‘Hybrid Vehicle Evolution and Future’
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Abstract:
These instructions give you guidelines for preparing papers for ‘Hybrid vehicle evolution and future’ in International Journal of Science Technology and Engineering. Use this document as a template if you are using Microsoft Word 6.0 or later. Otherwise, use this document as an instruction set. The electronic file of your paper will be formatted further at International Journal of Engineering and Technology. Define all symbols used in the abstract. Do not cite references in the abstract. Do not delete the blank line immediately above the abstract; it sets the footnote at the bottom of this column.

Keywords — About four key words or phrases in alphabetical order, separated by commas.

1.1 Problem Statement
In any direct mechanical drive system, there exists a need to couple the variety of driven elements that may be included. The majority of drive

1.INTRODUCTION
A coupling is a device used to connect two shafts together at their ends for the purpose of transmitting power. Couplings do not normally allow disconnection of shafts during operation, however there are torque-limiting couplings which can slip or disconnect when some torque limit is exceeded.
The primary purpose of couplings is to join two pieces of rotating equipment while permitting some degree of misalignment or end movement or both. By careful selection, installation and maintenance of couplings, substantial savings can be made in reduced maintenance costs and downtime.
The Thomson constant velocity joint is a constant velocity joint with no parasitic bearing of sliding surfaces. This invention offers a revolution in the design of many transmission system, for instance in vehicular, marine, manufacturing, industrial and aeronautical application.
Thomson constant velocity joint is essentially two cardan joints assembled co-axially where the cruciform-equivalent members of each are connected to one another by trunions and bearings which are constrained to continuously lie on the homokinetic plane of the joint.
Basically the TCVJ has the same constructions as a normal cardan joint but does not suffer the dynamic loads due to fluctuating angular velocity of intermediate shaft and load, as is the case where cardan joints are used. As a result, the Thomson constant velocity joint has a life exceeding an ordinary cardan joint. There is no untried technology in the Thomson constant velocity joint. It is essentially identical to two cardon joints in its torque transmission. There are various constant velocity joint series available; constant, double constant velocity variable angle joints, for shaft angles to 30 degree. Even 90 degrees can be realized wit rigid and multiple rigid, angle constant velocity joint.

RESEARCH ARTICLE
OPEN ACCESS
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Keywords — About four key words or phrases in alphabetical order, separated by commas.
elements, including gear reducers, lead screws, and a host of other components, are driven by shafting that is supported by multiple bearings. This allows for shafting to be held extremely straight and rigid while rotating, avoiding any possible balancing and support problems. Because of this rigid support, it is virtually impossible to avoid slight misalignments between a driving and driven shaft when they are connected. Present technology in joints offers higher cost of joints, larger space and variable speed ratio if misalignment is present. The main concept is to lower cost of production, space requirement and simply technology of manufacture as compared to present CVJ in market.

1.2 Objectives
a) Design & drawing of kinematic linkage to deliver parallel as well angular offset over a range.
b) Development & manufacturing of drive.
c) Testing of drive to derive the performance.
d) Plot Performance Characteristic Curves.

e) Singular control: Entire range of offset is covered by a single hand wheel control.

1.3 Scope
The following features of the drive will lead to application of drive in variety of field applications:
a) **Step-less variation of angular offset**: Any displacement between 0 to 60 mm can be obtained. Hence the drive provides flexibility in operation and setting as prime mover location can be varies as per space available.
b) **Wide range of angular displacement**: The wide range of angular displacement 30 to 65 degrees enables to get vibration free power transmission at high speed. This will be especially useful in spring making machinery, printing machinery and automatic transfer lines.

c) **Compact size**: The size of the gear less variable speed reducer is very compact; which makes it low weight and occupies less space in any drive.
d) **Ease of operation**: The changing of angular and angular offset is gradual one hence no calculations of speed ratio required for change gearing. Merely by rotating hand wheel speed can be changed

e) **Singular control**: Entire range of offset is covered by a single hand wheel control.

1.4 Methodology
Following activities will be carried out during this dissertation work. It includes literature survey, system design, mechanical design, fabrication, assembly, testing and experimental analysis, and comparative study etc.

1.4.1 Literature review. Study of various power transmission drives in machine tool systems using various drive-train handbooks, United State Patent documents, Technical papers, etc.

1.4.2 Development of theory.
A) **System Design**: This part includes the design and development for the kinematic linkage as per the geometry to produce the desired output

B) **Mechanical Design**: This part includes the design and development of linkages, selection of suitable drive motor, strength analysis of various components under the given system of forces

1.4.3 Fabrication:
Suitable manufacturing methods will be employed to fabricate the components and then assemble the test
1.4.4 Testing:
Testing of the joint to derive performance characteristics namely:
a) Torque vs. Speed.
b) Power vs. Speed.
c) Efficiency vs. Speed.
d) Maximum angular offset, and performance at maximum parallel offset.
e) Maximum angular offset and performance at maximum angular offset.

2 LITERATURE REVIEW
2.1 Ian Watson, B. Gangadhara Prusty and John Olsen have stated in research paper titled “Conceptual design optimization of a constant velocity coupling” that The Thompson Coupling operates using the robust double Cardan mechanism. Constant velocity and determinate linkage kinematics are maintained by a spherical pantograph. This mechanism forms an extra loop attached to the intermediate shaft in the double Cardan linkage, and consequently constrains this shaft to bisect the axis of input and output. Closed-form expressions for its motion and the rotation of the double Cardan joint are derived by consideration of spherical linkage kinematics. These expressions are then used to drive basic conceptual design optimization, whose goal is to reduce induced driveline vibration. The findings of this optimization are discussed with respect to the current design of the Thompson joint. Improvements in induced driveline vibration are possible, subject to the satisfaction of other coupling design criteria.

2.2 Chul-Hee Lee and Andreas A. Polycarpou has proposed in their research paper titled “A phenomenological friction model of tripod constant velocity (CV) joints” that constant velocity (CV) joints have been favored for automotive applications, compared to universal joints, due to their superiority of constant velocity torque transfer and plunging capability. High speed and sport utility vehicles with large joint articulation angles, demand lower plunging friction inside their CV joints to meet noise and vibration requirements, thus requiring a more thorough understanding of their internal friction characteristics. A phenomenological CV joint friction model was developed to model the friction behavior of tripod CV joints by using an instrumented CV joint friction apparatus with tripod-type joint assemblies. Experiments were conducted under different operating conditions of oscillatory speeds, CV joint articulation angles, lubrication, and torque. The experimental data and physical parameters were used to develop a physics-based phenomenological CV joint dynamic friction model. It was found that the proposed friction model captures the experimental data well, and the model was used to predict the external generated axial force, which is the main source of force that causes vehicle vibration problems.

