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Design, Development and Testing of Parallel offset Coupling with Angular offset

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Abstract- **Parallel and angular offset Couplings were developed to fill a gap in the family of torque- rigid couplings. Most couplings are designed to accommodate axial, angular, or parallel shaft displacements only. For some applications, however, the operational conditions require all possible shaft misalignments. If these shaft misalignments exceed the limit of the selected coupling capacity, excess side loads are introduced into the equipment which can cause vibrations, life reduction or failure of vital machine components such as bearings, motors, etc.**

The Parallel and angular offset Couplings are a modification of the Inline Coupling, designed to accommodate 5 degrees of angular shaft misalignment. This coupling allows easy adjustment to any possible misaligned shaft position without imposing heavy side loads on shafts, bearings or other machine equipment. This Couplings offer large shaft misalignment capabilities and constant angular velocity. The acting forces within the coupling can be precisely calculated, assuring a sound coupling design which is especially important for heavy-duty applications.

Key words: Parallel and angular offset coupling, Misalignment, axial load and Power Transmission.

I. INTRODUCTION

Shaft misalignment has major implications for modern-day rotating equipment reliability. Although effective alignment techniques have been applied successfully on a wide range of equipment for some time, deterioration of the alignment state can frequently occur due to, for example, changes in equipment operating conditions. Because of this rigid support, it is virtually impossible to avoid slight misalignments between a driving and driven shaft when they are connected. Restoring forces that occur as the two coupled shafts compete to maintain their original positions can put unwanted strain on shaft bearings, causing them to wear out prematurely. Additional axial loads are also placed on the bearings as thermal growth occurs in shafting during operation.

This situation can lead to the imposition of excessive forces on the equipment rotating and static elements, most commonly resulting in bearing or coupling failure. In extreme circumstances contact between rotating and stationary components can be expected to occur. The presence of shaft misalignment can greatly influence machinery vibration response. However, it’s detection through vibration diagnostics is not a straightforward matter due to the lack of a clear understanding of the physical mechanism relating shaft misalignment to vibration. multi-harmonic response from rotor dynamic systems subjected to angular and parallel misalignment by assuming coupling transmitted forces represented by a half-sinusoid function having fundamental frequency equal to twice rotational speed. Assumptions and investigated the transient response of a misaligned rotor system.

1.1 PARALLEL OFFSET MISALIGNMENT

Offset misalignment, sometimes referred to as parallel misalignment, is the distance between the shaft centers of rotation measured at the plane of power transmission. This is typically measured at the coupling center. The units for this measurement are mils (where 1 mil = 0.001 in.). A measure of the offset distance between the centerlines of driving and driven shafts. Coupling catalogs will show the maximum parallel misalignment tolerable in each coupling. A coupling should not be operated with both parallel and angular misalignment at their maximum values.

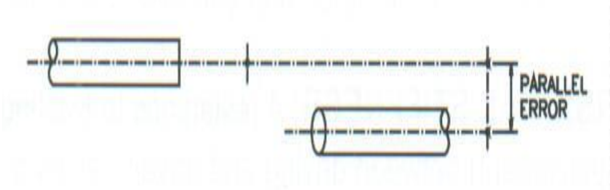

Fig 1.1 Parallel Offset

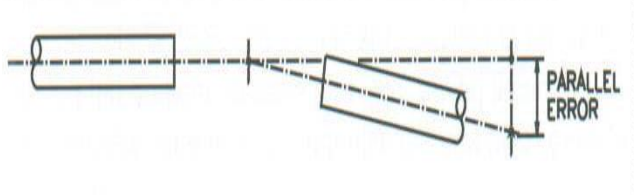

Fig 1.2 Angular Offset

II. EXPERIMENTAL ANALYSIS

The Couplings were developed to fill a gap in the family of torque-rigid couplings. Most couplings are designed to accommodate axial, angular, or parallel shaft displacements only. For some applications, however, the operational conditions require all possible shaft misalignments. If these shaft misalignments exceed the limit of the selected coupling capacity, excess side loads are introduced into the equipment which can cause vibrations, life reduction or failure of vital machine components such as bearings, motors, etc. The Couplings are a modification of the Inline Coupling, designed to accommodate 5 degrees of angular shaft misalignment. This coupling allows easy adjustment to any possible misaligned shaft position without imposing heavy side loads on shafts, bearings or other machine equipment. The Couplings offer large shaft misalignment capabilities and constant angular velocity. The acting forces within the coupling can be precisely calculated, assuring a sound coupling design which is especially important for heavy-duty applications. The experimental setup as shown in the figure-2.1.

