Research Article

Dynamic Analysis Of Hydraulic Actuated Mandrel

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ABSTRACT: Machining is an important part of the manufacturing process, as it involves precise metal removal to achieve quality, accuracy, and precision. The proper selection of cutting tools and work holding devices is essential in machining. In the manufacturing process, a work holding device is crucial. A mandrel is a work holding device that is used to grip a component during machining at high speeds and low feeds.

The current work describes the design of a Hydraulic Actuated Mandrel, which is a useful work-holding device that firmly grips the work piece. The proposed material for the mandrel manufacture is Chromium Vanadium steel, because it provides uniform gripping forces and leaves no residual stresses on the work piece.

Empirical relationships are used to calculate the forces and pressures exerted by the cutting tool on the work piece and the work holding device. The theory of thin cylinders is used to calculate the theoretical maximum hoop and longitudinal stresses. ANSYS software is used to create the statical model. The developed model was assessed by applying forces and pressures to the mandrel, which results in deformations and stresses. The theoretical stresses on the mandrel are compared to the stresses obtained from ANSYS. The correlation between the two analyses is found to be satisfactory. Modal analysis is used to obtain the natural frequencies and mode shapes for the same model, which are also useful for dynamic analysis.

To determine the dynamic response of a structure under the action of dynamic loads, the “Mode Superposition Transient Dynamic Analysis” method is used. By using the natural frequencies and mode shapes obtained from modal analysis the dynamic analysis is done in ANSYS on the model. At various nodes, time-varying deformations are obtained. The deformations obtained in dynamic analysis at various nodes are compared to those obtained in static analysis. The level of agreement between them is deemed to be reasonable.

Keywords: Mandrel, Work holding device, Chromium Vanadium steel, Thin cylinder, Dynamic Analysis.

1. INTRODUCTION

Manufacturing is derived from the Latin word "manufactus," which means "made by hand."[1]. Manufacturing in the modern sense entails using various processes to produce products from raw materials. Casting, forming and shaping, machining, joining, and finishing are all forms of manufacturing processes.

2. NEED OF WORK HOLDING DEVICES IN MANUFACTURING PROCESSES

Work holding devices are essential for any manufacturing process. Any operation, such as cutting or turning, requires that the work be held firmly. The required parameters can be maintained by firmly holding the work piece. Regardless of the machining operation, if the work holding devices are not in proper alignment, the work piece may be damaged or irregularities such as concentricity, ovality, and cylindricity may occur. As a result, a good work holding device is required for firm gripping of the components at high speeds and low feeds, as well as maintaining the required parameters after machining. These work holding devices are often called as jigs and fixtures, which are the most common devices used in modern manufacturing. [2], [3].

3. OVERVIEW OF THE PAPER

HINO model liner design parameters were used to investigate various types of mandrels. The design parameters (maximum pressure, maximum stress, etc.) that may occur during the machining operation were calculated with the help of a few books. Certain assumptions were made for the mandrel dimensions based on the liner specifications. To identify the proper material for the design of the mandrel, the spring steel family was chosen, and ‘50 Cr 1 V 23' (chromium vanadium steel) was selected. With these parameters, a mandrel was designed that works on hydrostatic force, and the design was verified using computer aided analysis.

4. THEORY OF MANDREL

A mandrel is a tool component that grips or clamps the material to be machined, pronounced "mandrul" and also transliterated as "manderil." [4].

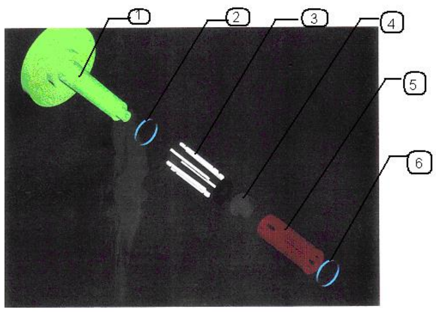
Mandrels are not recent inventions. Metal machining that uses the spinning process has been documented since Egyptian hieroglyphic times. A wood or metal spinning mandrel is used in the form of the internal contour of the part to be produced. This method clamps the raw material securely and allows for precise machining into the desired final shape. Since the material is clamped internally then there is no interference to the operator from the lathe/mandrel assembly during production.

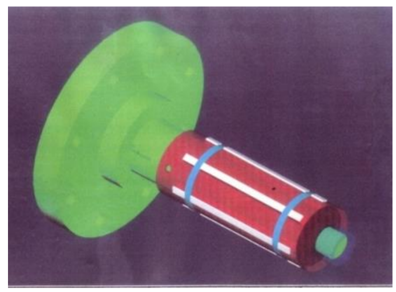
Also around the year 1700, the traversing mandrel was introduced, which entails designing a lathe mandrel that can slide axially in its bearings under the control of the operator. So that components having short lengths of thread could be produced, such as screws. During this era, the traversing mandrel was primarily used by clockmakers and ornamental turners. This device was eventually superseded by a mandrel-driven device known as a lead screw, which uses a train of gears that can be adjusted as needed for the turning application.

