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# Design and Development of Combined Mechanical and Pneumatic Valve Spring Remover

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***Abstract:** A combined mechanical and pneumatic valve spring remover is used to remove or install valve springs from an engine while the cylinder head is mounted on the engine or supported on a workbench (off the head). The designed tool was adjustable and mounted over the cylinder head by a stud. It can operate manually to depress the spring or by using an air compressor for a pneumatic actuator. The model of the tools is done by SOLIDWORK 2021 and we selected the stainless steel AISI grade-304(12) material. Tools were made following the calculations that have been done on valve springs. From the data calculation, the force required to depress the valve spring is 1350 N but the tools were designed to produce 1962.5 N by considering some friction loss. The analysis of the pneumatic system done by ANSYS software shows acceptable results like, Von Mises stress (20.036 MPa), maximum principal stress (26.426 MPa), and the total deformation becomes 1.6029×10-3. The prototype was built, and the test was carried out in time efficiently during the removal or installation of the valve spring on Engine YaMZ-238. The test result shows that to remove 4 springs, the designed tool is very effective to use by optimizing the difference in time savings of an average of 1 minute and 17 seconds when removing the spring valve, and 1 minute and 26 seconds when installing the valve, even when we use it mechanically. The developed valve spring remover has several advantages over the standard C-clamp tools, such as greater precision and accuracy in removing valve springs, reduced labor and time required for the task, and Increased safety with minimal human force.*

***Keywords:** Combined Valve Spring Remover, Prototype, Engine Yamz-238, and Pressure Piece.*

## 1. INTRODUCTION

Automotive service requires a wide range of excellent work. The work done must meet the needs of the customers in terms of time, money, and effort. Working in a commercial automotive shop or automotive service centre needs tools and equipment that are very useful. If we have performed any kind of repair on our vehicle's cylinder head or camshaft that involves taking-off or replacing the valve springs, we will know that compressing these springs is not that easy to pull off. There are several faulty methods out there that many people use to do this, which have resulted in damaging their valve springs. Most mechanics might have struggled with this too.

### 1.1 Cylinder Head

The cylinder head is a crucial part of an engine, limiting the combustion chamber and protecting the top of the engine block. Its structure is strong and rigid, and undergoes deep thermal cycle shock testing for durability. The cylinder head serves various functions, including the combustion chamber, valve mechanism placement, spark plug installation, inlet and exhaust installation, and water jacket coating. The valve train is essential for proper engine operation, and proper matching of valves, retainers, keepers, springs, and rocker arms ensures long-lasting operation and optimal performance [1].

### 1.2. Cylinder Head Valve Spring

A Cylinder head valve spring is a crucial component in an internal-combustion engine, holding closed valves in the cylinder head. Its primary function is to build engine compression and maintain specific pressure on all moving parts. Valve springs function nearly 70,000 times in an hour and trillion times in their life, ensuring consistent application of spring pressure to prevent valve bounce, which can lead to engine failure, power loss, and even valve breakage. Continuous use wears every part, making valve springs essential for maintaining engine performance [2]. Valve springs come in single and dual designs, with the former supporting the valve independently and the latter having a smaller spring. The right spring pressure is crucial for fast retraction, valve float prevention, and cam wear [3].

### 1.3. Function of Valve Spring Remover

Valve spring remover tools enable mechanics to replace or change stem seals while a vehicle cylinder head is still in the engine. They compress the valve spring, allowing mechanics to remove keepers on a retainer and groove on the top of the valve stem. The main function of a valve spring remover is to control the entire valve train by applying the right amount of spring pressure to prevent valve bouncing [4].

## 2. RELATED WORKS

Automotive workshops often struggle with the valve removal process during engine rebuilding due to inadequate valve spring removers. Existing tools are prone to issues such as lock keys falling

 

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into oil ducts, causing potential injuries or loss of parts, and are generally time-consuming, damaging to valve components, and uncomfortable to use.

The project aims to design and develop an innovative combined mechanical and pneumatic valve spring remover for medium and light-duty vehicles. The CMPSR combines mechanical advantage and pneumatic power, allowing for controlled and consistent compression of springs. This results in swift and efficient removal without compromising engine integrity. The project aims to minimize downtime during repairs, optimize workflow in automotive workshops, and reduce labor intensity associated with valve spring maintenance. Traditional methods often involve manual labor and specialized tools, which can be cumbersome and time-consuming. The project emphasizes the importance of innovation in enhancing efficiency and reliability in engine maintenance tasks.

The literature review discusses various studies and innovations related to valve spring removers. Key findings include Effendi's 2021 valve spring compressor for CAT Diesel engines [5]. Simons et al.'s poppet compressor [6]. Thesius S. Sillero's portable hydraulic compressor [7]. Burlian and Liwaldo's lever-operated tool [8]. Pecaso's valve spring compressor for I-Head engines, and [9]. Swapnil et al.'s dual-valve spring compressor [10]. Overall, these studies demonstrate advancements in valve spring compressor design, focusing on efficiency, ease of use, and suitability for various applications.

## 3. METHODOLOGY AND MATERIALS

### 3.1. Design Modelling and Construction

The newly developed valve spring remover can efficiently and safely compress valve springs on any overhead valve engine and camshaft block valves, whether the cylinder head is attached to the engine block or not, and even when the engine is still in the vehicle.

#### 3.1.1. A Brief History of Pneumatics

Pneumatics involves using compressed air to transmit energy and force. An everyday example is inflating a balloon and letting it propel around a room. A pneumatic cylinder converts this pressure energy into motion. A double-acting cylinder uses compressed air alternately applied to the rear and front ports to move the piston back and forth. Airflow is controlled by a directional control valve, which directs the compressed air to the appropriate port for the desired movement

### 3.2. Design of Pneumatic Cylinder

Based on our selection of materials conducted in methodology our selected material for pneumatic cylinder wall is stainless steel AISI grade-304(12).