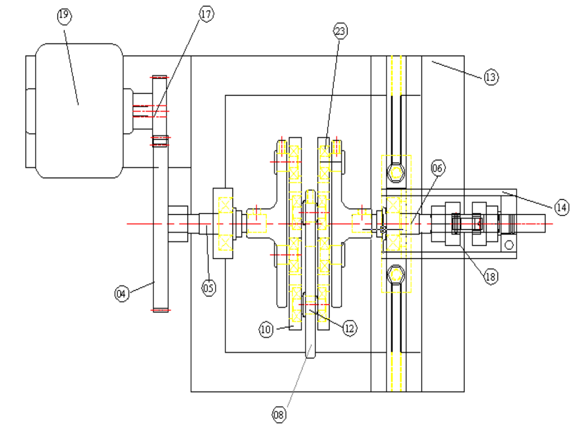

Fig-2.1 Experimental Setup

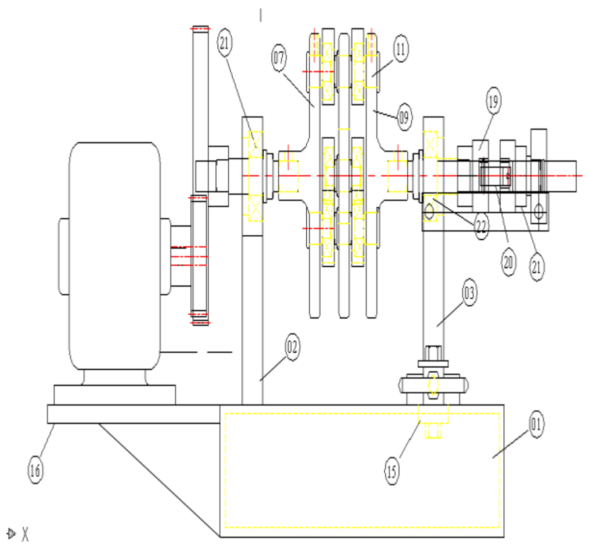

Table-2.1 Description of parts

Part no.	Description	Material
1.	FRAME	MS
2.	BRG_HSG_L_PLATE	EN9
3.	BRG_HSG	EN9
4.	MAIN PULLEY	EN9
5.	IP_SHAFT	EN24
6.	OP_SHAFT	EN24
7.	DRIVER _DISK	EN24
8.	INT_DISK	EN24
9.	DRIVEN_DISK	EN24
10.	LINKS	EN9
11.	D_D_PINS	EN24
12.	I_D_PINS	EN24
13.	SLIDE BAR	EN9
14.	SLIDE NUT	EN9
15.	CLAMP PLATE	EN9
16.	MOTOR PLATE	MS
17.	HANDLE	MS
18.	BOLT REST	EN9
19.	MOTOR	STD
20.	BELT(6 X 600)	STD
21.	Bearing 6204ZZ	STD
22.	Bearing 6203ZZ	STD
23.	Bearing 6200ZZ	STD
24.	Grub screw M8 x 8	STD
25.	Grub screw M6 x 8	STD
26.	HEX BOLT M8 x 25	STD
27.	HEX BOLT M10 x 30	STD
28.	HEX BOLT M10 x 50	STD
29.	HEX BOLT M10 x 200	STD

III. DESIGN OF COMPONENTS

3.1 SELECTION OF DRIVE MOTOR

The metric system uses kilowatts (kW) for driver ratings.

Converting kW to torque: $T = \frac{KW \times 84518}{RPM}$

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

$$0.3 = \frac{KW \times 9550}{1200}$$

$$kW = 0.038$$

Thus the minimum input power required will be 38 watt.
Thus selecting a drive motor as follows
DRIVE MOTOR
Type: - Single Phase Ac Motor.
Power: - $\frac{1}{15}$ Hp (50 Watts)
Voltage: - 230 Volts, 50 Hz

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Current: - 0.5 Amps
Speed: - Min = 0 Rpm
Max = 6000 Rpm

3.2 DESIGN OF GEAR DRIVE FROM MOTOR TO INPUT SHAFT

3.2.1DESIGN OF SPUR PINION & GEAR

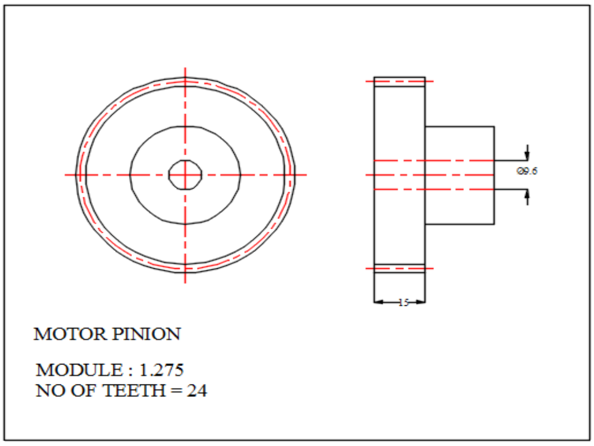


Fig 3.1 Spur Pinion

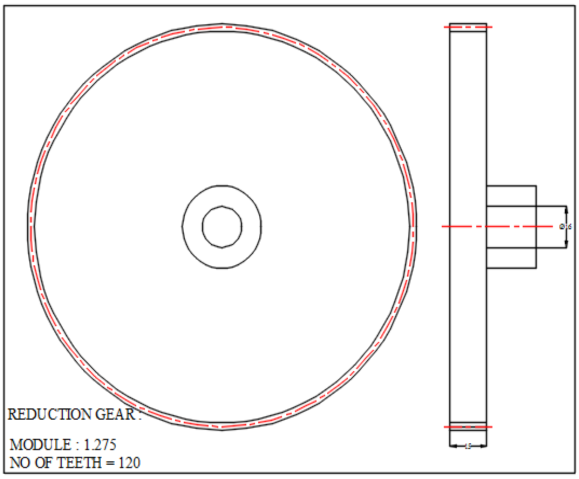


Fig3.2 Spur Gear

Stage: Drive as gear and pinion arrangement
Maximum load =Maximum Torque / Radius of gear
Maximum torque = 0.4 N-m
No of teeth on gear = 120
Module = 1.275mm