5. TYPES OF MANDRELS

5.1 Manually operated blade type mandrel

When pushed through a tapered surface, this type of mandrel works on the principle of lifting the blades. The blades have counter tapered and shaft has a tapered surface.





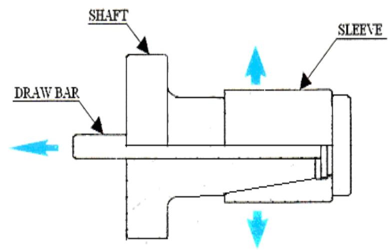


Fig.1. Blade type mandrel disassembled parts
(1) Solid tapered shaft (4) Nut
(2) Front Ring (5) Cover
(3) Blades (6) Rare Ring

Fig. 2. Assembling of blade type mandrel

Fig.3. Sleeve type mandrel

Manual effort is required to operate this type of mandrel. It's similar to a nut and bolt system. As shown in Fig.1 and Fig.2, it has a solid tapered shaft, blades, rings, cover, and nut. The tapered shape of the shaft has flexibility to catch various components.

The shaft is fitted to the lathe, the corresponding cover with blades is placed on the shaft, and finally the nut is used to tighten the cover, causing the blades to move upward. This blade movement grips the work piece. [5].

Manually operated blade type mandrels have more complicated mechanisms, suffer from more wear and tear, and thus have lower accuracy. Improper tightening of the mandrel reduces the efficiency of the mandrel, and elapsed time during assembly and disassembly is longer.

5.2 Sleeve type expansion mandrel

Sleeve type mandrel is shown in Fig.3, Fig.4 and Fig.5. This type of mandrel works on the principle of expansion of metal sleeves when they are pushed against two tapered surfaces. Here the pushing force is providing by a hydraulic operated drawbar [6].

This type of mandrel consists of three major components sleeve, shaft and drawbar. When certain hydraulic pressure is applied to push the drawbar then the sleeve starts to move in the direction to the left as it does not have the space to move due to the two counter tapers provided on the shaft to the sleeve. Metal starts to expand throughout the cylindrical surface it causes the liner holding during machining.

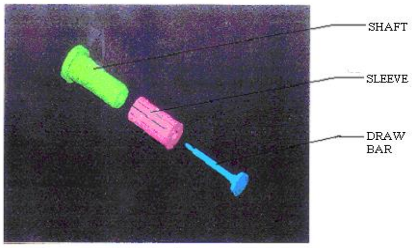


Fig.4. Sleeve type mandrel disassembly

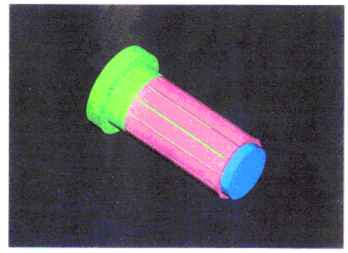


Fig.5. Sleeve type mandrel assembly

Problems encountered with sleeve type mandrel are Improper gripping, Handling of the mandrel is difficult due to its more weight, Difficulty of holding various components and Wear out problems of the taper surfaces due to frictional effects.

6. SCOPE OF THE PRESENT WORK

To resolve the aforementioned issues, a hydraulically actuated mandrel based on the principle of hydrostatic pressure is proposed.

7. DESIGN AND MODELLING OF HYDRAULIC ACTUATED MANDREL

This paper gives a systematic way of designing a hydraulic actuated mandrel. In which force by empirical relation, external pressure on the mandrel and maximum stresses on the mandrel are calculated.

7.1 Need for a new design

Blade type mandrel and Sleeve type mandrel having the following disadvantages.

As the contacting area of the sleeve is less, uniform gripping is not possible. Holding of various sized components is not possible. Handling of the existing mandrel is very difficult due to its more weight. Wear out problems of the taper surfaces due to frictional effects. To overcome the problems and as per the product requirements hydraulic actuated mandrel is designed.

7.2 Requirements of design

Different sized components are to be handled by the same mandrel, Handling should be simple, Wear and tear should be minimum and uniform gripping is required.

8. MODELLING OF HYDRAULIC ACTUATED MANDREL

Fig. 6 shows uniform hydrostatic pressure is used to expand the mandrel and Fig.7 shows the Pressure distribution in the mandrel with dimensions. An uninterrupted cylindrical sleeve of uniform wall thickness, made of high quality spring steel (chromium - vanadium steel) expands uniformly under hydrostatic pressure. Here the hydro - static pressure acts on the outer wall of the sleeve, which expands uniformly. When a liner or any cylindrical component is placed on the mandrel due to the hydro - static expansion the mandrel expands; it centers the component and holds it with great security [7].

9. DESIGN SPECIFICATION OF THE MANDREL

The main consideration in the design of the mandrel is its dimensions as we have taken a standard liner "HINO" model liner the specifications of the mandrel dependent on it. The specifications of the liner are shown in Table.1 and Design specifications of the mandrel are shown in Table.2.