2.3 Majid Yaghoubi, Seyed Saeid Mohtasebi, Ali Jafary and Hamid Khaleghi in their research work titled “Design, manufacture and evaluation of a new and simple mechanism for transmission of power between intersecting shafts up to 135 degrees (Persian Joint)” has introduced a new mechanism which is designed for the transmission of power between two
intersecting shafts. The mechanism consists of one drive shaft and one driven shaft, six guide arms, and three connecting arms. The intersecting angle between the input shaft and the output shaft can be varied up to 135° while the velocity ratio between the two shafts remains constant. The research also includes a kinematic analysis and a simulation using Visual NASTRAN, Autodesk Inventor Dynamic and COSMOS Motion. The software showed that this mechanism can transmit constant velocity ratios at all angles between two shafts. By comparing the graphs of analytical analysis and simulation analysis, validity of equations was proved.

2.4 Katsumi Watanabe and Takashi Matsuura in their research paper titled “Kinematic Analyses of Rzeppa Constant Velocity Joint by Means of Bilaterally Symmetrical Circular-Arc-Bar Joint” has proposed that mechanism whose elements are bilaterally symmetrical with respect to the bisecting plane of driving and driven rotational axes is able to use as the constant velocity joint. The constant velocity joint that is composed of input and output shafts, two circular-arc elements and the frame is a most elementary joint. The closed loop equation of the circular-arc-bar joint whose kinematic constants are any values is deduced in the form of the quadratic equation of the output angle. The Rzeppa constant velocity joint is composed of several sets of the ball and two circular-arc grooves. A relative motion of the ball to two circular-arc grooves is analyzed and the output angle error in a practical use which contains sinusoidal fluctuations with periods $2\pi$, $2\pi/3$, and $2\pi/6$ is simulated by the circular-arc-bar constant velocity joint.

2.5 Tae-Wan Ku, Lee-Ho Kim and Beom-Soo Kang in their research work titled “Multi-stage cold forging and experimental investigation for the outerrace of constant velocity joints” has explored that as an important load-supporting automobile part that transmits torque between the transmission and the driven wheel, the outer race of CV (constant velocity) joints with six inner ball grooves has been conventionally produced by the multi-stage warm forging processes, which involves several operations including forward extrusion, upsetting, backward extrusions, sizing and necking, as well as additional machining. There is still no choice but to produce the complex shaped components other than by this warm forging process. As an alternative, multi-stage cold forging process is presented to replace the traditional warm forging. The multi-stage cold forging procedure is first considered through a process assessment regarding the traditional multi-stage warm forging one. Then, the process is simplified and redesigned as one operation to produce the forged outer race and the backward extrusions of the traditional process, and the sizing and necking are also combined into a single sizing necking process.

2.6 Research Update: U-Joints versus Constant Velocity Joints (ISSN 1188-4770, Group 12 (h))
2.6.1 Cardan Joint
Until fairly recently, Cardan joints were the only option available for agricultural applications, and are still very common today. A single Cardan joint consists of a pair of U-shaped yokes on the ends of the adjoining shafts joined through beatings to a metal cross. However, a single Cardan joint is limited to a 15° deviation from a straight line before fluctuations in drive shaft speed and/or vibration begin to occur. The useful life of the Cardan joint can be drastically reduced because of vibration. They are usually used in pairs to increase the maximum operating range to 30° and to minimize speed fluctuations and vibration. With older square telescopic drive shafts, it was possible to connect the shaft so that the Cardan joints were out of phase (rotated 90° to each other). This could create a significant vibration problem. However, modern drive shafts are designed so that the telescopic shaft will only fit together by turning the sections in increments of 180°, ensuring the joints are always in phase.

It was also important that the vertical and horizontal angles of the two Cardan joints was equal to further reduce velocity fluctuation and associated driveline vibration. This could be accomplished by modifying the drawbar and machine hitch lengths so that the distances between the hitch point and the ends of the output and input shafts were equal. A more detailed explanation of PTO vibration can be found in the PAMI publication, Gleanings, 441. The life of a typical double Cardan joint drive shaft is reduced to 75% for 200 deviation from a straight line, and is halved when operating at a straight-line deviation of 30°.

### 2.6.2 Constant Velocity Joint

The development of Constant Velocity (CV) joints has greatly improved the angle at which a driveline may operate from a straight line before loss of power and/or vibration occurs. The Constant Velocity joint's driving members are steel balls constrained in curved grooves between the forks of the joint. The design is such that a CV joint may operate efficiently up to a 80° deviation from a straight line. By operating in pairs, the angle can be increased accordingly. As with the Cardan joint, the effective life of a CV joint will be shortened as joint angles increase. While equalization of joint angles is still important, it is less of a concern for CV joints by their nature. For large angles, there still may be some vibration if the joint angles are not equal. Some new equipment designs require drivelines that have
large joint angles. This is where wide-angle Constant Velocity joints shine. CV joints are necessary for high velocity power transmission and for axles and kingpins of steered traction wheels on modern farm machinery.

2.6.3 Cost Considerations
Constant Velocity Joints are more expensive than Cardan joints. But replacing Cardan joints over time may in fact, become more expensive than an investment in the more versatile CV joint. CV joints are necessary where high velocity power transmission is required and operating angles are acute. Paired Cardan joints are not able to transmit power properly where angles exceed 30° without losing power and/or causing vibrations. Cardan joints pressed into operation where they are unsuitable results in dramatically reduced life of the joint.