Radius of gear by geometry $= \frac{120 \times 1.275}{2} = 76.5mm$

Maximum load $= \frac{T}{r} = \frac{0.4 \times 10^3}{76.5} = 5.3N$

b = 10 m

Material of spur gear and pinion = Nylon-66
Sult pinion = Sult gear= 85 N/mm^2
Service factor (Cs) = 1.5
The gear and pinion arrangement where as pinion has 10 teeth and gear has 30 teeth share the entire tooth load…
Pt = (W x Cs) =8 N.
P eff = 8 N (as Cv =1 due to low speed of operation)

$\boxed{\text{P eff = 8 N}}$ ---------- (A)

Lewis Strength equation
WT = S b y m
Where;

$Y = 0.484 - \frac{2.86}{z}$

$Y_p = 0.484 - \frac{2.86}{24} = 0.058$

$Y_g = 0.484 - \frac{2.86}{120} = 0.460$

$S_{yp} = 4.930$
$S_{yg} = 39.10$
As $S_{yp} < S_{ys}$, Pinion is weaker

W_T = (Syp) x b x m
=4.93 m x m
W_T = 4.93 m^2---------- (B)
Equation (A) & (B)
4.93 m^2 =8
m=1.274mm
Selecting standard module = 1.275 mm, for ease of construction as we go for single stage gear box for making size compact and achieving maximum strength and proper mesh.

3.3 SELECTION OF INPUT SHAFT

Table-3.3 Stress Values from Data Book

Designation	Ultimate Tensile Strength N/Mm^2	Yield Strength N/Mm^2
En 24	800	680

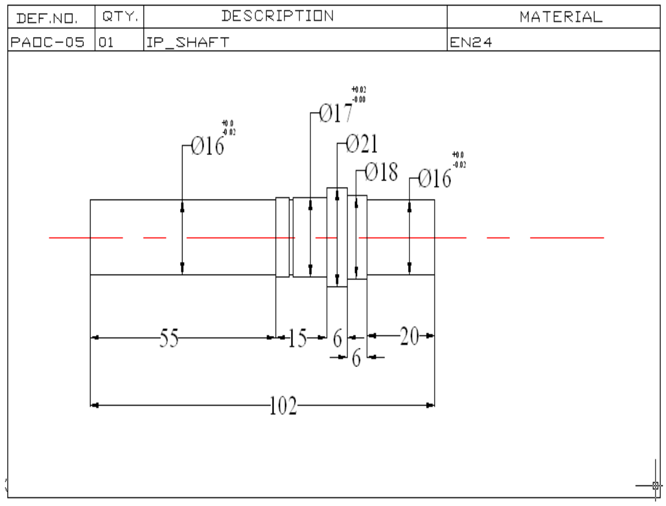


Fig-3.3 Input shaft

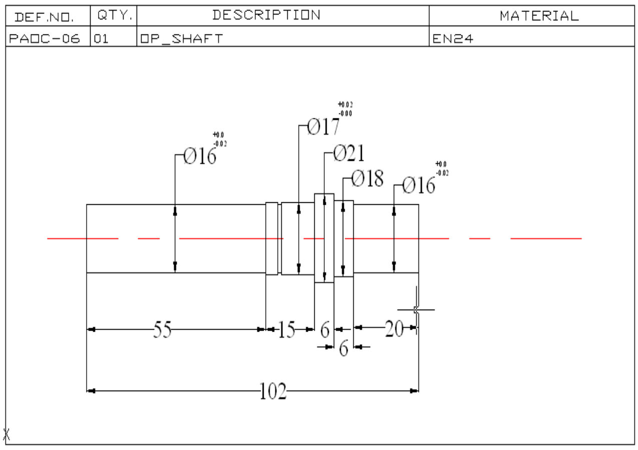


Fig-3.4 Output shaft

3.4 SELECTION OF DRIVER DISK HUB

Driver disk hub can be considered to be a hollow shaft subjected to torsional load.