Table.1, Specifications of the liner

S.NO	DESCRIPTION	DIMENSION
1	Total Length	203.0-203.5 mm
2	Outer Diameter	113.65-113.85 mm
3	Inner Diameter	105.0-105.20 mm
4	Diameter of The Collar	113.65mm
5	Depth of Collar	8mm

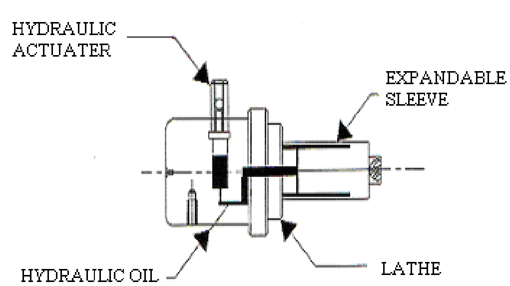


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Table.2, Specifications of the mandrel

S.NO	DESCRIPTION	DIMENSION
1	Total Length	210 mm
2	Outer Diameter	105 mm
3	Sleeve Thickness	2 mm
4	Inner Solid Diameter	93 mm
5	Cylindrical Space	4 mm

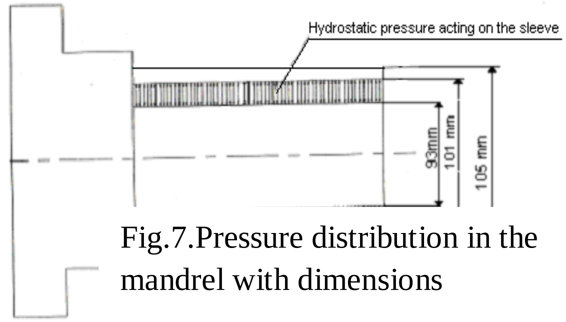

Fig.7.Pressure distribution in the mandrel with dimensions

10. MATERIAL SELECTION:

Select the material, which will be the most suitable one as per the requirement of the mandrel is High strength, More elastic and Lesser in weight.

For the fulfillment of the above requirements the CHROMIUM-VANADIUM STEEL (50 Cr 1 V 23) material is chosen from the spring steel family and selected the required material properties as shown in Table.3 [8].

Table.3 Material properties of “50 Cr 1 V 23” [8]

Density (x1000 kg/m^3)	7.7. - 8.03
Poisson’s Ratio	0.27 - 0.30
Tensile Strength (Mpa)	1900-2400
Yield Strength (Mpa)	1800
Carbon (%)	0.45-0.55
Silicon (%)	0.1-0.35
Manganese (%)	0.5-0.8
Chromium (%)	0.9-1.2
Vanadium (%)	0.23
Hardness (BHN)	240

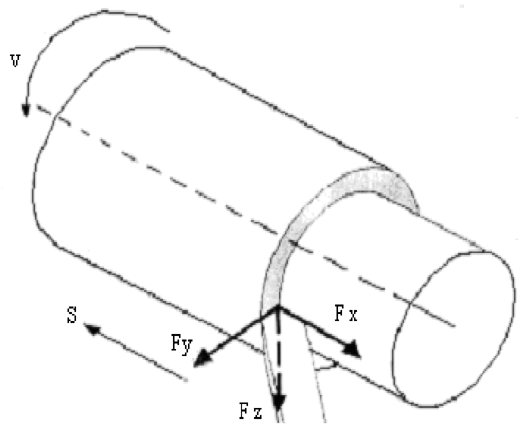

Fig.8 Forces during turning operation [10]

11. HYDRAULIC FLUID SELECTION

As the uses of high pressure proliferated, so does the need for understanding the properties of fluid under pressure. Fluids will exhibit strange and rarely helpful changes as pressure rises. These changes are not quickly nor easily predicated nor understood, not even at fairly moderate pressures, to use standard hydraulic oil it has many good properties at room pressure that you would like in a pressure fluid it lubricates; it has reasonable viscosity; it is relatively inert, inexpensive; and, at ambient pressure at least, it transmits pressure very well. The system, however, would not deliver enough pressure. The pressure in the vessel, remote from the pump, would not increase beyond 3500 psi. Selected WHITE GASOLINE (ANOCO) from many available hydraulic fluids, that can produce a pressure from 0 to 200,000 psi. and which will be the most suitable one for this application [9].

12. FORCE CALCULATION

Fig.8 shows Forces during turning operation. To calculate forces in turning operation “GRANOVSKY’S EMPIRICAL LAW” for cutting is used.

Cutting force is affected by the various cutting variables such as feed, depth of the cut, tool geometry, cutting speed, tool wear and hardness of the work piece material.

By considering the effect of various cutting variables on the cutting force, Granovsky’s empirical equation is given by [11]

$$\mathbf{F\,(x, y, z) = [C \cdot K_{\square} \cdot K_{\varnothing} \cdot K_{\square} \cdot K_{r} \cdot K_{hf} \cdot K_{cf} \cdot K_{m} \cdot (B.H.N.)^{m} \cdot t^{p} \cdot s^{q}]\,(x, y, z)} \ldots\ldots\ldots\ldots(1)$$

From the literature the following values are considered to find out the forces Fx, Fy and Fz during turning operation.