Table 3-1: Mechanical Property of Cylinder Barrel

| **Material properties** | **Values** |
|---|---|
| Density($kg/m^3$ ) | 8000 |
| Poisson’s ratio | 0.3 |

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| | |
|---|---|
| Tensile strength(MPa) | 515 |
| Elastic modulus(GPa) | 193 |
| Yield strength(MPa) | 205 |
| Elongation(%) | 40 |
| Hardness(HRB) | 88 |

### 3.2.1. Minimum Bore Diameter:

To calculate the force exerted by an air cylinder, use the formula:

$$Ft = P \times Au \quad 3.1$$

The theoretical force accounts for the required force plus friction and any spring force:

$$Ft = Fr + Ff + Fs \quad 3.2$$

Ft = 1350 N + 106.8 N + 2 N = 1456.8 N

The bore of a pneumatic cylinder is the circular chamber where pressurized air acts to generate force. The term "bore diameter" refers to the diameter of this chamber, often simply called the "bore," and is denoted as D for the inner diameter of the cylinder.

$$D = \sqrt{\frac{4\, x Ft}{P\pi}} \quad 3.3$$

$$D = \sqrt{\frac{4\ x\ 1456.8}{3.14} x \frac{1N}{mm^2}} = 43.08 \text{ mm}$$

The calculated diameter of the cylinder is 43.08 mm is theoretical size. For our design the factor safety with let take D= 50mm

### 3.2.2. Thickness of Pneumatic Cylinder

The thickness without corrosion allowance for each component of a piping system based on the appropriate design code calculations and code allowable stress that consider pressure, mechanical and structural loadings [11]. Where P is internal pressure, D is the pipe diameter (54 mm), σ is the allowable stress (137.895 MPa), E is the material quality factor (0.85), Y is the wall thickness coefficient (0.2), and C is the corrosion allowance (1 mm/year). Were as our simple formula is:

$$t = \frac{PD}{2\sigma E + Py} + C \quad 3.4$$

$$t = \left(\frac{1.1\, MPa \times 54mm}{(2 \times 137.895\, MPa \times 0.85) + (1.1MPa \times 0.2)}\right) + 1mm = 1.342mm$$

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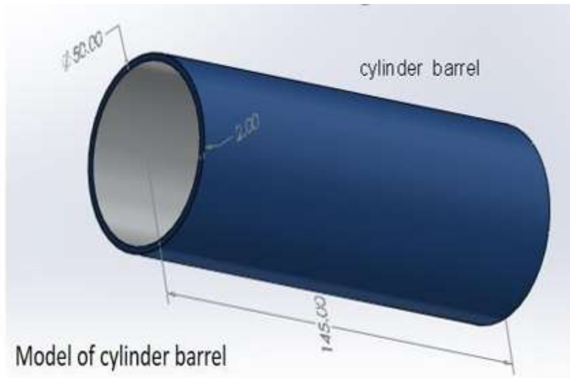


Figure 3-1: Solid Work Model of Cylinder Barrel and Its Cup

Test a range of thicknesses from 0.5t to 1.5t in the physical device to determine which fails. Add a safety margin based on these tests. The theoretical required thickness is 1.8513 mm, but with a safety factor, the design thickness is set to 2 mm.

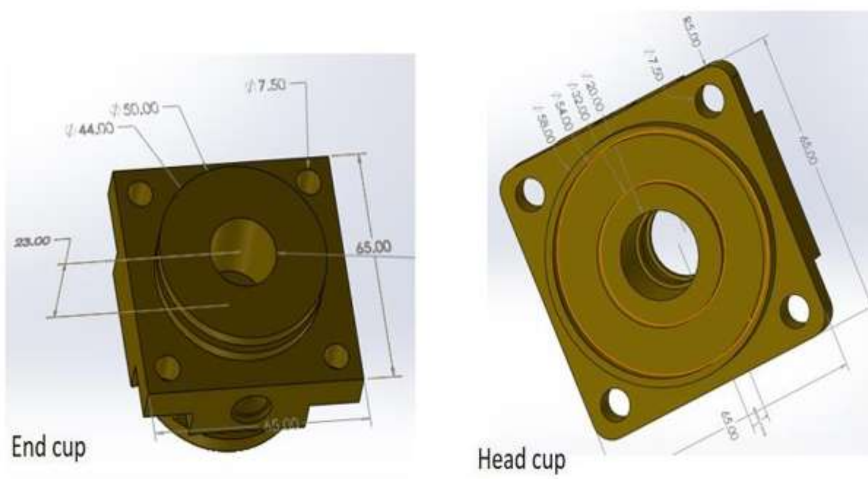


Figure 3-2: Solid Work Model of Cylinder Cup

### 3.2.3 Design of Piston and Piston Rod

3.2.3.1. Piston rod design

In the design calculation, the Young's modulus for stainless steel is 210 GPA. The cylinder force, with a safety factor applied, is 1350 N. The total stroke length is 100 mm (0.1 m). A factor K=0.7K = 0.7K=0.7 is used, with K=0.8K = 0.8K=0.8 chosen to accommodate maintenance and potential adjustments, such as an increase in the stroke length.

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$$I = \frac{\pi D^4}{64} = \quad 3.5$$

$$Pc = \frac{(\pi^2 EI)}{L^2 K^2} \quad 3.6$$

$$Pc = \ (\pi^2 * 190.10^9 * 3.2169 * 10^9)/0.195^2 * 0.8^2) = 113.4NK$$

The moment of inertia and the maximum permissible stress to avoid buckling is dependent on the type of end fixing of the cylinder. Hence diameter of the piston rod required, d = 16mm. Finally, for other calculations and in a case of construction, the diameter of the piston rod used will be 16mm, safety.

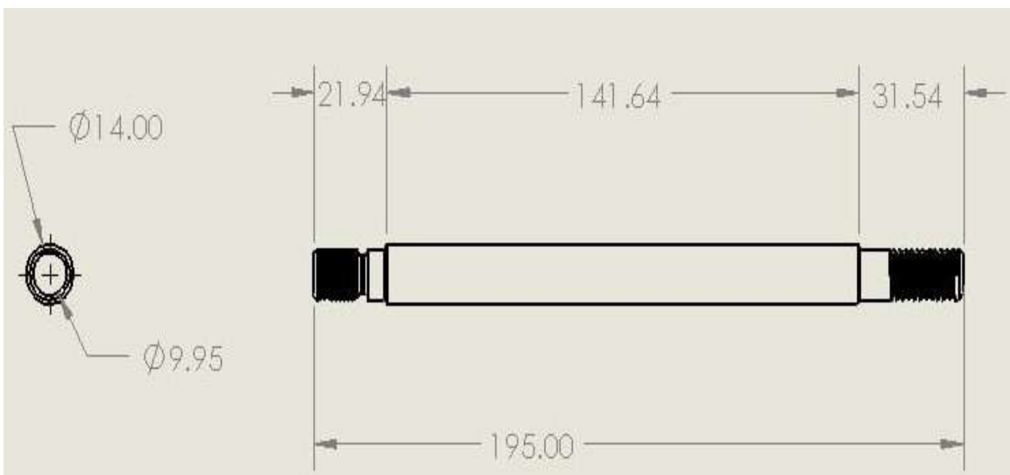


Figure 3-3: piston rod

### 3.2.3.1. Design of the Piston

In a double-acting pneumatic cylinder, pressure applies to both sides of the rod, and the force produced is determined by the piston's full area and the rod's cross-sectional area.

$$F \ = \ P\ (A \ - \ a) \quad 3.7$$

Since the piston and the piston rod are circular in nature, therefore area of the pressurized part is given by

$$A - a = \frac{\pi(D^2 - d^2)}{4} \quad 3.8$$

**Where:**
F = force = assume force * factor of safety (3) = 1350N
P = pressure, 10 bar =100,000
D = diameter of piston
d = diameter of piston rod, 16mm

 

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By substituting the above value into those equations, we have, hence diameter of piston required D = 47 mm. Finally, for other calculations and construction, the diameter of the piston is taken to be 50mm.