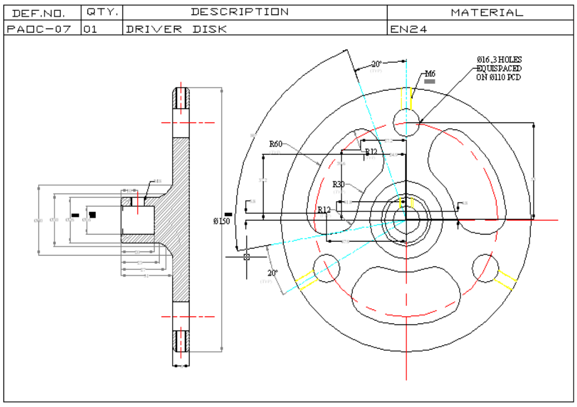


Fig 3.5 Driver Disk

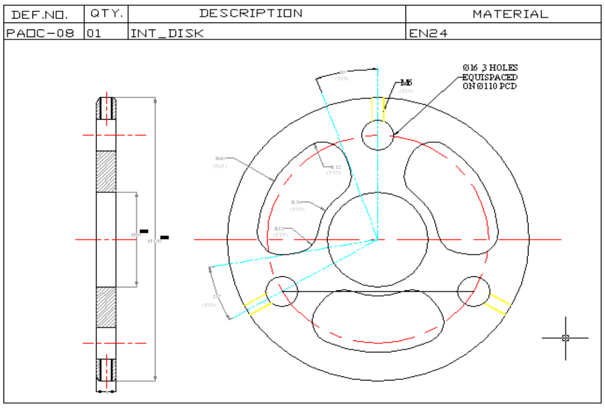


Fig 3.6 Intermediate Disk

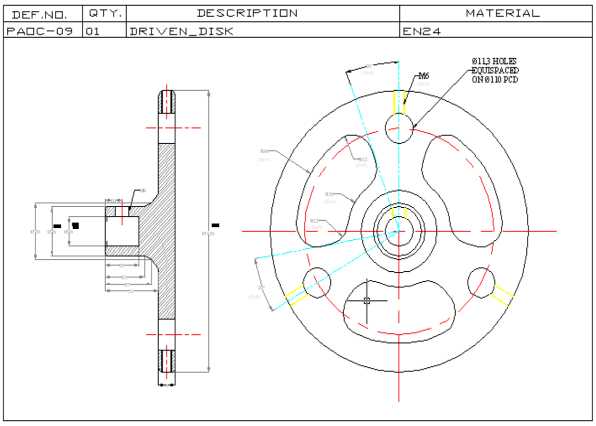


Fig-3.7 Driven Disk

3.5 SELECTION OF LINK

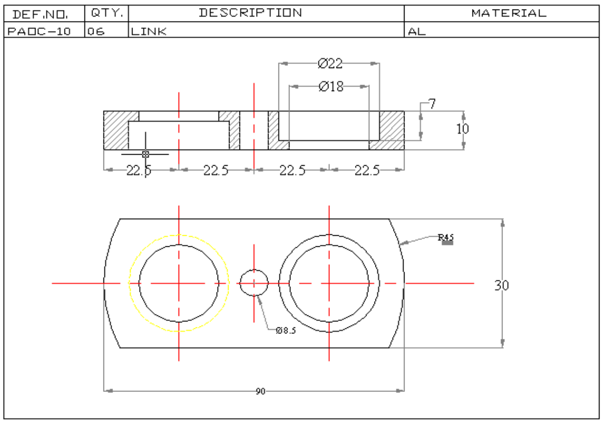


Fig-3.8 Link

3.6 MISCELLANEOUS PARTS DRAWING

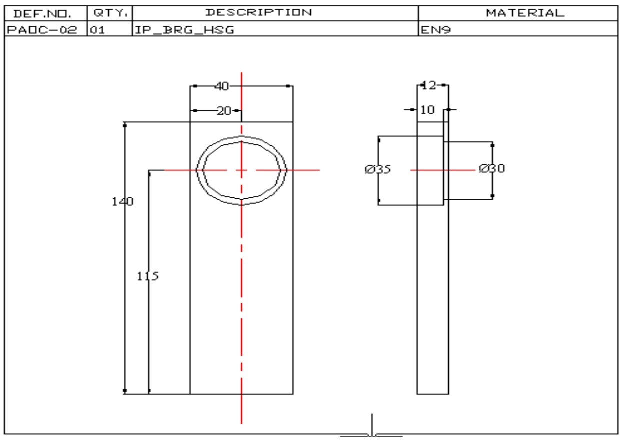


Fig-3.9 Housing of Input Bearing

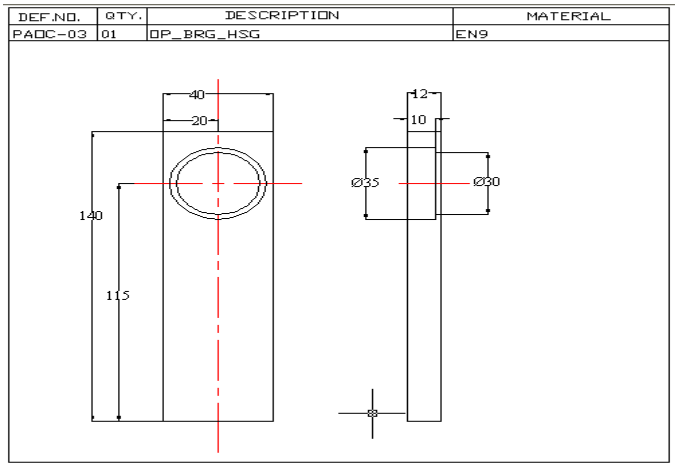


Fig 3.10 Housing of Output Bearing

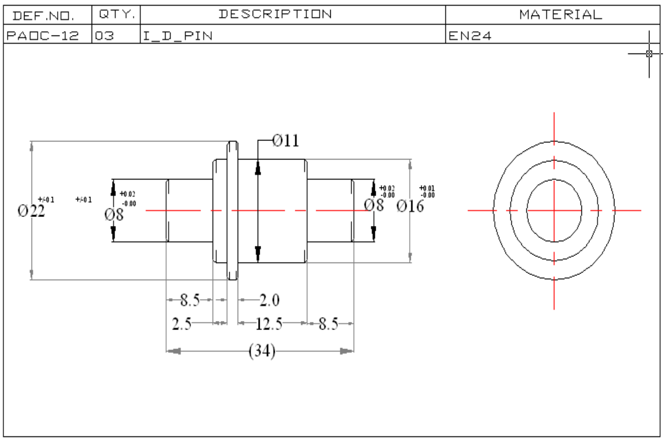


Fig-3.11 I D PIN

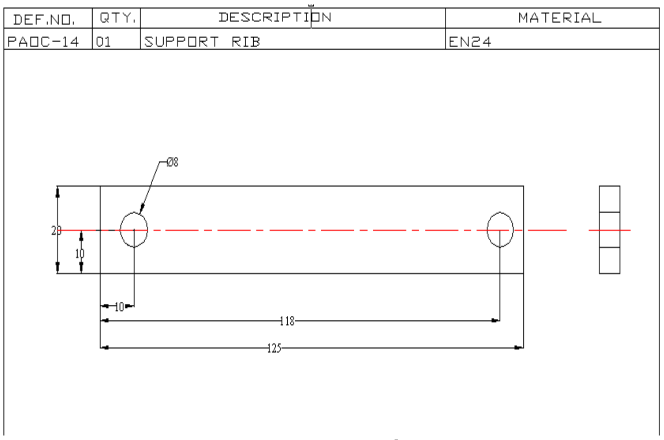


Fig 3.12 Support Rib

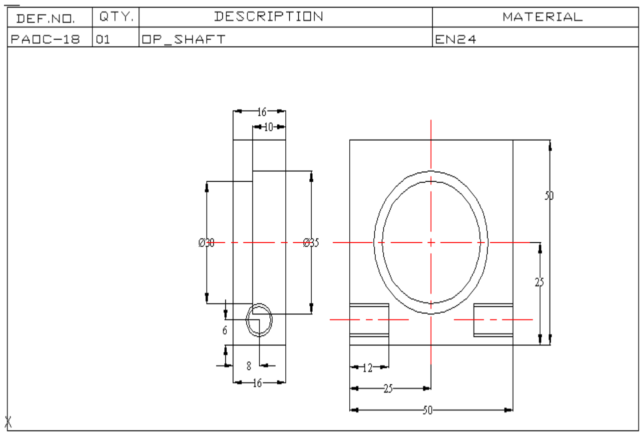


Fig 3.13 Output shaft

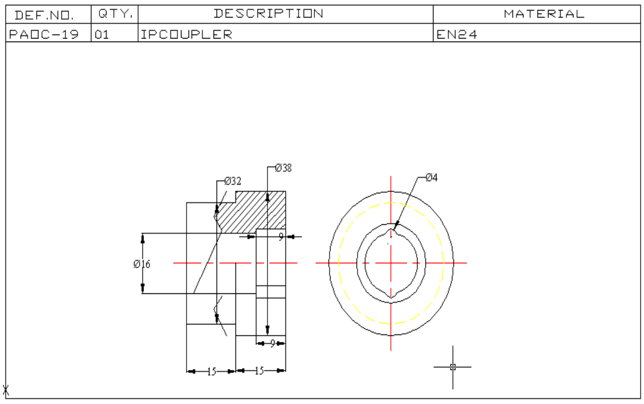


Fig-3.14 Input Coupler

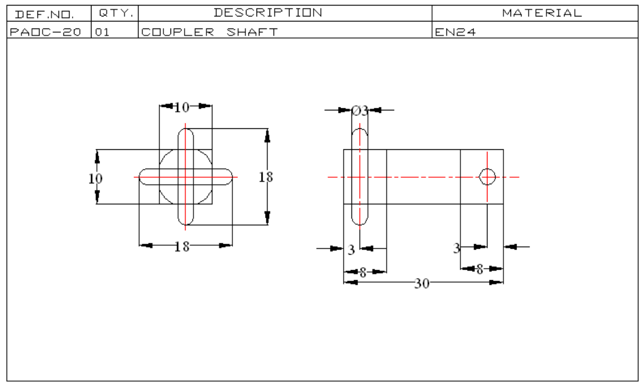