For the turning operation if the BHN value of the material is >170 then the material constant, C in x, y and z directions are 0.051, 0.045 and 5.14 respectively. Coefficients of rake angle, $K_{\square\square}$ in x, y and z directions are 1.28, 1.4 and 1.08 respectively. If the side cutting edge angle of the tool is 75^0 the coefficient of principle side cutting edge angle, $K_{Ø}$ □in x, y and z directions are 1.20, 0.77 and 0.94 respectively. At 200 m/minute cutting speed the coefficient of cutting speed, K□□in x, y and z directions are 0.60, 0.60 and 0.80 respectively. If nose radius R=0.80 mm, the coefficient of nose radius, K_r□□in x, y and z directions are 1.224, 0.826 and 0.934 respectively. The value of tool wear from 0.5 to 0.75 mm then the coefficient of tool wear, K_r□□in x, y and z directions is 1.03. Mineral oil as a cutting fluid for the operation and 0.85 is the coefficient of cutting fluid, K_{cf}. The material transfer coefficient, K_m is 0.50 for Cast Iron material at ultimate stress □$_u$ =20 Kgf/mm^2. The value of exponent 'm' of equation (1) for Cat Iron material in x, y and z directions are 1.1, 1.3 and 0.55 respectively. Effect of depth of cut, p and effect of Feed, q for Cast Iron material in turning operation in x, y and z directions are 1.2, 0.9 and 1.0 respectively. Effect of Feed, q for Cast Iron material in turning operation in x, y and z directions is 0.75. Substituting corresponding values in the force equations (1) which gives the forces along F_x, F_y and F_z [11], [12].

12.1 Specifications of Turning Parameters

The following specifications (table 4) are taken in to consideration while designing a mandrel [12].

Table 4, Specifications of turning operation

S.NO	DESCRIPTION	SPECIFICATION
1	Rake Angle	-6 Degrees
2	Side Cutting Edge Angle	75 Degrees
3	Cutting Speed	200 m/minute
4	Nose Radius	0.8 mm
5	Tool Wear	0.5-0.75mm
6	Type of Lubricant	Mineral Oil (ASTMA32)
7	Liner Material	Cast Iron (HINO)
8	B.H.N Value	230-280
9	Depth of Cut	2 mm
10	Feed	0.36 mm

Equation (1) can be written in the three directions as shown in equations (1a), (1b) and (1c)

12.2 Force acting along X-direction

$$F_x = [C \cdot K_{□} \cdot K_{Ø} \cdot K_{□} \cdot K_r \cdot K_{hf} \cdot K_{cf} \cdot K_m \cdot (B.H.N.)^m \cdot t^p \cdot s^q] \quad \text{.......................(1a)}$$

Substituting the corresponding values from the data then the value is

$$F_x = 11.1639 \text{ kgf}$$

12.3 Force acting along Y-direction

$$F_y = [C \cdot K_{□} \cdot K_{Ø} \cdot K_{□} \cdot K_r \cdot K_{hf} \cdot K_{cf} \cdot K_m \cdot (B.H.N.)^m \cdot t^p \cdot s^q] \quad \text{.......................(1b)}$$

Substituting the corresponding values from the data

$$F_y = 26.35037 \text{kgf}$$

12.4 Force acting along Z-direction

$$F_z = [C \cdot K_{□} \cdot K_{Ø} \cdot K_{□} \cdot K_r \cdot K_{hf} \cdot K_{cf} \cdot K_m \cdot (B.H.N.)^m \cdot t^{p(z)} \cdot s^{q(z)}] \quad \text{...............(1c)}$$

Substituting the corresponding values from the data

$$F_z = 32.3259 \text{kgf}$$

13. PRESSURE CALUCULATION

Now the pressure has to be calculated for a mandrel to withstand the above said forces, considering the liner dimensions

13.1 Dimensions of the liner (HINO)

The following are the actual dimensions of the liner before machining are shown in Table 5

Table 5. Dimensions of the liner

S.NO	DESCRIPTION	DIMENSION
1	Total Length	203.0-203.5mm
2	Outer Diameter	113.65-113.85mm
3	Inner Diameter	105.0-105.20mm
4	Diameter of the Collar	113.65mm
5	Depth of the Collar	8mm

Table 6. Design values of the liner

S.NO	DESCRIPTION	DIMENSION
1	Minimum value of outer diameter	113.65 mm
2	Maximum value of inner diameter	105.20 mm
3	Minimum length	195 mm

To calculate the external pressure acting on the liner that requires the contact area of the liner. For safe design minimum values of the liner should be taken as shown in Table 6.

13.2 Contact Area of the liner (C.A.)

Contact area of the liner (C.A.) = (Outer diameter - Inner diameter) X (Length) ……(2)

$$C.A. = 1647.75\ mm^2 = 16.4775\ cm^2$$

13.3 Pressure acting along X - direction

External Pressure acting on the liner along x – direction(P_x) = Force (x) ÷ contact area

$$P_x = F_x \div C.A. \quad \text{………………..(3)}$$

By substituting corresponding values, Px= 0.6775231 kgf/cm^2.