2.6.4 The Constant Velocity Mystery
Constant velocity in PTO drivelines is an ideal operating condition, and can be achieved with both Cardan joints and constant velocity joints. But there is more than one method of achieving constant velocity in drivelines. A typical driveline with Cardan joints at each shaft end will have constant velocity if the operating geometry is arranged so: that the yokes on the intermediate shaft are in phase and the hitch point is centred between the PTO output shaft on the tractor and the PTO input shaft on the implement. Another method of achieving constant velocity is through the use of Double Cardan joints, which overcome the limitations of PTO drivelines that have two or more sets of single Cardan joints. Double Cardan joints are typically used where operating angles are too large for single cardan joints. A Double Cardan joint is essentially two single Cardan joints connected by a coupling yoke that contains a centering mechanism. This centering mechanism keeps the input and output shafts in the same plane, regardless of the operating angle. A Wide-angle Double Cardan joint uses a centering mechanism comprised of a flat disc with sockets that support the ball stud yokes. This centering mechanism compensates for velocity fluctuations of the two Cardan joints, thereby providing a constant velocity output. Other Double Cardan joints use centering mechanisms that incorporate a ball and stud mechanism, or a ball and seat mechanism. Double Cardan joints with these centering mechanisms are considered to be near constant velocity joints because their centering mechanisms do not split the misalignment between the shafts equally for all operating angles. Consequently, these joints do not produce true constant velocity output - except at the design angle. (All joints are designed to transfer power efficiently up to a maximum angle - the design angle. Operation beyond the design angle results in excessive vibration.) For practical purposes, the resulting velocity fluctuation is negligible. In comparison, the centering mechanism in a Wide-angle Double Cardan joint always splits the misalignment between shafts equally. As a result, the wide-angle Double Cardan joint has a true constant velocity output at all operating angles up to the design angle. Double Cardan joints with ball-and-stud or ball-and-seat mechanisms are typically designed for higher speeds than are Wide-angle Double Cardan joints. The Wide-angle Double Cardan joint is most commonly used where speeds do not exceed 1000 rpm.
3 SYSTEM DESIGN
3.1 Design Considerations
In system design we mainly concentrated on the following parameters:

3.1.1 System Selection Based on Physical Constraints
While selecting any machine it must be checked whether it is going to be used in a large-scale industry or a small-scale industry. In our case it is to be used by a small-scale industry. So space is a major constrain. The system is to be very compact so that it can be adjusted to corner of a room. The mechanical design has direct norms with the system design. Hence the foremost job is to control the physical parameters, so that the distinctions obtained after mechanical design can be well fitted into that.

3.1.2 Arrangement of Various Components
Keeping into view the space restrictions the components should be laid such that their easy removal or servicing is possible. More over every component should be easily seen none should be hidden. Every possible space is utilized in component arrangements.

3.1.3 Components of System
As already stated the system should be compact enough so that it can be accommodated at a corner of a room. All the moving parts should be well closed & compact. A compact system design gives a high weighted structure which is desired.

3.1.4 Man Machine Interaction
The friendliness of a machine with the operator that is operating is an important criteria of design. It is the application of anatomical & psychological principles to solve problems arising from Man – Machine relationship.

3.1.5 Chances of Failure
The losses incurred by owner in case of any failure is an important criteria of design. Factor safety while doing mechanical design is kept high so that there are less chances of failure. Moreover periodic maintenance is required to keep unit healthy.

3.1.6 Servicing Facility
The layout of components should be such that easy servicing is possible. Especially those components which require frequents servicing can be easily disassembled.

3.1.7 Height of Machine from Ground
For ease and comfort of operator the height of machine should be properly decided so that he may not get tired during operation. The machine should be slightly higher than the waist level, also enough clearance should be provided from the ground for cleaning purpose.

3.1.8 Weight of Machine
The total weight depends upon the selection of material components as well as the dimension of components. A higher weighted machine is difficult in transportation & in case of major breakdown, it is difficult to take it to workshop because of more weight.
3.2 Selection of Motor
The metric system uses kilowatts (kW) for driver ratings. Converting kW to torque:
\[ T = kW \times 84518 \text{ rpm} \]
Where
- \( T \) = the torque in inch pounds
- \( kW \) = the motor or other kilowatts
- \( rpm \) = the operating speed in revolutions per minute
84518 = a constant used when torque is in inch-pounds. Use 7043 for foot-pounds, and 9550 for Newton-meters
0.3 = \( kW \times 9550 /1200 \)
\( kW = 0.038 \ kW \)
Thus the minimum input power required will be 38 watt.

3.2.1 Drive Motor
Type: - Single Phase Ac Motor.
Power: - 1 /15 Hp. (50 Watts)
Voltage: - 230 Volts, 50 Hz
Current: - 0.5 Amps
Speed: - Min = 0 rpm ,Max = 9500 rpm
TEFC Construction, Commentator Motor.

3.3 Design of Belt Drive
Power is transmitted from the motor shaft to the input shaft of drive by means of an open belt drive,
Motor pulley diameter = 20 mm
Input shaft pulley diameter = 100 mm
Reduction ratio = 5
Input shaft speed = 9500/5 = 1900 rpm
T motor = 0.05 Nm
Torque at Input shaft = 5 x 0.05 = 0.25 Nm

3.3.1 Design of Open Belt Drive
Motor pulley diameter = 20 mm
Input shaft pulley diameter = 110 mm
Reduction ratio = 5
Coefficient of friction = 0.23
Maximum allowable tension in belt = 200 N
Center distance = 120
Wrap angle of pulley
\[ \alpha = 180 - 2\sin^{-1}\left\{\frac{(110-20)}{2 \times 120}\right\} \]
\[ \alpha = 136^0 \]
\[ \alpha = 2.37^c \]
Now,
\[ e^{\mu \alpha \sin(\theta/2)} = e^{0.2 \times 2.37 \sin(40/2)} = 4 \]
Width \( (b_2) \) at base is given by
\[ b_2 = 6-2(4 \tan 20) = 3.1 \]
Area of cross section of belt = \( \frac{1}{2}(6 + 3.1) \times 4 \)
\[ A = 18.2 \text{ mm}^2 \]
Now mass of belt /m length = 0.23 kg/m
\( V = \Pi DN / (60 \times 1000) = 4.188 \text{m/sec} \)
\( Tc = m \cdot V^2 \)
\( Tc = 4.034 \text{N} \)
\( T_1 = \text{Maximum tension in belt} - Tc \)
\( T_1 = 195.966 = 196 \text{ N} \)
\( T_1 / T_2 = e^{\mu \alpha \sin(\theta/2)} = 4 \)
\( T_2 = 49 \text{ N} \)