Fig-3.15 Coupler Shaft

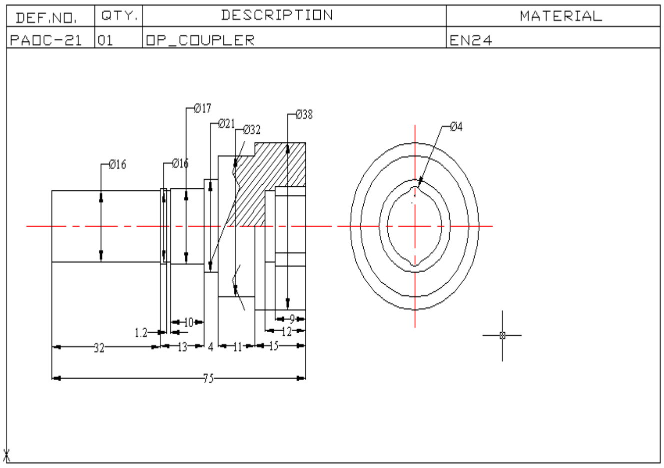


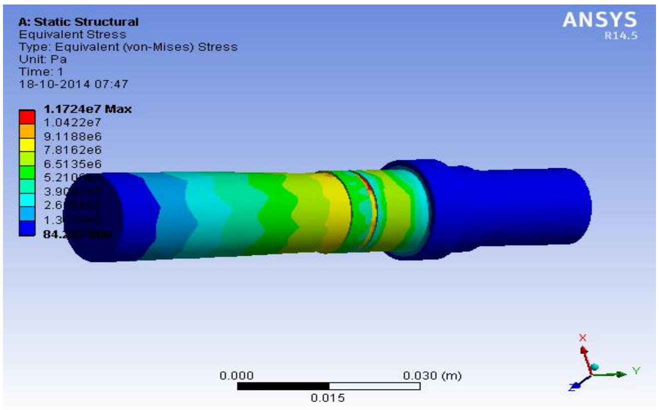
Fig-3.16 Output Coupler

IV. ANALYSIS OF COMPONENTS

4.1 ANALYSIS PROCEDURE

1. Modeling of the geometry is being done in Unigraphics software.
2. The generated IGES file is exported to ANSYS workbench
3. The model is discretised into finite elements by triangular mesh elements.
4. Applying boundary conditions and loads.
5. Solve the problem.

4.1.1 ANALYSIS OF INPUT SHAFT



. Fig-4.1 Stress distribution on input shaft

Stress distribution on input shaft as shown in fig-4.1, as the maximum stress induced in the material (1.17 N/mm^2) < allowable stress (144 N/mm^2) the input shaft is safe under pure Torsional load.