13.4 Pressure acting along Y - direction

External Pressure acting on the liner along y-direction (P_y)= Force (y) ÷ contact area.

$$P_y = F_y \div C.A. \quad \text{………………….(4)}$$

By substituting corresponding values, P_y = 1.5991728 kgf / cm^2.

13.5 Pressure acting along Z - direction

External Pressure acting on the liner along z-direction (P_z) = Force (z) ÷ contact area.

$$P_z = F_z \div C.A. \quad \text{…………………(5)}$$

By substituting corresponding values, P_z = 1.9618207 kgf / cm^2.

13.6 Resultant external pressure acting on the liner

No need to consider the external pressure acting along the x – direction, because the x-direction is along the axis of the liner shown in fig.8 and there is no expansion of the mandrel along x-direction the expansion only along y and z directions.

Therefore the y and z directions are taken in to consideration. The resultant pressure along y and z- directions.

$$R = [(P_y)^2 + (P_z)^2]^{1/2} \quad \text{……………(6)}$$

$$=255427.8\ N/m^2.$$

Hence the external pressure acting on the liner during machining is R = 255427.8 N/m^2.

As the resultant external pressure acting on the mandrel is less than that of the liner. For the design purpose the same pressure is considered on the mandrel also.

14. STRESS CALCUATIONS

The stresses acting on the mandrel are calculated by using the thin cylinder concept, where the ratio of inner diameter to the thickness of the cylinder is more than 15. There are mainly two types of stresses in the thin cylinders, viz., Hoop stress and longitudinal stress. The dimensions of the mandrel to be designed are given bellow

Outer diameter of the cylinder D_0 = 105 mm, Inner diameter of the cylinder D_i = 101 mm, Thickness of the cylinder t = 2mm and the internal design pressure P_i = 39.24 N/mm^2.

Considering the equilibrium of forces in the circumferential direction [13]

$$D_i x\ P_i = 2x\square_c x\ t \quad \text{………………….(7)}$$

From (7) the Hoop stress can be written as, $\square_c = \frac{P_i D_i}{2t}$ …………………..(8)

Maximum hoop stress, $\square_c$ = 990.81 N/mm^2.

Considering the equilibrium of force in longitudinal direction, now

$$P_i \times \pi \times D_i = 4 \times \sigma_L \times \pi \times t \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

From (9) Longitudinal stress can be written as, $\sigma_L = \frac{\sigma_t}{2}$

Maximum longitudinal stress σ_L = 495.405 N/mm^2.

15. STATIC ANALYSIS OF THE MANDREL

Maximum circumferential (hoop) stress is analyzed in the software and plotted the result. Fig.9 shows the plot of maximum circumferential stress of the mandrel due to the internal pressure.

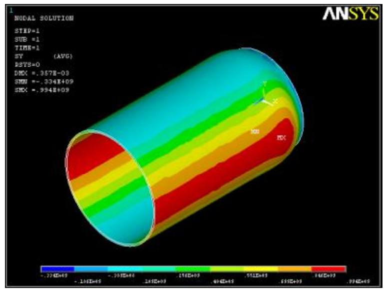

Fig.9 Maximum circumferential stress

The maximum circumferential (hoop) stress obtained from the simulation is

$$\sigma_c = 994 \times 10^6 \text{ N/m}^2.$$

Maximum longitudinal stress (σ_L) is half of the circumferential (hoop) stress σ_c

$$\sigma_L = 497 \times 10^6 \text{ N/m}^2.$$

16. VALIDATION OF RESULTS

Theoretically,

The Maximum circumferential (hoop) stress developed during expansion of the mandrel.

$$\sigma_c = 990.81 \times 10^6 \text{ N/m}^2.$$

The maximum longitudinal stress developed during expansion of the mandrel.

$$\sigma_L = 495.405 \times 10^6 \text{ N/m}^2.$$

The results obtained from the simulation,

Maximum circumferential (hoop) stress that can withstand by the mandrel during expansion

$$\sigma_c = 994 \times 10^6 \text{ N/m}^2.$$

Maximum longitudinal stress that can withstand by the mandrel during expansion

$$\sigma_L = 497 \times 10^6 \text{ N/m}^2.$$

17. MODAL ANALYSIS OF THE MANDREL

Use of modal analysis is to determine the natural frequencies and mode shapes of a structure. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions.

The modal analysis can be performing on a pre stressed structure, such as hallow cylinder, spinning turbine blade etc. In this work, modal analysis is performed on the mandrel and the results were plotted.

Fig.10 shows the mode shapes for the mandrel from modal analysis results of simulation process and plotted five natural frequencies and performed eight natural frequencies for the mandrel which are 1198.1 Hz, 1228.4 Hz, 1719.3 Hz, 1720.2 Hz, 2789.8 Hz, 2790.6 Hz, 3370.8 Hz and 3375.6 Hz.