3.3.2 Result
Tension in tight side of belt \( (T_1) = 196 \text{ N} \)
Tension in slack side of belt \( (T_2) = 49 \text{ N} \)

3.4 Design of Input Shaft
3.4.1 Material Selection: Ref - PSG

(1.10 & 1.12) + (1.17)

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile Strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800</td>
</tr>
</tbody>
</table>

Table 3.4.1: Material selection of input shaft

\[ f_{s_{\text{max}}} = \frac{\text{UTS}}{\text{FOS}} = \frac{800}{2} = 400 \text{ N/mm}^2 \]

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft.

\[ Te = \frac{\pi f_s d^3}{16} \]

\[ f_{s_{\text{act}}} = \frac{16 \times 0.25 \times 10^3}{\pi x 16^3} \]

\[ f_{b_{\text{act}}} = 0.310 \text{ N/mm}^2 \]

As; \( f_{s_{\text{act}}} < f_{s_{\text{all}}} \)

Input is safe under torsional load.

3.4.1 Ansys Model
Fig 3.4.1: Ansys model of input shaft
### 3.4.2 Result & discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm$^2$</th>
<th>Von-mises stress N/m$^2$</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Shaft</td>
<td>0.310</td>
<td>0.58</td>
<td>0.00018</td>
<td>safe</td>
</tr>
</tbody>
</table>

Table 3.4.2: Result table for input shaft

### 3.4.3 Conclusion

a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the input shaft is safe.

b) Shaft shows negligible deformation.

### 3.5 Design of Input Coupler Body

#### 3.5.1 Material Selection

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength N/mm$^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>400</td>
</tr>
</tbody>
</table>

Table 3.5.1: Material selection of input coupler body

\[
\text{fs}_{\text{max}} = \frac{\text{UTS}}{FOS} = \frac{400}{2} = 200\text{N/mm}^2
\]

Check for torsional shear failure:

\[
T = \pi x \text{fs}_{\text{act}} x \frac{D_o^4}{16} - \frac{D_i^4}{16}
\]

\[
D = 25 \times 10^3 = \pi x \text{fs}_{\text{act}} x 22.5^4 - 16^4
\]

\[
\text{fs}_{\text{act}} = 0.15\text{N/mm}^2
\]

As; fs$_{\text{act}}$$<$fs$_{\text{all}}$

Input coupler body is safe under torsional load.

#### 3.5.2 Ansys Model
Fig 3.5.2: Ansys model of input coupler body
3.5.3 Result & Discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm$^2$</th>
<th>Von - mises stress N/m m$^2$</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Coupler Body</td>
<td>0.15</td>
<td>0.09</td>
<td>9.06E-6</td>
<td>Safe</td>
</tr>
</tbody>
</table>

Table 3.5.3: Result table for input coupler body

3.5.4 Conclusion.

a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the input coupler body is safe.

b) Input coupler body shows negligible deformation.
3.6 Design of Input Coupler Ring

Fig 3.6: Design of input coupler ring

3.6.1 Material selection.

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800</td>
<td>680</td>
</tr>
</tbody>
</table>

Table 3.6.1: material selection for input coupler ring

\[ fs_{\text{max}} = 400 \text{N/mm}^2 \]

Check for torsional shear failure:

\[ T = \frac{\pi}{16} \times \frac{f_{\text{s act}}}{D_o^4 - D_i^4} \times D_o \]

\[ 0.25 \times 10^3 = \frac{\pi}{16} \times f_{\text{s act}} \times 88^4 - 73^4 \]

\[ 0.25 \times 10^3 = 16 \times \frac{f_{\text{s act}}}{88} \]

\[ f_{\text{s act}} = 0.0035 \text{mm}^2 \]

As; \( f_{\text{s act}} < f_{\text{s all}} \)

Input Coupler ring is safe under torsional load

3.6.2 Ansys Model

Geometry

Input Coupler ring is safe under torsional load
Fig 3.6.2: Ansys model of input coupler ring
### 3.6.3 Result & discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm²</th>
<th>Von-mises stress N/m²</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input Coupler Ring</td>
<td>0.0035</td>
<td>0.013</td>
<td>1.045E-6</td>
<td>safe</td>
</tr>
</tbody>
</table>

Table 3.6.3: Result table for input coupler ring

### 3.6.4 Conclusion.

a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the input coupler ring is safe.

b) Input coupler ring shows negligible deformation.

### 3.7 Selection of Ball Bearing for Input Shaft

Selection of bearing 6004 ZZ

The input shaft is held in two ball bearings that equally share the radial load on the shaft. Selecting; Single Row deep groove ball bearing as follows.

Series 60

<table>
<thead>
<tr>
<th>IsI No</th>
<th>Bearing of basic design No (SK F)</th>
<th>D1</th>
<th>D2</th>
<th>D3</th>
<th>D4</th>
<th>B</th>
<th>Basic capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>2A</td>
<td>600</td>
<td>2</td>
<td>2</td>
<td>4</td>
<td>3</td>
<td>1</td>
<td>45</td>
</tr>
<tr>
<td>CO</td>
<td>4</td>
<td>0</td>
<td>3</td>
<td>2</td>
<td>6</td>
<td>2</td>
<td>00</td>
</tr>
</tbody>
</table>

Table 3.7.1: Bearing data (6004)

P = X F_r + Y F_a

Neglecting self-weight of carrier and gear assembly

For our application F_a=0

P = X F_r

Where F_r=Pt = T1+T2 = 196 +49 =245 N

Max radial load = F_r =245 N.

P= 145 N

Calculation dynamic load capacity of bearing.

L = ( C / P )^p , where p= 3 for ball bearings.

For m/c used for eight hr of service per day;

L_H = 4000- 8000hr

But;   L = 60 n L_H /10^6
L = 60 x 1900 x 4000 / 10^6 mrev .... here speed of shaft is considered to be 1900 rpm
L = 456
Now; 456 = (C)^3
(145)^3
C = 1885 N
As the required dynamic capacity of bearing is less than the rated dynamic capacity of bearing:
Bearing is safe.