4.1.2 ANALYSIS OF INPUT COUPLER

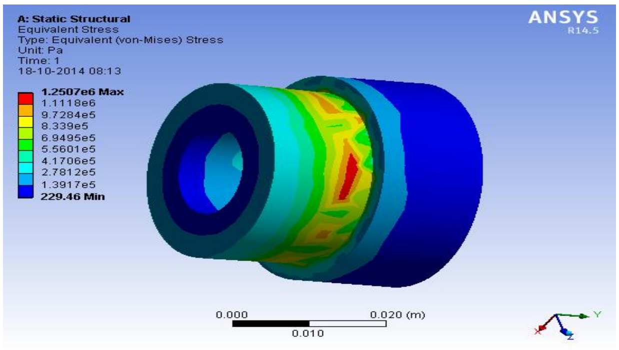


Fig-4.2 Stress distribution on input coupler

Stress distribution on input coupler as shown in fig-4.2, as the maximum stress induced in the material (1.25 N/mm2) < allowable stress (144 N/mm2) the IP Coupler is safe under pure torsional load.

4.1.3 ANALYSIS OF DRIVER DISK

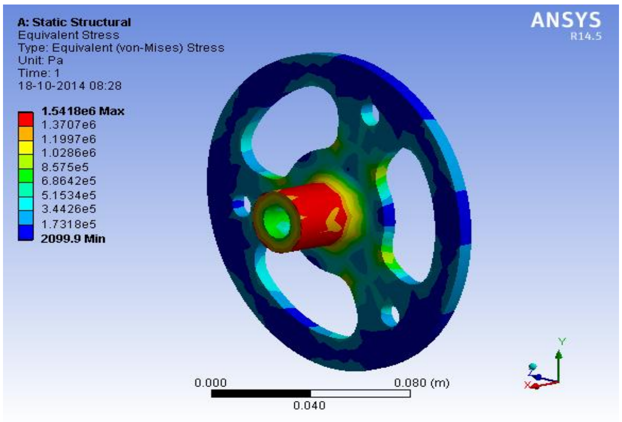


Fig-4.3 Stress distribution on driver disk

Stress distribution on driver disk as shown in fig-4.3, as the maximum stress induced in the material (1.54 N/mm2) < allowable stress (144 N/mm2) the Driver Disk is safe under pure torsional load.

4.1.4 ANALYSIS OF INTERMEDIATE DISK

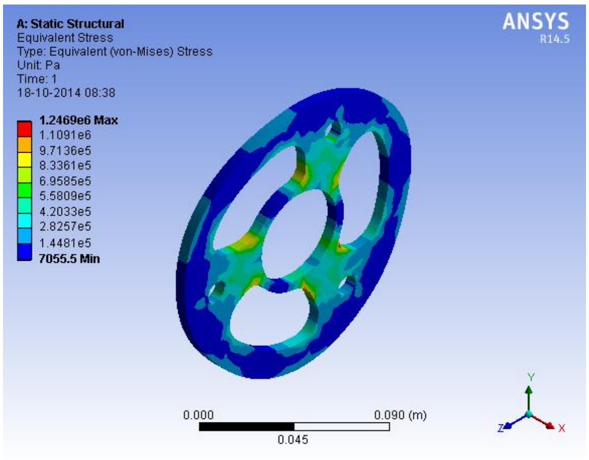


Fig-4.4 Stress distribution on intermediate disk

Stress distribution on intermediate disk as shown in fig-4.4, As the maximum stress induced in the material (1.24 N/mm2) < allowable stress (144 N/mm2) the Intermediate Disk is safe under pure torsional load.

4.1.5 ANALYSIS OF COUPLER SHAFT

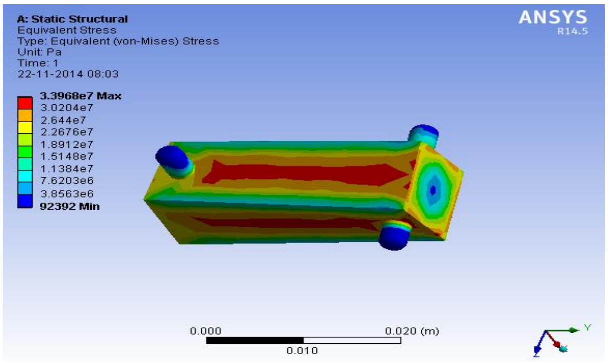
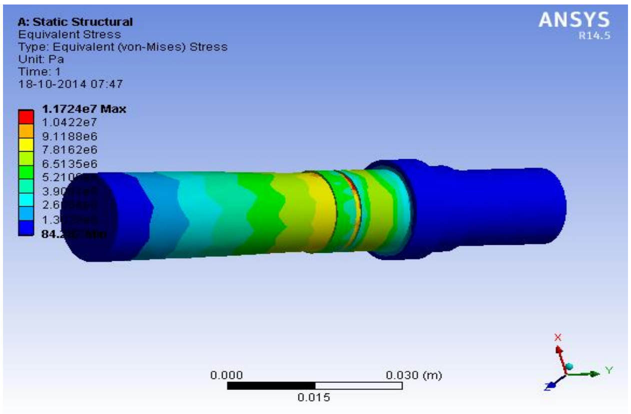


Fig-4.5 Stress distribution on Coupler Shaft

Stress distribution on coupler Shaft as shown in fig-4.5, As the maximum stress induced in the material (3.39 N/mm^2) < allowable stress (144 N/mm^2) the Coupler shaft is safe under pure torsional load.