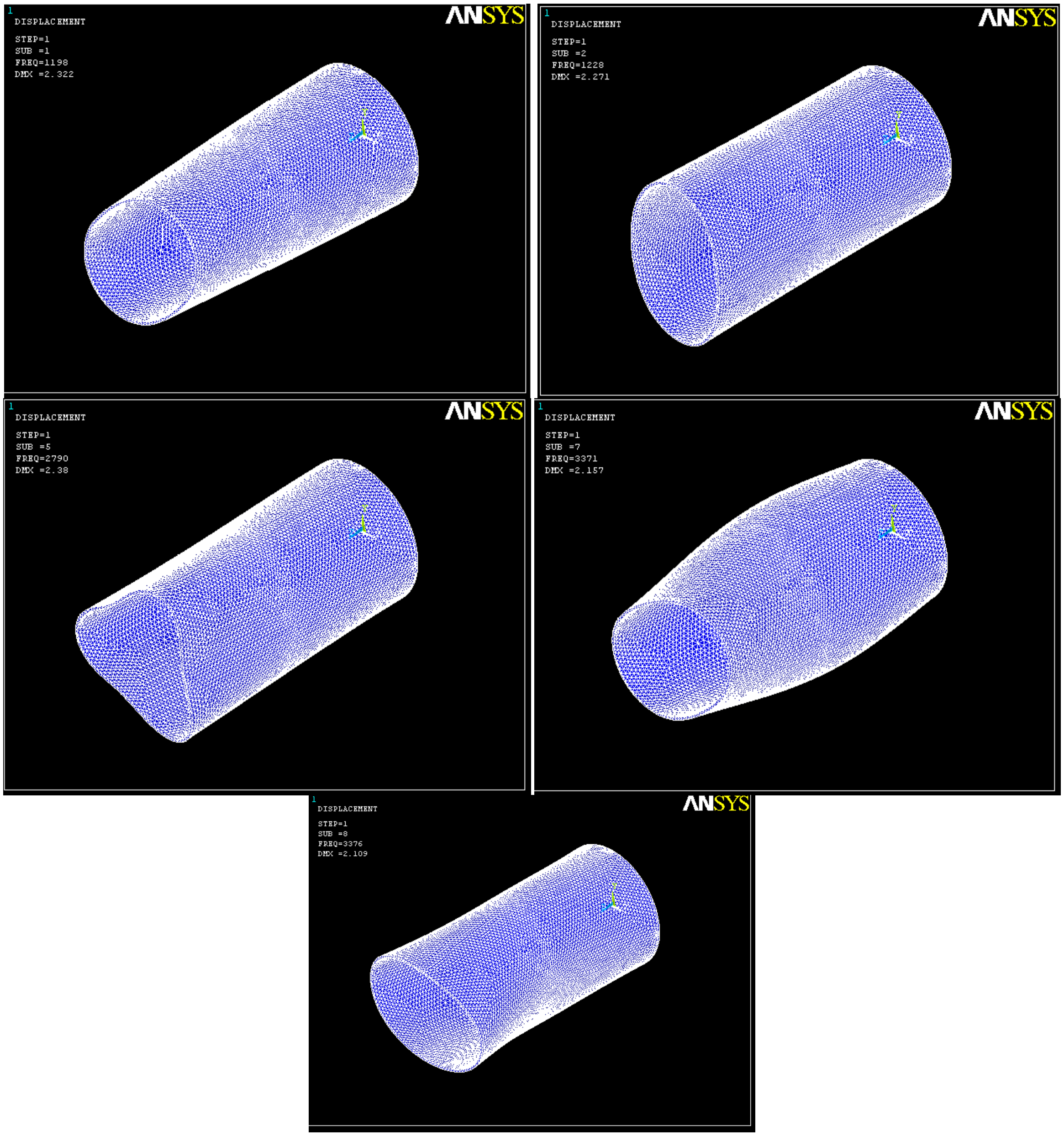

Fig.10 Natural frequencies of the Mandrel

18. TRANSIENT DYNAMIC ANALYSIS OF THE MANDREL

Transient dynamic analysis (sometimes called time-history analysis) is a technique used to determine the dynamic response of a structure under the action of any general time-dependent loads. It is used to determine the time-varying displacements, strains, stresses, and forces in a structure as it responds to any combination of static, transient, and harmonic loads. The time scale of the loading is such that the inertia or damping effects are considered to be important.

The basic equation of motion solved by a transient dynamic analysis is

$$(M)\{\ddot{U}\} + (C)\ \{\acute{u}\} + (K)\{u\} = \{F(t)\} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (10)$$

Where:

(M)	= mass matrix	(C)	= damping matrix
(K)	= stiffness matrix	(Ü)	= nodal acceleration vector
(ú)	= nodal velocity vector	(u)	= nodal displacement vector
{F(t)}	= load vector		

Simulation process runs about 60 seconds and Fig. 11, Fig.12, Fig.13 and Fig.14 illustrate the dynamic response of the mandrel in which time–displacement results are plotted at four different nodes and Fig.15 shows comparative deformation analysis of same four nodes. It is observed that the response of the mandrel of the dynamic analysis is matched with that of static analysis result.