### 3.8 Design of Input Coupler Female Liner

![Design of Input Coupler Female Liner](image)

#### 3.8.1 Material Selection

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile</th>
<th>Yield Strength</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800 N/mm²</td>
<td>680 N/mm²</td>
</tr>
</tbody>
</table>

Table 3.8.1: Material selection of input coupler female liner

fs_max = 400 N/mm²

Check for torsional shear failure:

\[
T = \pi \times \frac{f_{s, act} \times D_0^4 - D_i^4}{16 \times D_0^4 - D_i^4}
\]

\[
0.25 \times 10^3 = \pi \times \frac{f_{s, act} \times 65^4 - 57^4}{16 \times 65^4 - 57^4}
\]

fs_{act} = 0.0113 N/mm²
As; fs_{act} < fs_{all}
Input Coupler female liner is safe under torsional load.

### 3.8.2 Ansys Model

![Ansys Model](image)
Fig 3.8.2: Ansys model of input coupler female liner

### 3.8.3 Result & Discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm$^2$</th>
<th>Von-mises stress N/m$^2$</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input coupler female liner</td>
<td>0.0113</td>
<td>0.40</td>
<td>1.045E-6</td>
<td>Safe</td>
</tr>
</tbody>
</table>

Table 3.8.3: Result table for input coupler female liner

### 3.8.4 Conclusion

a) Maximum stress by theoretical method and Von-mises stress are
well below the allowable limit, hence the input coupler ring is safe. 

b) Input coupler ring shows negligible deformation 

3.9 Design of Coupler Pin

Fig. 3.9: Design of coupler pin

3.9.1 Material Selection: -Ref: - PSG (1.10 & 1.12) + (1.17) 

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800</td>
<td>680</td>
</tr>
</tbody>
</table>

Table 3.9.1: Material selection of coupler pin

\[ f_s_{\text{act}} = 16 \times 0.25 \times 10^3 \frac{\pi}{8^3} \]

\[ f_b_{\text{act}} = 2.4860 \text{ N/mm}^2 \]

As; \( f_s_{\text{act}} < f_s_{\text{all}} \)

Coupler pin is safe under torsional load.

3.9.2 Ansys Model

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft 

\[ T_e = \pi f_s d^3 \]
### Table 3.9.3: Result table for coupler pin

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm²</th>
<th>Von-mises stress N/mm²</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coupler pin</td>
<td>2.486</td>
<td>5.02</td>
<td>0.0011</td>
<td>Safe</td>
</tr>
</tbody>
</table>

#### 3.9.4 Conclusion

a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the coupler pin is safe.

b) Coupler pin shows negligible deformation.

---

**Fig 3.9.2: Ansys model of coupler pin**

**3.9.3 Results & Discussion**
3.10 Design of Output Shaft

![Design of output shaft](image)

**3.10.1 Material selection**

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile Strength N/mm²</th>
<th>Yield Strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800</td>
<td>680</td>
</tr>
</tbody>
</table>

Table: 3.10.1: Material selection for output shaft

\[
fs_{\text{max}} = \frac{UTS}{FOS} = \frac{800}{2} = 400 \text{ N/mm}^2
\]

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft

\[
Te = \frac{\Pi}{16} fs d^3
\]

\[
fs_{\text{act}} = 16 \times 0.25 \times 10^{-3} \times \frac{\Pi \times 16^3}{16} = 0.310 \text{ N/mm}^2
\]

As; \( fs_{\text{act}} < fs_{\text{all}} \)

Output is safe under torsional load.

3.10.2 Ansys Model

![Ansys Model](image)
Table 3.10.2: Result table for output shaft

3.10.4 Conclusion.

a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the output shaft is safe.

b) Output Shaft shows negligible deformation.

3.11 Design of Output Coupler Body
### 3.11.1 Material selection

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum</td>
<td>400</td>
<td>280</td>
</tr>
</tbody>
</table>

Table 3.11.1: Material selection for output coupler body

\[
fs_{\text{max}} = \frac{\text{UTS/FOS}}{2} = \frac{400}{2} = 200 \text{N/mm}^2
\]

Check for torsional shear failure:

\[
T = \frac{\pi x fs_{\text{act}} x Do^4 - Di^4}{16} - \left( \frac{22.5}{16} \right)
\]

\[
0.25 \times 10^3 = \frac{\pi x fs_{\text{act}} x 22.5^4 - 16^4}{16}
\]

\[fs_{\text{act}} = 0.15 \text{N/mm}^2\]

As \(fs_{\text{act}} < fs_{\text{all}}\)

Output coupler body is safe under torsional load.

### 3.11.2 Ansys Model
Fig 3.11.2: Ansys model of output coupler body

### 3.11.3 Result & Discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm²</th>
<th>Von-mises stress N/mm²</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output Coupler Body</td>
<td>0.15</td>
<td>0.098</td>
<td>9.06E-6</td>
<td>safe</td>
</tr>
</tbody>
</table>

Table 3.11.3: Result table for output coupler body

**3.11.4 Conclusion.**

a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the output coupler body is safe.

b) Output coupler body shows negligible deformation.

### 3.12 Design of Output Coupler Ring

#### 3.12.1 Material selection.

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800</td>
<td>680</td>
</tr>
</tbody>
</table>

Table 3.12.1: Material selection for output coupler ring

\[ f_s \text{ max} = 400 \text{N/mm}^2 \]

Check for torsional shear failure:

\[ T = \frac{\Pi \times f_s \text{ act} \times D_o^4 - D_i^4}{16 D_o} \]
\[ 0.25 \times 10^3 = \Pi \times f_{s_{\text{act}}} \times 88^4 - 73^4 \]
\[ 1688 \]
\[ f_{s_{\text{act}}} = 0.0035 \text{ N/mm}^2 \]
As; \( f_{s_{\text{act}}} < f_{s_{\text{all}}} \)
Output coupler ring is safe under torsional load.

### 3.12.2 Ansys Model

![Ansys Model](image)

Fig 3.12.2: Ansys model of output coupler ring

### 3.12.3 Result & Discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm²</th>
<th>Von-mises stress N/mm²</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Output Coupler Ring

Table 3.12.3: Result table for output coupler ring

<table>
<thead>
<tr>
<th>Safe No</th>
<th>Bearing of</th>
<th>D1</th>
<th>D2</th>
<th>D3</th>
<th>B</th>
<th>Basic capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>2A</td>
<td>600</td>
<td>2</td>
<td>2</td>
<td>4</td>
<td>3</td>
<td>45</td>
</tr>
<tr>
<td>C0</td>
<td>4</td>
<td>0</td>
<td>3</td>
<td>6</td>
<td>2</td>
<td>00</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>50</td>
</tr>
</tbody>
</table>

3.12.4 Conclusion.

a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the output coupler ring is safe.

b) Output coupler ring shows negligible deformation.