### 3.3. Design of Mechanical Part of Special Service Tools

#### 3.3.1. Design of Stand and Base

The design of stand of tools typically aligns axial loads with the column's axis, but some applications may subject columns to off-center loading, causing more severe bending stress. The distance between the column's axis and the eccentric load line of action is known as eccentricity, indicated by e. For example, if a load P is applied to a column at a distance e from the axis, the column will be subjected to an eccentric load, with the load's line of action being at e. [12]. The constant thickness is denoted as t. and (b = t, B = 2t, H =1.7t, h = 0.3t) Area of cross section A = Bh + Hb= A= 2.3t. hence centroid is:

$$Yc = \frac{\left[\left(H + \frac{h}{2}\right) hB + \frac{H^2 B}{2}\right]}{A} \quad 3.9$$

$y_c$= 1.739t

Area moment of inertia

$$Ixx = bH\left(yc - \frac{H}{2}\right)^2 + \frac{bH^3}{12} + \mathrm{hB}\left(\mathrm{H} + \frac{\mathrm{h}}{2} - yc\right)^2 + \frac{h^3 B}{12} \quad 3.10$$

$$Ixx = 1.9855t^4$$

Minimum section modulus, $Zxx = \frac{Ixx}{yxx}$ 3.11

$Z_{xx}$= 1.1417$t^3$

When maximum load P, axial stress due to the load (σd)

$$\sigma d = \frac{P}{M} \quad 3.12$$

$$\sigma d = \frac{P}{M} = \frac{P}{2.3t^2}$$

bending stress due to the weight of tool,

$$\sigma b = \frac{M}{Z} = \frac{P * e}{Z} \quad 3.13$$

Resultant stress can be combination stress of direct stress and bending stress

$$\sigma r = \sigma d + \sigma b$$

$$\sigma r = \frac{P}{M} + \frac{M}{Z}$$

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$$2.6259\sigma t^3 - 1.1417Pt - 2.3\,Pe = 0$$

From the equivalent equation above, solve for *t*. Once t is determined, substitute it into the equations for b (base) and d (depth). This will allow you to calculate the thickness of the stand. P = 1350N and $\sigma_r$ = 540 MPa /4 = 135Mpa and eccentricity = 100mm

$$2.6259\sigma t^3 - 1.1417Pt - 2.3\,Pe = 0$$
$$2.6259 \times 135Mpa \times t^3 - 1.1417 \times 1350\,N \times t - 2.3 \times 1350\,N \times 100mm = 0$$
$$354.4965t^3 - 1541.295t - 310{,}500 = 0$$
$$t^3 - 4.3478t - 875.8901 = 0$$

By using newton Raphson method the above polynomial equation the value of t = 9.719mm so we can take the value of t 10mm
B = 2t = 20mm
H = 1.7t = 17mm
b = 1t = 10mm
h = 0.3t = 3mm

**The Moment of Inertia of Stand**
The following steps should be followed to find the moment of inertia of the T section.
Cross section area [A] A = Bh + Hb= 230mm$^2$
Mass [m], M = ALρ = m= 0.36 kg
Second moment of area [Ixx]

$$Ixx = bH\left(yc - \frac{H}{2}\right)^2 + \frac{bH^3}{12} + \mathrm{hB}\left(\mathrm{H} + \frac{\mathrm{h}}{2} - yc\right)^2 + \frac{h^3B}{12} = 8573.949\ \mathrm{mm}^4$$

Second moment of area [Iyy], $Iyy = \frac{b^3H}{12} + \frac{B^3h}{12} = 3416.667\mathrm{mm}^4$
Minimum section modulus [Sxx], $Sxx = \frac{Ixx}{yc} = 771.823\ mm3$
Radius of gyration [rx] r = $\left(\frac{I}{A}\right)^{0.5} = 6.106\ mm$
Centroid distance in x direction [xc] = $\frac{B}{2} = 10mm$
Centroid distance in y direction $yc = \left[\left(H + \frac{h}{2}\right)hB + \frac{H^2b}{2}\right] = 11.109$ mm
Cross section Area of stand (A), $A = 25mm \times 8mm = 200mm$

$$Ix = \frac{b^3d}{12} = \frac{25^3 \times 10}{12} = 13{,}020.83mm4$$

Where the Euler load or buckling load is Pcr Young's modulus for stand is E= 190 MPa $I_{min}$ is $Ixx = 2083.3mm^4$

$$Pcr = \frac{\pi^2 EImin}{L} = \frac{(3.14^2 \times 190 \times 2083.33mm^4)}{350} = 11.15KN$$

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To calculate the deflection boom of stand length 10 centimeter which has support at one end only. Assuming Young's modulus of the metal is $190\times10^9$ N/ m². At the end force applied is 1350 N. mass of 0.02kg

Moment of inertia can be $I = \frac{b + d^2}{12} = \frac{20 + 8^3}{12} = 2083.3mm^3$

Now applying the formula

$$df = \frac{WL^3}{3EI} \qquad 3.14$$

$$df = \frac{WL^3}{3EI} = \frac{1350\,N \times (0.1m)^3}{3 \times 200 \times \frac{10^9 N}{m^2} \times 2083.33 \times 10^{-5}\,kg.m2} = 3.25 \times 10^{-5}\text{mm}$$

Therefore, the value of beam deflection will be Approximately 32.5 micrometer so there is no deflection on the boom.