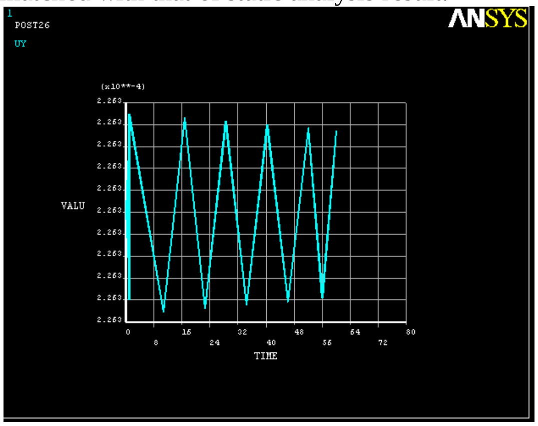

Fig.11 Deformation at node ‘13727’

Table.7 demonstrates the time varying displacements shown in Fig. 11.

Table.7 ANSYS variable listing for node ‘13727’

Time (Sec)	NSOL UY ‘13727’ (m)	Time (Sec)	NSOL UY ‘13727’ (m)
0.16667	$0.226321x10^{-3}$	10.833	$0.226320x10^{-3}$
0.26667	$0.226321x10^{-3}$	16.733	$0.226321x10^{-3}$
0.36667	$0.226320x10^{-3}$	22.633	$0.226320x10^{-3}$
0.46667	$0.226321x10^{-3}$	28.533	$0.226321x10^{-3}$
0.56667	$0.226320x10^{-3}$	34.433	$0.226320x10^{-3}$
0.66667	$0.226321x10^{-3}$	40.333	$0.226321x10^{-3}$
0.76667	$0.226320x10^{-3}$	46.233	$0.226320x10^{-3}$
0.86667	$0.226321x10^{-3}$	52.133	$0.226321x10^{-3}$
0.93333	$0.226320x10^{-3}$	56.067	$0.226320x10^{-3}$
1.0000	$0.226321x10^{-3}$	60.0	$0.226321x10^{-3}$

Table.8 ANSYS variable listing for node ‘3402’

Time (Sec)	NSOL UY ‘3402’ (m)	Time (Sec)	NSOL UY ‘3402’ (m)
0.16667	$0.232384x10^{-3}$	10.833	$0.232384x10^{-3}$
0.26667	$0.232384x10^{-3}$	16.733	$0.232384x10^{-3}$
0.36667	$0.232384x10^{-3}$	22.633	$0.232384x10^{-3}$
0.46667	$0.232384x10^{-3}$	28.533	$0.232384x10^{-3}$
0.56667	$0.232384x10^{-3}$	34.433	$0.232384x10^{-3}$
0.66667	$0.232384x10^{-3}$	40.333	$0.232384x10^{-3}$
0.76667	$0.232384x10^{-3}$	46.233	$0.232384x10^{-3}$
0.86667	$0.232384x10^{-3}$	52.133	$0.232384x10^{-3}$
0.93333	$0.232384x10^{-3}$	56.067	$0.232384x10^{-3}$
1.0	$0.232384x10^{-3}$	60.0	$0.232384x10^{-3}$

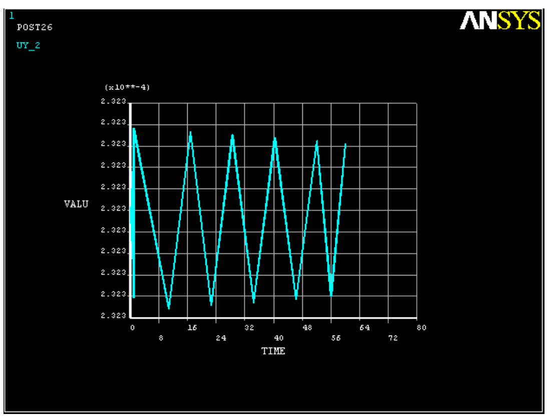

Fig.12 Deformation at node ‘3402’

Table.8 demonstrates the time varying displacements at node 3402 of the mandrel at different time intervals as shown in Fig. 12.

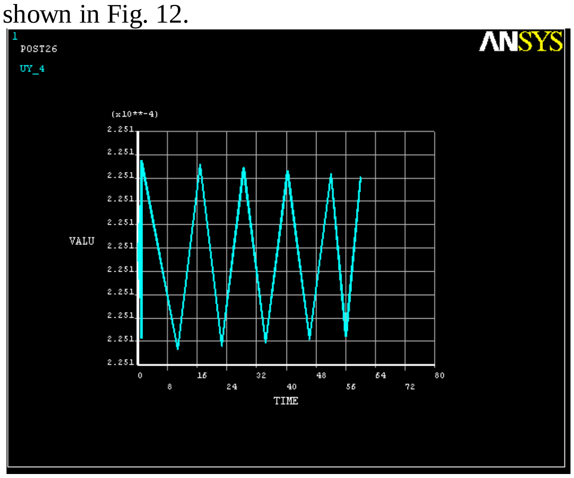

Fig.13 Deformation at node ‘3894’

Table.9 ANSYS variable listing for node ‘3402’