3.13 Selection of Ball Bearing for Output Shaft

Selection of Bearing 6004 ZZ

The input shaft is held in two ball bearings that equally share the radial load on the shaft.

Selecting: Single Row deep groove ball bearing as follows.

Series 60

Neglecting self-weight of carrier and gear assembly

For our application \( F_a = 0 \)

\[ P = X F_r + Y F_a \]

\( P = X F_r \)

Where \( F_r = P_t = T1 + T2 = 196 + 49 = 245 \) N

Max radial load = \( F_r = 245 \) N.

\( P = 145 \) N

Calculation dynamic load capacity of bearing.

\[ L = \left( \frac{C}{P} \right)^p \]

Where \( p = 3 \) for ball bearings

For m/c used for eight hr of service per day;

\( L_{H} = 4000 - 8000 \) hr

But;

\[ L = \frac{60 n L_{H}}{10^6} \]
L = 60 x 1900 x 4000 / 10^6 mrev,... here speed of shaft is considered to be 1900 rpm
L = 456
Now: 456 = (C)^3
(145)^3
C = 1885N
As the required dynamic capacity of bearing is less than the rated dynamic capacity of bearing;
Bearing is safe.

### 3.14 Design of Output Coupler Female Liner


<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength</th>
<th>Yield strength</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24</td>
<td>800</td>
<td>680</td>
</tr>
</tbody>
</table>

Table 3.14.1: Material selection for output coupler female liner

fs_{max} = 400N/mm^2

Check for torsional shear failure:

\[
T = \frac{\pi \times f_{s_{act}} \times \frac{D_o}{16} - \frac{D_i}{4}}{D_o - \frac{D_i}{4}}
\]

0.25 \times 10^3 = \frac{\pi \times f_{s_{act}} \times 0.65^4 - 0.57^4}{16}

fs_{act} = 0.0113N/mm^2

As; fs_{act} < fs_{all}

Output Coupler female liner is safe under torsional load.

#### 3.14.2 Ansys model

Fig 3.14: Design of output coupler female liner
Fig 3.14.2: Ansys model of output coupler female liner

### 3.14.3 Result & Discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum theoretical stress N/mm²</th>
<th>Von Mises stress N/m²</th>
<th>Total deformation mm</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output coupler female liner</td>
<td>0.0113</td>
<td>0.40</td>
<td>1.045E-6</td>
<td>Safe</td>
</tr>
</tbody>
</table>

Table 3.14.3: Result table for output coupler female liner

#### 3.14.4 Conclusion
a) Maximum stress by theoretical method and Von-mises stress are well below the allowable limit, hence the output coupler female liner is safe.

b) Output coupler female liner shows negligible deformation.

15 Design of Trunion Holder

![Design of trunion holder](image)

3.15.1 Material selection.

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>400</td>
<td>280</td>
</tr>
</tbody>
</table>

Table 3.15.1: Material selection for trunion holder

fs_{\text{max}} = \frac{\text{UTS/FOS}}{2} = \frac{400}{2} = 200\text{N/mm}²

Check for torsional shear failure:

\[
T = \pi \times f_{s_{\text{act}}} \times D_o \left( \frac{D_i}{4} \right)^4
\]

\[
= \frac{0.25 \times 10^3 \times \pi \times f_{s_{\text{act}}} \times 36.4^4 - 23^4}{16}
\]

36.4

fs_{\text{act}} = 0.2 \text{N/mm}²

As; f_{s_{\text{act}}} < f_{s_{\text{all}}}

Trunion holder is safe under torsional load.

3.15.2 Ansys Model

![Ansys Model](image)
3.15.2: Ansys model of trunion holder

3.15.3 Result & Discussion

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Maximum stress by theoretical method</th>
<th>Von-mises stress</th>
<th>Total deformation</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Trunion holder</td>
<td>0.2</td>
<td>0.9</td>
<td>0.000234</td>
<td>Safe</td>
</tr>
</tbody>
</table>

Table 3.15.3: Result table for trunion holder

3.15.4 Conclusion

a) Maximum stress by theoretical method and Von-mises stress are
well below the allowable limit, hence the trunion holder is safe.

b) Trunion holder shows negligible deformation

4 EXPERIMENTAL VALIDATION

4.1 Experimental Setup

![Diagram of three pin constant velocity joint]

Figure 4.1.1: setup for three pin constant velocity joint

4.2 Test & Trial

4.2.1 Coupling Bronze Trunion:

Parallel Offset: 12mm

Aim: To conduct trial and plot
a) Torque vs. Speed Characteristics
b) Power vs. Speed Characteristics

Arrangement: In order to conduct trial, a dynamobrake pulley cord, weight pan are provided on the output shaft.

Procedure:

a) Start motor.

b) Let mechanism run & stabilize at certain speed (say 1500 rpm).

c) Place the pulley cord on dynamobrake pulley and add 0.1 Kg weight into the pan, note down the output speed for this load by means of tachometer.

d) Add another 0.1 Kg cut & take reading.

e) Tabulate the readings in the observation table.

f) Plot Torque vs. speed characteristic.

g) Plot Power vs. speed characteristic.

![Diagram of experimental setup for parallel offset]

Fig 4.2.1: Experimental setup for parallel offset.