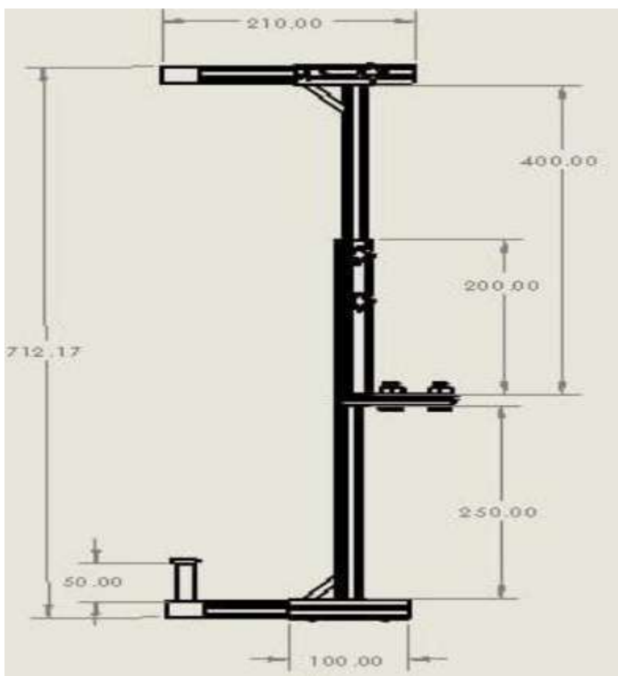

Figure 3-5 Stands and Its Components In 2D Drawing.

### 3.3.2 Designing of Lead Screw and Nut

Lead screws are components used to transmit power or force. They consist of a cylindrical screw with a helical thread that engages with a matching nut. When the screw rotates, it moves axially within the nut, which remains stationary [13].

 

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**Direct Stress:** The axial load (force) is W, compressive in nature and the area which carries the force is the core cross section of diameter $d_1$ Hence, compressive stress

$$\sigma c = \frac{4W}{\pi d1^2} \quad 3.15$$

Additionally, the compressive stress is increased by 30% to account for other stresses.

$$\sigma c = \frac{1.3W}{\pi d1^2} \quad 3.16$$

$$d1 = \sqrt{\left(\frac{4\times 1.3W}{\pi\sigma}\right)} = \sqrt{\frac{4\times 1.3\times 1350N}{\pi\times 42N/mm^2}} = 7.44mm$$

The materials selected for the screw and nut are steel and cast iron, respectively, with properties taken from B.D. Shiwalkar's Design Data Book (Page 94, Table IX-2) [14]. W = 1350 N, μ = 0.15, $\sigma_c$ = 42 N/mm2, $\tau_s$ = 28 N/mm2, $\tau_n$ = 21 N/mm$^2$, $P_b$ = 14 N/mm$^2$

As the screw is subjected to a twisting moment the higher value of core diameter is selected from the table of square thread normal series. dc = 16 mm, Pitch (p) = 4 mm
Major diameter = core diameter + pitch of screw = 16mm + 4mm = 20mm
Mean diameter, $d_m = d_1 + p/2$ = 16mm + 4/2 = 18mm
Consider single start square thread The angle of helix is related to the circumference of mean circle and the pitch

$$tan\alpha = \frac{p}{\pi dm} \quad 3.17$$

$$tan\alpha = \frac{4mm}{\pi \times 18mm} = 4.046$$

= μ = 0.15 β = 8.53070

Torque required to move screw against load

$$T = W\, tan\,(\alpha + \beta) \times \frac{dm}{2} \quad 3.18$$

$$1350N \times tan\,(4.0461 + 8.5307) \times \frac{18mm}{2} = 2{,}710Nmm = 2.71Nm$$

Efficiency of screw, $\dot{\eta} = \frac{tan\alpha}{tan(\alpha+\beta)}$ 3.19

$$\dot{\eta} = \frac{\tan(4.0461}{tan(4.0461 + 8.5307)} \times 100\% = 31.7\%$$

**Design for the Nut:**
Assuming load is uniformly distributed over the cross sectional area of Nut, the bearing pressure (Pb)

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$$\boldsymbol{Pb} = \frac{W}{\frac{\pi}{2}(do-dc)^2 n} \quad 4.20$$

$$n = \frac{1350N}{\frac{\pi}{2}(20-16)^2 14} = 0.43$$

$let\ take\ n = 3\ treads$

For standard square thread the depth or thickness of the thread,

$$t = \frac{\text{pitch}}{2} \quad 4.21$$

$$t = \frac{pitch}{2} = \frac{4mm}{2} = 2mm$$

The total height of nut (h) = n x p = 3 x 4mm = 12 mm
Shear induced in the nut thread (τn ):

$$\tau n = \frac{W}{n \times \pi \times dm \times t} \quad 4.22$$

$$\tau n = \frac{1350N}{3 \times 3.14 \times 18mm \times 2mm} = 3.981 N/mm^2$$

The permissible shear stress for nut is 21 N/mm2 and design value of shear stress for nut is 3.981 $N/mm^2$. Here, design value of shear stress is less than the permissible shear stress i.e. n < τ allowable for nuts so the design of nuts is safe.

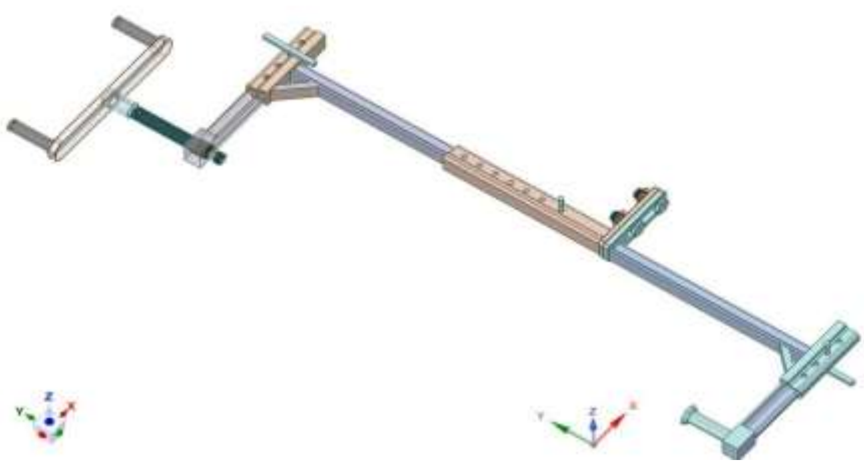

Figure 4-6 Stand with Support Tail IGS Form Model

### 3.4. Construction and Working Principle of CVSR

The combined pneumatic and mechanical valve spring remover features several main components, including a stand, extension rod, pneumatic part, screw part, and pressure pieces. The pneumatic section comprises a cylinder, piston, piston rod, and bush seal, which are essential for operating the pneumatic actuator. The stand, made up of a base, bolt, nut, cylinder holder, and threaded bolt,

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is adjustable and designed with a T-cross section to handle axial loads and bending stresses, supporting and securing the tool on the engine. Additionally, the tool includes a lead screw and handle for mechanical operation and pressure pieces for compression. The pressure piece features a cylindrical hollow magnet housed in a non-ferromagnetic case, allowing for the replacement of the magnetic disc to accommodate different valve tip diameters. The tool's operation involves fastening it to the cylinder head with bolts and nuts. The spring is compressed either pneumatically or mechanically, with the pressure disc unit pressing against the spring retainer. This process moves the retainer away from the cotter piece, which is then removed by a magnet. After compression, the piston is lifted, facilitating the removal of the spring from the cylinder head.