Time (Sec)	NSOL UY ‘3894’ (m)	Time (Sec)	NSOL UY ‘3894’ (m)
0.16667	$0.225158x10^{-3}$	10.833	$0.225158x10^{-3}$
0.26667	$0.225158x10^{-3}$	16.733	$0.225158x10^{-3}$
0.36667	$0.225158x10^{-3}$	22.633	$0.225158x10^{-3}$
0.46667	$0.225158x10^{-3}$	28.533	$0.225158x10^{-3}$
0.56667	$0.225158x10^{-3}$	34.433	$0.225158x10^{-3}$
0.66667	$0.225158x10^{-3}$	40.333	$0.225158x10^{-3}$
0.76667	$0.225158x10^{-3}$	46.233	$0.225158x10^{-3}$
0.86667	$0.225158x10^{-3}$	52.133	$0.225158x10^{-3}$
0.93333	$0.225158x10^{-3}$	56.067	$0.225158x10^{-3}$
1.0000	$0.225158x10^{-3}$	60.0	$0.225158x10^{-3}$

Table.9 demonstrates the time varying displacements at node 3894 of the mandrel at different time intervals as shown in Fig. 13.

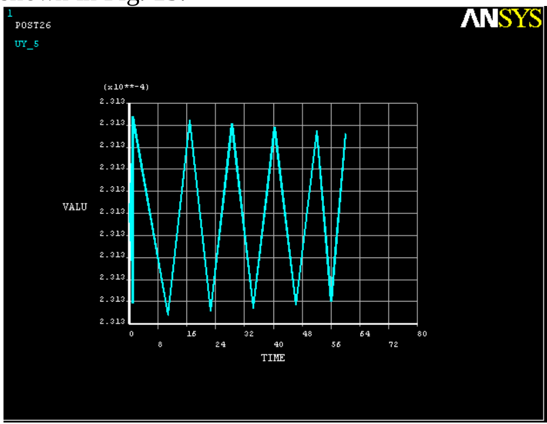

Fig.14 Deformation at node '635'

Table.10 ANSYS variable listing for node '635'

Time (Sec)	NSOL UY '635' (m)	Time (Sec)	NSOL UY '635' (m)
0.16667	0.231349×10^{-3}	10.833	0.231349×10^{-3}
0.26667	0.231349×10^{-3}	16.733	0.231349×10^{-3}
0.36667	0.231349×10^{-3}	22.633	0.231349×10^{-3}
0.46667	0.231349×10^{-3}	28.533	0.231349×10^{-3}
0.56667	0.231349×10^{-3}	34.433	0.231349×10^{-3}
0.66667	0.231349×10^{-3}	40.333	0.231349×10^{-3}
0.76667	0.231349×10^{-3}	46.233	0.231349×10^{-3}
0.86667	0.231349×10^{-3}	52.133	0.231349×10^{-3}
0.93333	0.231349×10^{-3}	56.067	0.231349×10^{-3}
1.0000	0.231349×10^{-3}	60.000	0.231349×10^{-3}

Table.10 demonstrates the time varying displacements at node 635 of the mandrel at different time intervals as shown in Fig. 14 and Fig. 15 shows the plot of deformation results comparison at nodes '13727, 3402, 3894 and 635, it is observed that the deformation is approximately equal at all nodes.

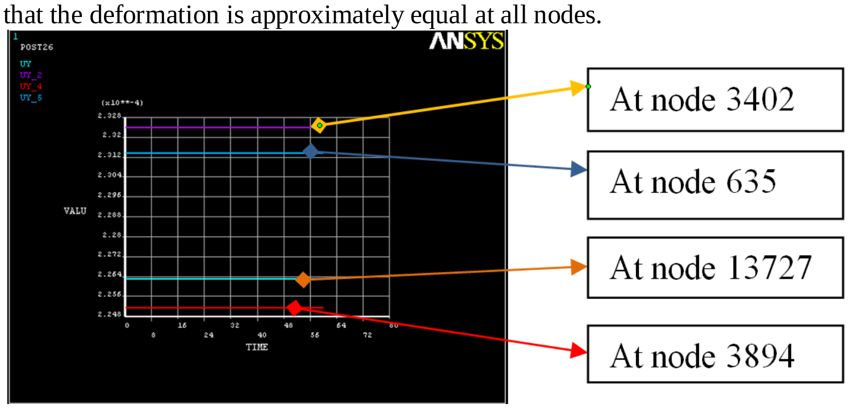

Fig .15 Comparative graph of deformation at nodes '13727, 3402, 3894 and 635

19. CONCLUSIONS

From the present work, the following conclusions are drawn

The problems such as improper gripping, less expansion rates and more weight were overcome with this hydraulic actuated mandrel. Chromium Vanadium steel is proposed as the material for this mandrel manufacture, as it offers uniform gripping forces and leaves no residual stress over the work piece. Aspects of precision and accuracy that depend on the factors such as rake angle, cutting speed, tool wear etc. are fulfilled. A good unison is obtained between the theoretically calculated values and the simulations done over the mandrel using computer-aided tool such as ANSYS.

20. FUTURE SCOPE

The present work is done for Static and Dynamic Analysis and it can be extended to perform Harmonic Analysis (vibrations and its effects on precision) and analysis can be done for various sizes of liners.

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