Observation Table
Table 4.2.1.1: Observation table for parallel offset

Sample Calculations:- (At .8 Kg Load)

Average speed :-

\[ N = \frac{N_1 + N_2}{2} = \frac{1210 + 1190}{2} = 1200 \text{rpm} \]

Output Torque:-

\[ T_{dp} = \text{Weight in pan} \times \text{Radius of Dynobrake Pulley} \]

\[ = (0.8 \times 9.81) \times 25 = 196.2 \text{ N.mm} \]

\[ T_{dp} = 0.1962 \text{ N.m} \]

Input Power:- \( (P_{i/p}) = 29.6 \text{ Watt} \)

Output Power:- \( (P_{o/p}) \)

\[ P_{o/p} = \frac{2 \pi N T_{o/p}}{60} \]

\[ = \frac{2 \times \pi \times 0.1962 \times 1200}{60} \]

\[ P_{o/p} = 24.6 \text{ watt} \]

Efficiency:-

\[ \eta = \frac{\text{Output power}}{\text{Input power}} \]

\[ = \frac{24.6}{29.6} \]

\[ \eta = 83.10\% \]

Efficiency of transmission of gear drive at 0.8 kg load= 83.10\%
Result table

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>Load (kg)</th>
<th>Speed (rpm)</th>
<th>Torque (N.m)</th>
<th>Power (watt)</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.2</td>
<td>47</td>
<td>0.04</td>
<td>7.551</td>
<td>25.512</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>905</td>
<td>0.09</td>
<td>14.43</td>
<td>48.768</td>
</tr>
<tr>
<td></td>
<td>0.6</td>
<td>0.14</td>
<td>20.49</td>
<td>69.247</td>
<td>66</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>0.19</td>
<td>24.65</td>
<td>83.305</td>
<td>46</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
<td>0.24</td>
<td>24.14</td>
<td>81.569</td>
<td>93</td>
</tr>
</tbody>
</table>

Table 4.2.1.2: Result table for parallel offset

Characteristics Plots

Torque vs Speed

Graph shows that torque increases with decrease in output speed of coupling.

Power vs Speed

Graph shows that maximum power is delivered by the coupling at close to 1200 rpm. Thus, this is recommended speed at maximum parallel offset condition.

Efficiency vs Speed

Graph shows that maximum efficiency is attained by the coupling at close to 1200 rpm. Thus, this is recommended speed at maximum parallel offset condition for maximum efficiency.
4.2.2 EN-24 Trunion: Angular Offset: 14 Degree Maximum

Aim: To conduct trial and plot
a) Torque vs. Speed Characteristics
b) Power vs. Speed Characteristics

Arrangement: In order to conduct trial, a dynobrake pulley cord, weight pan are provided on the output shaft.

Procedure:
a) start motor
b) Let mechanism run & stabilize at certain speed (say 1500 rpm)
c) Place the pulley cord on dynobrake pulley and add 0.1 Kg weight into, the pan, note down the output speed for this load by means of tachometer.
d) Add another 0.2 Kg cut & take reading.
e) Tabulate the readings in the observation table
f) Plot Torque vs. speed characteristic
g) Power vs. speed characteristic.

![Experimental setup for angular offset.]

**Observation Table**

<table>
<thead>
<tr>
<th>Sr. NO</th>
<th>Loading Weight (Kg)</th>
<th>Loading Speed rpm</th>
<th>Unloading Weight (Kg)</th>
<th>Unloading Speed rpm</th>
<th>Mean Speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.2</td>
<td>144</td>
<td>2</td>
<td>142</td>
<td>143</td>
</tr>
<tr>
<td>2</td>
<td>0.4</td>
<td>132</td>
<td>4</td>
<td>131</td>
<td>131.5</td>
</tr>
<tr>
<td>3</td>
<td>0.6</td>
<td>122</td>
<td>6</td>
<td>124</td>
<td>123.0</td>
</tr>
<tr>
<td>4</td>
<td>0.8</td>
<td>109</td>
<td>8</td>
<td>108</td>
<td>107.0</td>
</tr>
<tr>
<td>5</td>
<td>1.0</td>
<td>900</td>
<td>10</td>
<td>880</td>
<td>890</td>
</tr>
</tbody>
</table>
Table 4.2.2.1: observation table for angular offset

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>Load (kg)</th>
<th>Speed (rpm)</th>
<th>Torque (N.m)</th>
<th>Power (watt)</th>
<th>Efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>143</td>
<td>0.04</td>
<td>7.346</td>
<td>153</td>
<td>24.818</td>
</tr>
<tr>
<td>0.4</td>
<td>131</td>
<td>0.09</td>
<td>13.51</td>
<td>076</td>
<td>45.644</td>
</tr>
<tr>
<td>0.6</td>
<td>123</td>
<td>0.14</td>
<td>18.95</td>
<td>616</td>
<td>64.041</td>
</tr>
<tr>
<td>0.8</td>
<td>107</td>
<td>0.19</td>
<td>21.98</td>
<td>709</td>
<td>74.280</td>
</tr>
<tr>
<td>1.0</td>
<td>890</td>
<td>0.24</td>
<td>22.86</td>
<td>041</td>
<td>77.231</td>
</tr>
</tbody>
</table>

Graph shows that maximum power is delivered by the coupling at close to 900 rpm. Thus this is recommended speed at maximum angular offset condition.

Characteristics Plots

Torque vs Speed

Graph shows that torque increases with decrease in output speed of coupling.

Efficiency vs Speed

Graph shows that maximum efficiency is attained by the coupling at close to 900 rpm. Thus this is recommended speed at maximum angular offset condition for maximum efficiency.
## 5 COST ANALYSIS