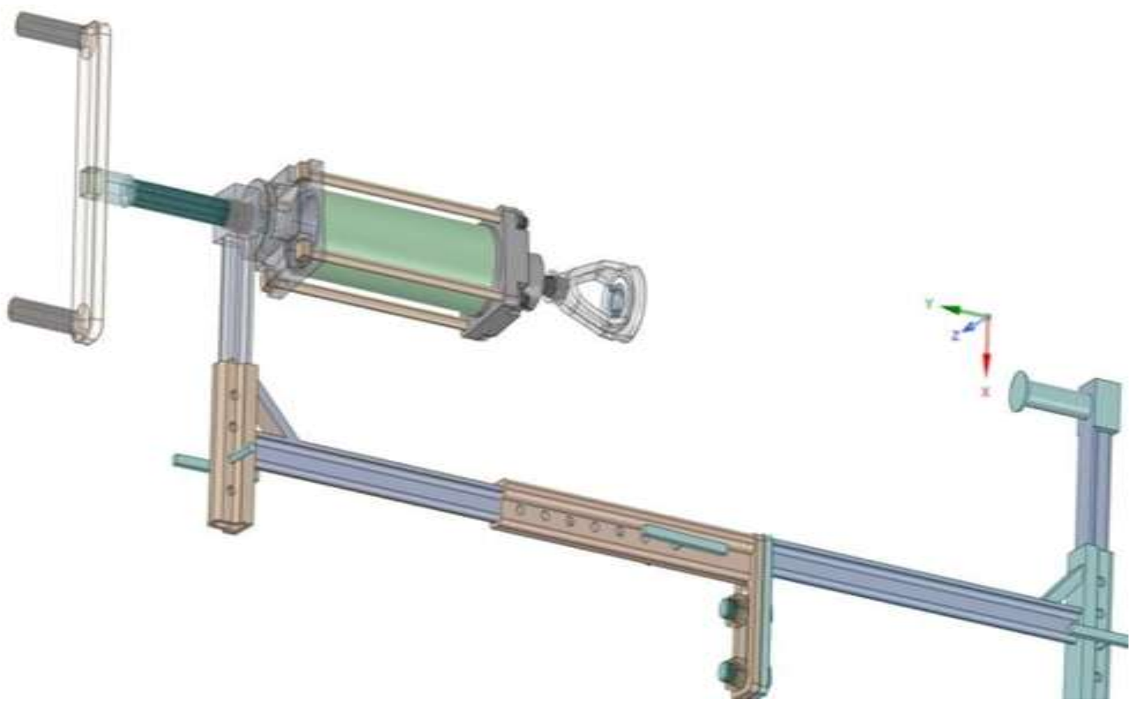

Figure 3-9: Full Assembly of CVSR With Support Tail.

## 4. RESULT AND DISCUSSION

The all the components and parts of the double acting double-end hydraulic cylinder were carefully developed and modelled using SOLIDWORKS workspace. The simulation results for both the stress analysis and deformation will be represented in this section.

### 4.1. Finite Element Analysis (FEA) of the Pneumatic Cylinder

A 3D model of pneumatic cylinder with the following specifications was designed using stainless steel AISI grade-304(12). When the cylinder subjected to an internal pressure of 10 bar, the maximum stress experienced at the ends will be way less than the yield strength of the material.

 

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Table 4-1 designed component specification and component size

| S/N | PARTS DIMENSION | SYMBOL | VALUES | UNIT |
|---|---|---|---|---|
| 1 | Piston rod diameter | D | 16 | mm |
| 2 | Piston diameter | D | 48 | mm |
| 3 | Cylinder Outside diameter | Do | 55 | mm |
| 4 | Cylinder wall thickness | T | 2 | mm |
| 5 | Stroke length | s | 100 | mm |
| 6 | Cylinder port diameter | _ | 10 | mm |
| 7 | Width of cylinder end flange | _ | 65 | mm |
| 8 | Length of cylinder flange | _ | 20 | mm |
| 9 | Tie rod diameter On flange | _ | 7.5 | mm |
| 10 | Flange Edge fillet radius | _ | 5 | mm |
| 11 | Cylinder port diameter | _ | 10 | mm |
| 12 | Length of tie rod | _ | 220 | mm |
| 13 | Required force | | 1350 | N |
| 14 | Pressure | P | 10 | Bar |

#### 4.1.1. Meshing and Boundary Condition

The meshing technique is used to divide the model into a number of small parts for better analytical accuracy. In both cases, meshing was carried out at an element size of 1mm for the cylinder barrel in direct contact with the pressure. The meshing results of cylinder barrel designs are as shown in fig 4.1.

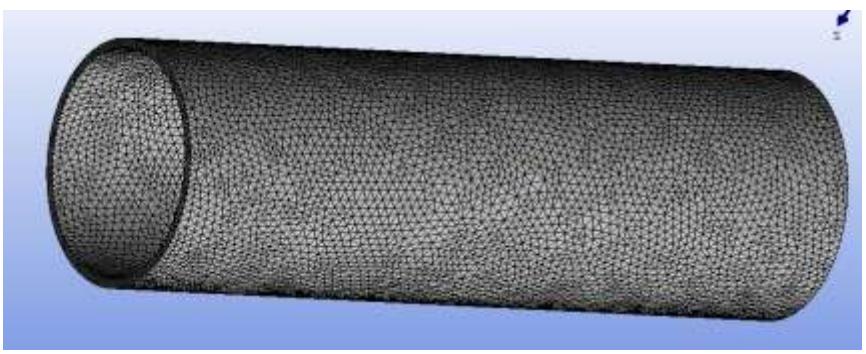

Figure 4-1: Mesh Result of Pneumatic Cylinder Barrel

### 1. Numerical Analysis of Stresses

The hoop stress ($\sigma H$) and longitudinal stress (σL) of a cylinder is calculated using the Barlow formula (Equation 4.1). Here, p1 is the air pressure (10 bar or $10\times10^5$Pa), p2 = External pressure (atmospheric pressure = 101325 pa), $d$o is the outer diameter (54 mm), $d$i is the inner diameter (50 mm), and $r$ is the radius ($d/2$).