4.1.6 ANALYSIS OF OUTPUT SHAFT



. Fig-4.6 Stress distribution on output shaft

Stress distribution on output shaft as shown in fig-4.6, as the maximum stress induced in the material (1.17 N/mm2) < allowable stress (144 N/mm2) the input shaft is safe under pure torsional load.

4.1.7 ANALYSIS OF OUTPUT COUPLER

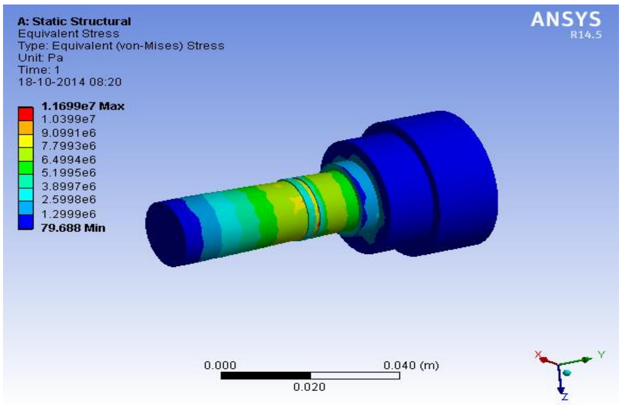


Fig-4.7 Stress distribution on output coupler

Stress distribution on output coupler as shown in fig-4.7, as the maximum stress induced in the material (1.16 N/mm2) < allowable stress (144 N/mm2) the OP Couplar is safe under pure torsional load.

4.1.8 ANALYSIS OF DRIVEN DISK

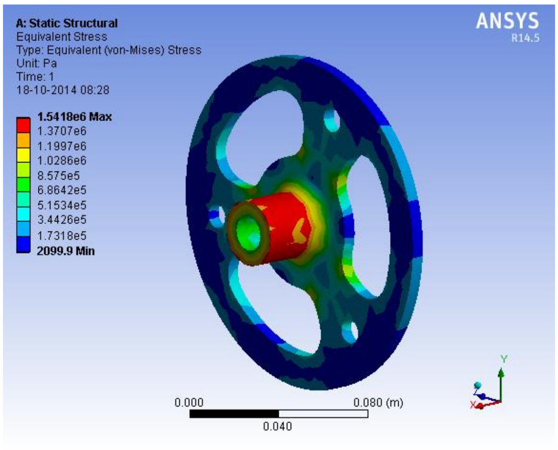


Fig-4.8 Stress distribution on driven disk

Stress distribution on driven disk as shown in fig-4.8, as the maximum stress induced in the material (1.54 N/mm2) < allowable stress (144 N/mm2) the Driven Disk is safe under pure torsional load.

V. RESULTS AND DISSCUSIONS

5.1 COMPARATIVE ANALYSIS OF ANGULAR OFFSET PERFORMANCE OF COUPLING

5.1.1 Torque analysis

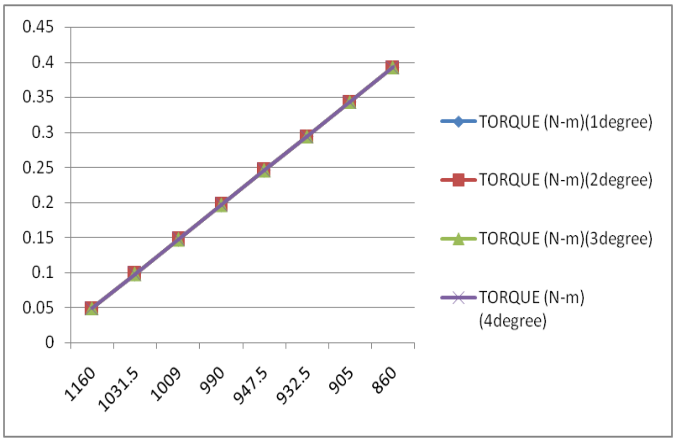


Fig-5.1 Variation of Torque V/s Speed for different angular offset angles

Variation of torque for different angular offset angles is shown in fiig-5.1. It is observed that the torque values remain almost same for all angular offset settings.

5.1.2 Output Power analysis

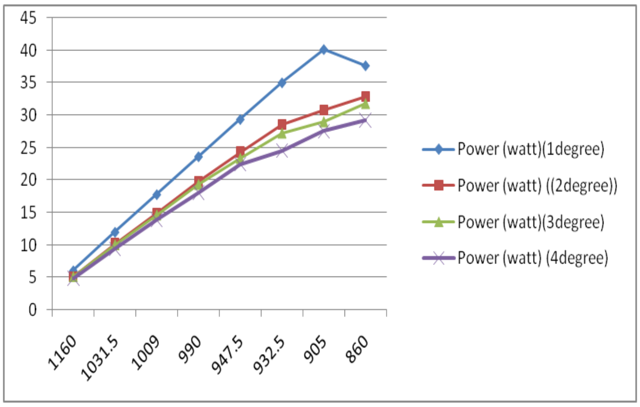


Fig-5.2 Variation of Power Output V/s Speed for different angular offset angles

Variation of power output for different angular offset angles is shown in fig-5.2. It is seen that there is marginal drop in output power with increase in angular offset thus it can be safely stated that coupling offers maximum power output at minimum angular offset.