### 5.1 Bill of Materials

<table>
<thead>
<tr>
<th>Sr No.</th>
<th>Part Code</th>
<th>Description</th>
<th>Quantity</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>TCVJ –1</td>
<td>Frame</td>
<td>01</td>
<td>MS</td>
</tr>
<tr>
<td>2</td>
<td>TCVJ –2</td>
<td>Bearing Housing LPlate</td>
<td>02</td>
<td>EN9</td>
</tr>
<tr>
<td>3</td>
<td>TCVJ –3</td>
<td>Bearing Housing</td>
<td>01</td>
<td>EN9</td>
</tr>
<tr>
<td>4</td>
<td>TCVJ –4</td>
<td>Main Pulley</td>
<td>01</td>
<td>EN9</td>
</tr>
<tr>
<td>5</td>
<td>TCVJ –5</td>
<td>Input Shaft</td>
<td>01</td>
<td>EN24</td>
</tr>
<tr>
<td>6</td>
<td>TCVJ –6</td>
<td>Output Shaft</td>
<td>01</td>
<td>EN24</td>
</tr>
<tr>
<td>7</td>
<td>TCVJ –7</td>
<td>Coupler Body</td>
<td>01</td>
<td>AL</td>
</tr>
<tr>
<td>8</td>
<td>TCVJ –8</td>
<td>Coupling Female Liner</td>
<td>02</td>
<td>EN24</td>
</tr>
<tr>
<td>9</td>
<td>TCVJ –9</td>
<td>Constrain Ring</td>
<td>02</td>
<td>EN24</td>
</tr>
<tr>
<td>10</td>
<td>TCVJ –10</td>
<td>Coupler Ring</td>
<td>02</td>
<td>EN24</td>
</tr>
<tr>
<td>11</td>
<td>TCVJ –11</td>
<td>Trunion Holder</td>
<td>01 AL</td>
<td>EACH</td>
</tr>
<tr>
<td>12</td>
<td>TCVJ –12</td>
<td>Trunion</td>
<td>03</td>
<td>BRASS/BRONZE/EN2</td>
</tr>
<tr>
<td>13</td>
<td>TCVJ –13</td>
<td>Slide Bar</td>
<td>02 EN9</td>
<td>Hole</td>
</tr>
<tr>
<td>14</td>
<td>TCVJ –14</td>
<td>Slide Nut</td>
<td>02 EN9</td>
<td>9 No’s</td>
</tr>
<tr>
<td>15</td>
<td>TCVJ –15</td>
<td>Clamp Plate</td>
<td>01 EN9</td>
<td>Rs/Hole</td>
</tr>
<tr>
<td>16</td>
<td>TCVJ –16</td>
<td>Motor Plate</td>
<td>01 MS</td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>TCVJ –17</td>
<td>Dyno Brake Pulley</td>
<td>01 CI</td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>TCVJ –18</td>
<td>Bolt Rest</td>
<td>02 EN9</td>
<td></td>
</tr>
<tr>
<td>19</td>
<td>TCVJ –19</td>
<td>Motor</td>
<td>01 STD</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>TCVJ –20</td>
<td>Belt(6 X 600)</td>
<td>01 STD</td>
<td></td>
</tr>
<tr>
<td>21</td>
<td>TCVJ –21</td>
<td>Grub Screw M8 X 8</td>
<td>03 STD</td>
<td></td>
</tr>
</tbody>
</table>

### 5.2 Raw Material Cost

The total raw material cost as per the individual materials and their corresponding rates per kg is as follows:

Total raw material cost = Rs. 4150/-

### 5.3 Machining Cost

<table>
<thead>
<tr>
<th>Operation</th>
<th>Rate (Rs/Hr)</th>
<th>Time (Hrs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lathe</td>
<td>90</td>
<td>25</td>
</tr>
<tr>
<td>Milling</td>
<td>105</td>
<td>16</td>
</tr>
<tr>
<td>Jig Boring</td>
<td>90/Hole 9 No's</td>
<td>25</td>
</tr>
<tr>
<td>Drilling</td>
<td>50</td>
<td>4.2</td>
</tr>
<tr>
<td>Tapping</td>
<td>5 Rs/Hole 18</td>
<td>18</td>
</tr>
</tbody>
</table>

Table 6.3: Machining cost table

Total machining cost = Rs. 11990/-

---

**Table 5.1: Material bills table**

**Table 6.3: Machining cost table**
5.4 Miscellaneous Costs

Table 5.4: Miscellaneous cost table

<table>
<thead>
<tr>
<th>Description</th>
<th>Quan.</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor 01</td>
<td>1</td>
<td>1150</td>
</tr>
<tr>
<td>Srdg Brg 6004zz</td>
<td>02</td>
<td>500</td>
</tr>
<tr>
<td>Dyno Brake Pulley</td>
<td>01</td>
<td>210</td>
</tr>
<tr>
<td>Belt</td>
<td>01</td>
<td>170</td>
</tr>
<tr>
<td>Circlips 09</td>
<td>1</td>
<td>130</td>
</tr>
<tr>
<td>Grub Screw M8 03</td>
<td>1</td>
<td>12</td>
</tr>
<tr>
<td>Grub Screw M6 9</td>
<td>2</td>
<td>27</td>
</tr>
<tr>
<td>Bolts &amp; Nut</td>
<td>-</td>
<td>166</td>
</tr>
</tbody>
</table>

Hence the total cost of machine = Rs 13600/- approx.

5.5 Cost of Purchased Parts

Table 5.5: Table of costs of purchased parts

<table>
<thead>
<tr>
<th>Sr No.</th>
<th>Description</th>
<th>Quan.</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor</td>
<td></td>
<td>01</td>
<td>1150</td>
</tr>
<tr>
<td>Srdg Brg 6004zz</td>
<td>02</td>
<td>500</td>
<td></td>
</tr>
<tr>
<td>Dyno Brake Pulley</td>
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<td>210</td>
<td></td>
</tr>
<tr>
<td>Belt</td>
<td></td>
<td>01</td>
<td>170</td>
</tr>
<tr>
<td>Circlips</td>
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</tr>
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<td></td>
</tr>
<tr>
<td>Bolts &amp; Nut</td>
<td>-</td>
<td>166</td>
<td></td>
</tr>
</tbody>
</table>

The cost of purchase parts = Rs 2365/-

5.6 Total Cost

Total cost = Raw Material Cost + Machine Cost + Miscellaneous Cost

SCOPE FOR FUTURE WORK

a) It is replacement to all present velocity joints.
b) One can reduce the cost and space required so that it will easily penetrate in the market.
c) Its efficiency can be increased up to 95% by using antifriction material.
d) If there are any hely design, that could apply this mechanism.
e) It is remarkable device to be used in industries, plane, helicopters, trains, tractors etc.

CONCLUSION

Three pin constant velocity joint is replacement to all present velocity joint. One can reduce the cost and space required so that it will easily penetrate in the market. Its efficiency can be increased up to 95% with antifriction material. It has less vibrations and less friction, hence runs cool. It is remarkable device to be used in industries, plane, helicopters, trains,
tractors etc. From this project stage 1 on three pin constant velocity joint we will be able to conclude that it is a joint with higher parallel and angular misalignment capability and it can be preferred over universal joint.

Thus we have performed analysis on 3 pin constant velocity joint for parallel and angular power transmission. We have conducted trial on 3 pin constant velocity joint and recorded the readings. We have plotted performance characteristics of the joint such as torque vs speed, power vs speed and efficiency vs speed both for parallel and angular power transmission. From the trial we can conclude that the joint has better performance characteristics than universal joint.

REFERENCES