 

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$\sigma H \ = \frac{p1(do^2+di^2)}{do^2-di^2}$ 4.1

$$\sigma H = \frac{0.1MPa((54mm)^2 + (50mm)^2)}{\llbracket 54mm)^2 - (50mm)^2} = 13.019\text{MPa}$$

When longitudinal stress is given in equation 5.2.

$\sigma L \ = \frac{(p1ro^2+p2ri^2)}{ro^2-ri^2}$ 4.2

$$\sigma L \ = \frac{0.1MPa(27mm)^2 + 101325(25mm)^2)}{(27mm)^2 - (25mm)^2} = \ \sigma\text{L} \ = \ 6.40\text{MPa}$$

Note that typical maximum allowable stress for stainless steel is below 80 MPa. Therefore, our design is safe. When

## 2. Numerical Analysis of Strain

The circumferential strain (Ɛc) and longitudinal strain (Ɛl) and μ=0.3 and E= 200 x $10^3$ MPa

Circumferential strain (Ɛc), $\quad \varepsilon c \ = \frac{P1 \times di}{4 \times t} \times \frac{2-\mu}{E}$ 4.3

$$\varepsilon c \ = \frac{0.1MPa \ \times \ 50mm}{4 \ \times \ 2mm} \times \frac{2 \ - \ 0.3}{200 \ x \ 10^3 \ MPa} = \ 5.525 \times 10^{-5} \text{ mm}$$

Change in diameter, $\varepsilon c$ ×d = 2.7625×$10^{-3}$mm.

Longitudinal strain, (Ɛl), $\varepsilon l \ = \frac{P1 \times di}{4 \times t} \times \frac{1-2\mu}{E}$ 4.4

$$\varepsilon c \ = \frac{0.1MPa \ \times \ 50mm}{4 \ \times \ 2mm} \times \frac{1 \ - \ 2(0.3)}{200 \ x \ 10^3 \ MPa} = \ 1.25 \times 10^{-5} \text{ mm}$$

Change in length, $\varepsilon l$ ×L = 1.687×$10^{-3}$m

Let's analyze the results from the ANSYS analysis of the pneumatic cylinder barrel with a 2 mm thick wall subjected to 1 MPa internal pressure, focusing on the key parameters: total deformation, equivalent (Von Mises) stress, maximum principal stress, equivalent elastic strain, and maximum principal elastic strain.

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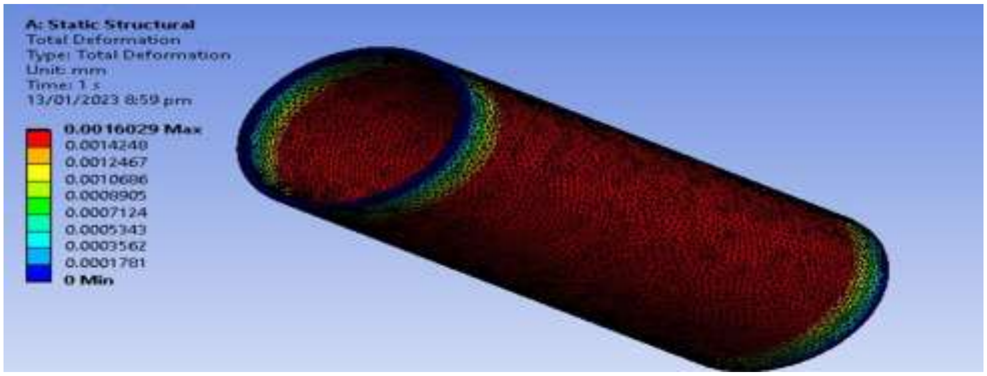


Figure 4-2: Total Deformation Analysis

Deformation: The total deformation of 1.6029 mm indicates the overall displacement of the cylinder barrel due to the internal pressure. It's crucial for assessing dimensional changes and potential interference with other components or operations.

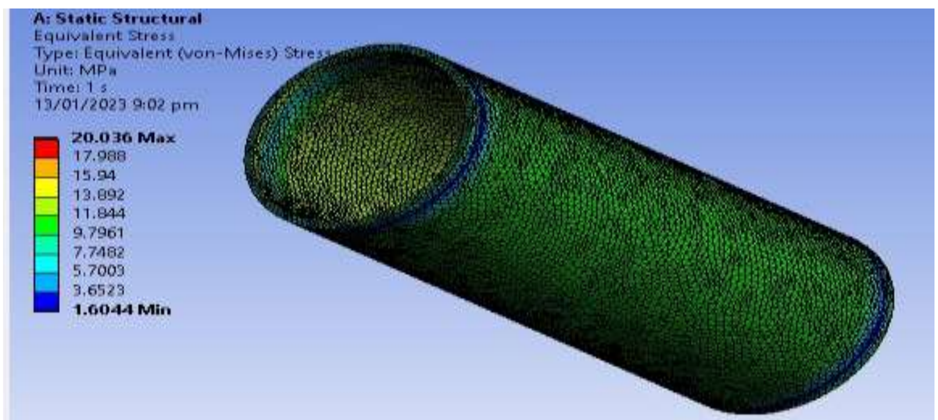


Figure 4-3: Maximum Equivalent Stress Analysis

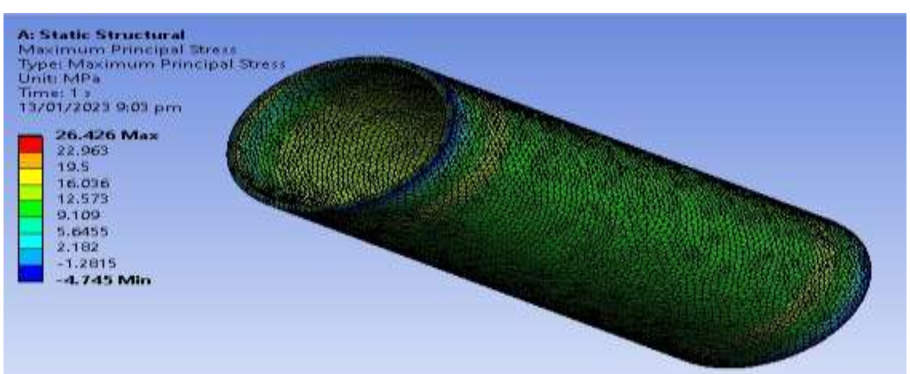


Figure 4-4: Maximum Principal Stress Analysis

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**Stress Analysis:**
The equivalent (Von Mises) stress of 20.036 MPa and maximum principal stress of 26.426 MPa indicate the stress levels within the cylinder barrel. Comparing these stresses to the material's yield strength is essential to ensure structural integrity and prevent plastic deformation or failure.

