubes carries quantity of gas (Q) =.00073631 m^3/sec

Velocity of flow (v) = 2.6 m/sec

From design Data Book,

Allowable tensile stress for polyurethane tube (σ) = 20 N/mm^2

Inside diameter of tube,

$D = 1.13\sqrt{(Q/v)}$

$D = 1.13 \sqrt{(0.00073631/ 2.6)}$

D = 6.013 mm

Wall thickness of tube:

$t = P.D / 2\sigma$

$t = 1 \times 6.013 / 2 \times 20$

t =0.1002 mm

Here t = Wall thickness of tube
P = pressure tube
D = Inside diameter of tube
σ = Allowable tensile stress

Outer diameter of tube = 6.013 + 0.1002

= 6.1132 mm

For design of tube:

1. D/t = 6.013 / .1002
   = 60 > 20
2. σ/p = 20/1

= 20 > 6

Both conditions are satisfied for designing of the tube.

Stress in pressure tube

Stress in the pressure tube due to internal pressure along the tangential axis

$\sigma t = p.r_i^2 \{ 1 + r_o^2/x^2 \}$

$r_o^2 - r_i^2$ here r_i = inner radius of tube

r_o = Outer radius of tube

$20 = \mathbf{1} \times 3.0065^2 \{ 1 + 3.0566^2 / x^2 \} / 3.0566^2 - 3.0065^2$

X = 5.3 mm

Therefore thistangential stress (hoop stress) is maximum inner radius of pressure tube & minimum at outer radius of tube.

Radial stress (inner radius of tube)

$\sigma r = p.r_i^2 \{ 1 - r_o^2/x^2 \} / r_o^2 - r_i^2$

$= \mathbf{1} \times 3.0065^2 \{ 1 - 3.0566^2 / 3.0065^2 \} / 3.0566^2 - 3.0065^2$

$\sigma r = -1 N/mm^2$

For outer radius of tube (σr)= 0 N/mm^2

Therefore thisradial stress is maximum inner radius of pressure tube & zero at outer radius of tube.

DESIGN OF CYLINDER

Diameter of cylinder

Pressure developed in cylinder=1 N/mm²
Since an allowance of 10% of operating force F is provided for friction in the cylinder.
Assuming 1.68 KN load is applied by piston.
Therefore total force (F) = 10%F+F

(F) = 1.1F

(F) =1.1×1.68 = 1.848 KN …… (Eq. 1)

But $F_1= (\pi/4) \times D^2P_1$
[Here P_1= compressor supply pressure-pressure drop in transmission]
Therefore P_1= 1-0.06=0.94 N/ mm²
So $F_1= (\pi/4) \times D^2\times 0.94$ …… (Eq. 2)
From equating Eq. 1& 2
So
$(\pi/4) \times D^2\times 0.94= 1.848 \times 1000$
$D^2= (1.848\times 4\times 1000)/\pi\times 0.94$
D= 50.03 mm = 50 mm (approx.)

Thickness of Cylinder

$t= (P\times D) /2\times \sigma_t$
Here σ_t = allowance stress
From Design Data Book
σ_t for cast iron =30 N/mm²
So
t= (0.94×50)/2×30 = 0.78mm
Thickness (t) = 0.78 =3.00mm (approx.)
So D/t = 184/0.78 = 64.10 > 20.
σ_t/P = 30/0.94 = 31.91 > 6.
Therefore both conditions are satisfied for thin cylinder.
So our design of cylinder is thin cylinder type.

Power Output of Cylinder

Suppose stroke of piston (L) = 60 mm
Both side clearance = 7.5mm
So total clearance in cylinder = 15mm
Total length of cylinder = 75 mm
Time required for working stroke = 2 second
Distance moved by piston per second = 60/(2×1000)
Work Done per Second = Force × Distance moved per second
W = 1.848 × 30
W = 55.44 N× m/s
Therefore power output of cylinder = 55.44 N× m/s

IMPACT STRESS

Area of die = 4.4×2.35 = 10.34 cm^2

Stress = force/ area = $1845.68\times 10^2 \times 10^2/10.34$

$= 178.79\ N/cm^2$

If stress is more than this value then die may be failed.

APPLICATION OF OUR MODEL

1. Soap industry
2. Textile industry
3. Automobile sector
4. Robotics field
5. Construction
6. Mining