5.1.3 Efficiency analysis

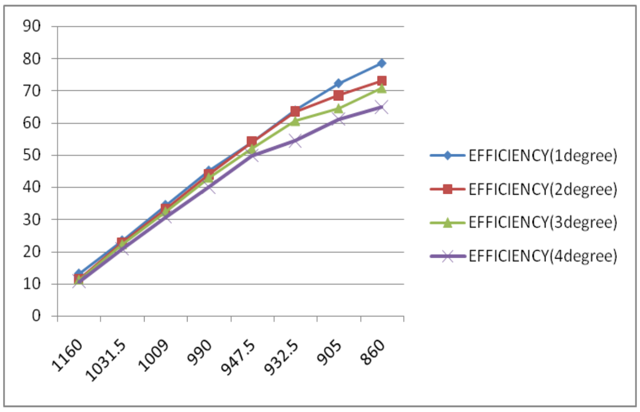


Fig-5.3 Variation of Efficiency V/s Speed for different angular offset angles

Variation of efficiency for different angular offset angles is shown in fig-5.3. It is seen that there is marginal drop in efficiency with increase in angular offset thus it can be safely stated that coupling offers maximum efficiency at minimum angular offset.

5.2 *COMPARATIVE ANALYSIS OF PARALLEL OFFSET PERFORMANCE OF COUPLING*

5.2.1 Torque analysis

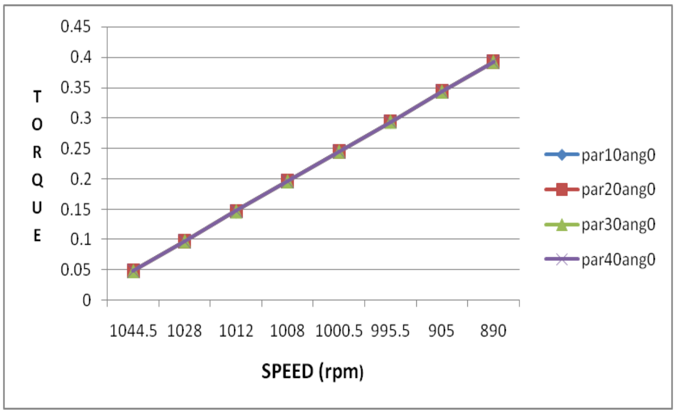


Fig-5.4 Variation of Torque V/s Speed for different parallel offset angles

Variation of torque for different parallel offset angles is shown in fig-5.4. It is observed that the torque values remain almost same for all parallel offset settings.

5.2.2 Output Power analysis

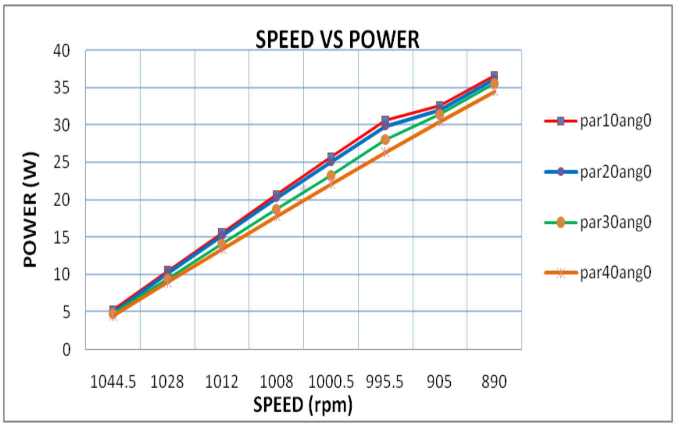


Fig-5.5 Variation of Power Output V/s Speed for different parallel offset angles

Variation of power output for different parallel offset angles is shown in fig-5.5. It is seen that there is a marginal drop in output power with increase in parallel offset thus it can be safely stated that coupling offers maximum power output at minimum parallel offset.

5.2.3 Efficiency analysis

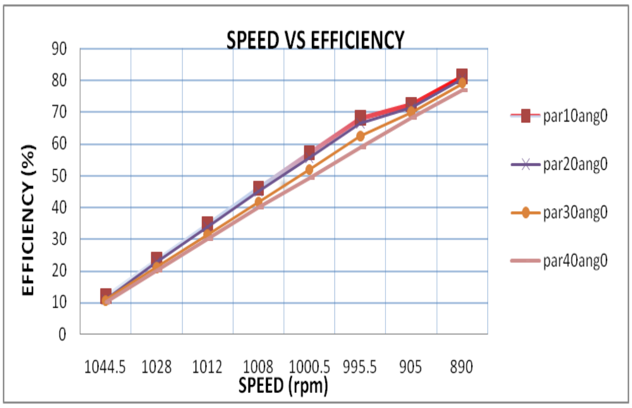


Fig-5.6 Variation of Efficiency V/s Speed for different parallel offset angles

Variation of efficiency for different parallel offset angles is shown in fig-5.6. It is seen that there is marginal drop in efficiency with increase in parallel offset thus it can be safely stated that coupling offers maximum efficiency at minimum parallel offset.

VI. CONCLUSIONS

From the experimental setup of parallel and angular offset coupling, the following results were obtained

- The maximum displacement or offset in parallel condition is 35mm on either side of mean.
- Torque transmitted by the coupling drops with increase in speed marginally.
- Maximum efficiency of coupling is achieved when operated at zero offset, but there is a marginal decrease in efficiency as offset is increased.
- The coupling can transmit angular offset from 1o to 5o .The angular offset is adjustable in step-less manner meaning that even an offset angle of 3.9 o is possible.
- Maximum efficiency of coupling is achieved when operated at zero angular offset, but there is a marginal decrease in efficiency as angular offset is increased.

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