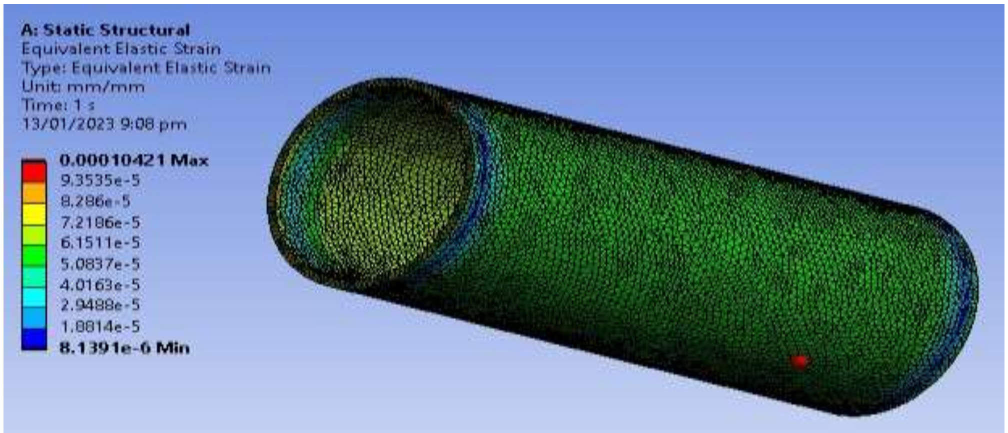

Figure 4-5 Equivalent Elastic Strain Analysis of Cylinder Barrel Under 10bar Internal Pressure.

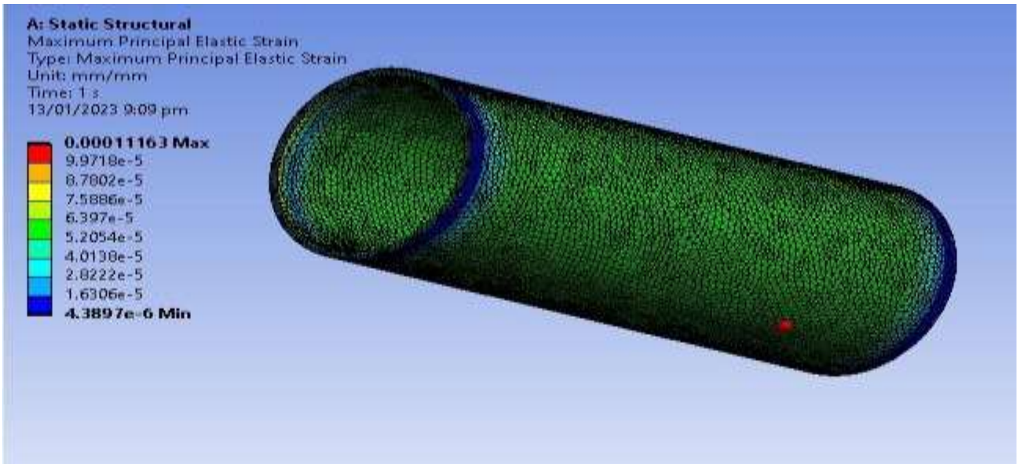

Figure 4-6: Maximum Principal Elastic Strain Analysis of Cylinder Barrel Under 10bar Internal Pressure.

**Strain Analysis:**
The equivalent elastic strain of 1.0421e-004 and maximum principal elastic strain of 1.1163e-004 show the elastic deformation of the material. These strains are within the elastic limit, indicating that the cylinder barrel should return to its original shape once the internal pressure is relieved, because, the material remains within its elastic range.

 

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### Static Structural Analysis of Cylinder Cup

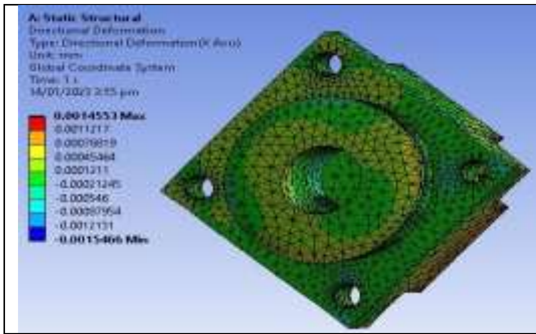


Figure 4-7: cylinder end cap directional deformation

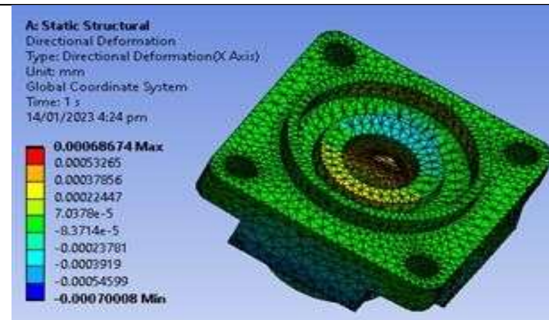


Figure 4-9: cylinder head cap directional deformation

The directional deformation values (0.0014553 mm for the cylinder end cap and 0.000686 mm for the cylinder head cap) are relatively small, indicating that the structural deformations due to the applied pressure are within acceptable limits for typical engineering applications.

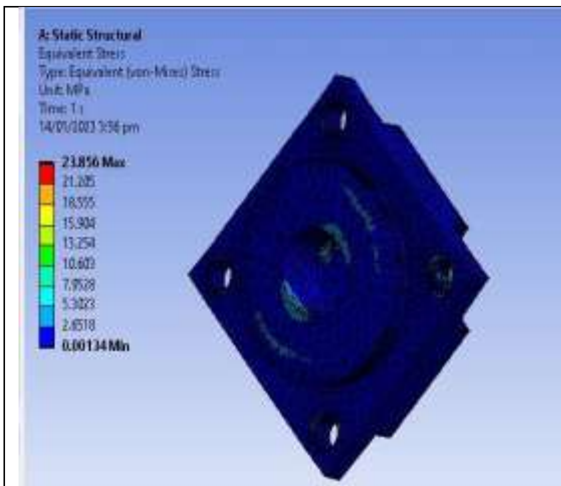


Figure 4-8: cylinder end cap equivalent stress analysis

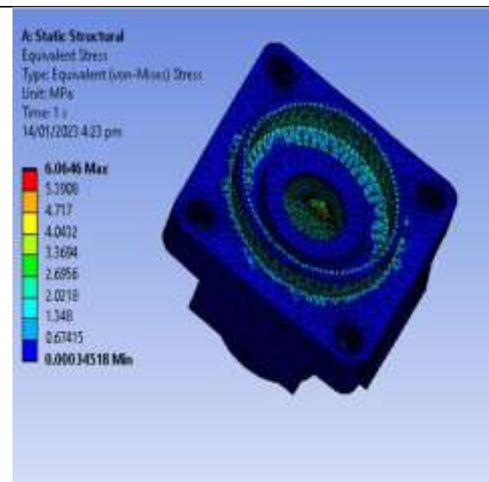


Figure 4-10: cylinder head cap equivalent stress analysis

**Stress Analysis:**
The higher equivalent stress of 23.856 MPa suggests that the cylinder end cap is experiencing more significant stress concentrations compared to the cylinder head cap. Engineers should carefully evaluate stress concentrations to ensure they are within allowable limits to avoid structural failure or plastic deformation. The lower equivalent stress of 6.0646 MPa indicates that the cylinder head cap is experiencing lower stress levels compared to the end cap.

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## 5. Prototype Fabrication

Most of the components are made of materials that exist in this university and some of the materials are purchased from the local market. We used different methods to manufacture this tools, the materials are machined using a milling machine, as well as a turning machine and welded based on the design contained in 2D drawings. Furthermore, it was assembled into a combined Valve Spring remover which is ready for testing.

Figure 5-1: Prototype Model without Tail Support.

### Test Variables and Test Design

The test variable is in the form of data when using the release of all valves. Direct testing on Engine YaMZ-238. The engine which intended for heavy vehicles "Minsk Automobile Plant (MAZ)" and "Ural 4320". both with standard tools and the new design Combined Valve Spring Compressor (CVSC). The test was carried out many times with different people who already had the skills to carry out general overhaul and top overhaul services.

### Result of Test on Head Engine

This result is the experimental result of disassembling and assembling the valve spring without removing the engine head only by doing the protection of valve floating.

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Figure 5-2 Prototype Model during Working in Engine Head Is on

### Removing a Valve Spring Without off the Cylinder Head

To remove valve springs without removing the cylinder head, position the piston at top-dead-center to prevent the valve from falling. Remove the spark plug from the affected cylinder and insert an adaptor into the spark plug hole to supply compressed air. Be cautious with the spark plug as it is fragile. Turn the engine manually using a socket on the front balance bolt until the piston is near bottom-dead-center. Ensure the valves are closed by removing the rocker arms. Tap the valve spring retainer with a hammer to disconnect it from the keeper. Insert a wire or rope into the cylinder through the spark plug hole, and continue rotating the engine manually. Use a valve compressor to compress the valve spring, then remove the keepers and the spring. After completing repairs, rotate the engine back to its original position

### Result of Test off Engine Head

Time efficiency measures the duration required to remove valve springs from each cylinder head. In this study, it was assessed directly using a stopwatch. Four participants performed the task, and the average time from these four trials was calculated for both the assembly and disassembly processes.

### Removing and Replacing a Valve Spring With off Cylinder Head

To remove a valve spring, first, with the cylinder head removed, use the handle at the bottom of the valve spring compressor to open the jaws wide enough to fit over the valve. Adjust the tool if necessary so it aligns with the edges of the spring.

Table 5-2: Table of Result for Disassembling of Valve Spring

| No | Examiner | New CVSC (second) | Standard Tool (seconds) |
|---|---|---|---|
| 1. | Tester 1 | 74 | 93 |
| 2. | Tester 2 | 70 | 88 |
| 3. | Tester 3 | 68 | 89 |

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| | | | |
|---|---|---|---|
| 4. | 4 tester | 67 | 86 |

The disassembly process with the CVSR (Combined Mechanical and Pneumatic Valve Spring Remover) averages 4 minutes and 39 seconds to remove four valve springs, while the C-clamp valve spring compressor takes 5 minutes and 56 seconds for the same task. Therefore, the CVSR tool is more time-efficient than the C-clamp tool.

| No | Examiner | New CVSC (second) | Standard Tool (seconds) |
|---|---|---|---|
| 1. | Tester 1 | 81 | 104 |
| 2. | Tester 2 | 77 | 90 |
| 3. | Tester 3 | 75 | 96 |
| 4. | 4 tester | 72 | 101 |

The assembly process with the CVSR (Combined Mechanical and Pneumatic Valve Spring Remover) takes an average of 5 minutes and 5 seconds to assemble four valve springs, whereas the C-clamp valve spring compressor takes 6 minutes and 31 seconds. This shows that the CVSR tool is more time-efficient compared to the C-clamp tool.

## 6. CONCLUSION AND RECOMMENDATION

**Conclusion**

In this project we designed and developed a combined mechanical and pneumatic valve spring remover that used to remove OHV and a camshaft located in the cylinder block valves easily and safely by Utilising available materials in the local automotive workshop and local market. The model of the tools is done by SOLIDWORK 2021 and we selected the stainless steel AISI grade-304(12) material. Tools were made following the calculations that have been done on valve springs. From the data calculation, the force required to depress the valve spring is 1350 N but the tools were designed to produce 1962.5 N by considering some friction loss. The analysis of the pneumatic system done by ANSYS software shows acceptable results like, Von Mises stress (20.036 MPa), maximum principal stress (26.426 MPa), and the total deformation becomes 1.6029×10-3. The test was carried out in the form of data when using the remove and assemble of valve spring conducted on Engine YaMZ238. By doing a complete overhaul of the combined mechanical and pneumatic compression system in order to increase work effectiveness, reduce the time of operation, minimize the cause of injuries to the technician, and have a more concise and usable way of working even though there is no energy source for the power tool.

**Recommendation**

Suggestions for further development are:
1. Flexible design optimization for all cylinder head shapes and types
2. Analysis of materials and strength and fatigue factors during working operation.

 

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3. Working on compactness and reducing the weight of the tools.

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