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DESIGN OF AN AERO GAS TURBINE ENGINE GEARBOX POWER TRANSMISSION SPLINE-SPIGOT JOINT

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ABSTRACT

In aero engines, the angular drive, which is used for transmission of torque during engine start and to provide power to the drive accessories during normal operation, functions through two pairs of bevel gears viz. the engine bevel pair and the gearbox bevel pair. These bevel gear pairs are critical components, the failure of which will lead to fatal accidents. Successful performance of the bevel gears depends on accuracy of gear design and effectiveness of coupling methodology adopted. This paper presents the design of spline - spigot nut coupling of spiral bevel and spur gear on to a main shaft in the gearbox based on a novel approach of quadruple split of input torque with known concepts. The work involves investigation of potential causes that lead to the failure of spline spigot –nut coupling between the gearbox spiral bevel-spur gear and main shaft in the initial design, modification of design incorporating the quadruple split of input torque, testing and prove out of modified design. The design was analytically verified and tested subsequently to validate the design and to assess the performance of the modified main shaft assembly against the old design. The results showed performance improvement of the new design with respect to the old design.

Keywords: Gear, Shaft, Bevel, spline Interference fit, Locknut, coupling methodology.

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INTRODUCTION

The main power transmission train is one of the vital unit of the aero-engine which is designed to transmit torque to engine during the starting mode and to siphon power from the engine for driving accessories during normal operating mode. The starter, mechanically coupled through a clutch to the transmission train, generates the torque to accelerate the engine to the “light up” speed during the starting mode and supplements the engine torque till “self sustaining speed” is achieved. Clutch mechanism decouples the starter from power train during the normal operating mode.

The mechanical power required to drive the engine and aircraft gearboxes, plus thereby connected accessories, is tapped from the engine spool by a pair of engine bevel gears. Engine and aircraft accessories are generally mounted on two separate gearboxes viz. Engine gearbox and Aircraft gearbox to drive engine and aircraft accessories respectively. The accessories driven by the engine gearbox include main fuel pump, reheat pump, engine hydraulic power pack, alternator, dynamic air-oil separator and oil pump. Aircraft gearbox provides the drive for aircraft hydraulic pump, generator and oil pump. During the normal operating mode, power from the engine bevel gear is transmitted to the engine gearbox through a tower shaft and a pair of bevel gears in the gearbox. Greater portion of this power is distributed in the engine gearbox to drive the accessories by means of parallel axis gears [1] and the other portion is transmitted to the aircraft gearbox through the main shaft. The gear train with the HP shaft is shown in figure 1.

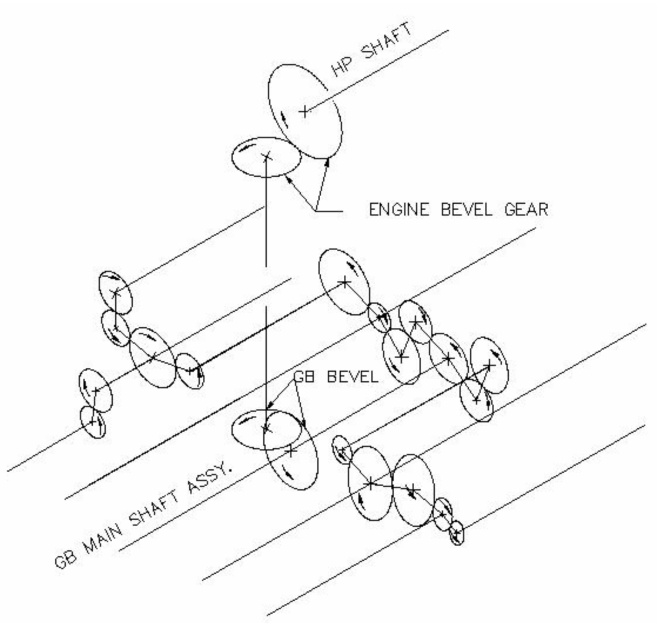

Fig1-: Engine gearbox gear train with HP shaft

Mechanical couplings for each of the components in the power transmission train are designed to meet the functional and life requirements with the specified reliability. Coupling of higher criticality is that of the spiral bevel gear onto the shaft. Engine bevel is coupled to the engine spool shaft or an independent shaft by spline –spigot and nut or pins-spigot and nut depending on the Engine architecture followed. Engine and gearbox bevel pinions are generally integrated to the shaft and mounted on the respective casings through bearings. These two bevel pinions are connected by the tower shaft with splines on either side of the shaft and functioning as flexible splines. Gearbox bevel gears are either integral with the shaft or mechanically coupled to the shaft based on the design methodology of the gearbox. The configuration, design and manufacturing constraints may demand mechanical coupling though gears integral with the shaft are more reliable due to reduced eccentricities and fretting free operation.

Two main coupling philosophies followed for the mechanical connectivity of gearbox bevel gear to main shaft are spigot- bolted and spline-spigot nut. This paper is on the design of a gearbox power transmission spline –spigot nut joint with brief mention on potential causes that led to the failure of the initial design and validation of the new design through testing on actual hardware.

BASIC CONFIGURATION

Gearbox main bevel shaft assembly comprises of main shaft, spiral bevel gear, spur gear and locknut. Both the gears are designed with fillet root side fit involute internal splines and the main shaft with matching external splines for coupling.

Main shaft bevel assembly is shown in fig 2. Bevel gear is radially located on the main shaft through an accurately ground spigot ‘a’ and is axially located through a butting face. The spur gear in turn is radially aligned to the shaft through a double spigot. The front of spur gear spigots on the bevel gear and rear spigots on the main shaft and the spur gear is axially positioned against a butting face on the Bevel gear. Spiral bevel gear and spur gear are held in position to the main shaft by a nut which is torque tightened and locked. The spline- spigot joint between the spiral bevel, spur gear and the main shaft performs the following function:

a. Starting torque received by the main shaft from the starter is transmitted to the gearbox bevel gear.
b. Power received by the gearbox bevel gear from the engine during normal operation mode is transmitted to the main shaft and spur gear.
c. Safeguards the Engine gearbox on the event of a failure of the shear quill shaft (mechanical fuse) in the aircraft gearbox.

Spiral bevel gear designed to operate under torque reversal based on the mode of operation transmits peak torque of T_{cs} to the engine from a starter and receives P_{peak} from the Engine to drive various accessories during normal operation. Engine gearbox mounted accessories are driven by the power transmitted through the spur gear at the rear of the main shaft. Power from the main shaft is transmitted through a power take off shaft to drive the aircraft gearbox mounted accessories. Power take off shaft is designed with a diaphragm coupling on either side to permit specified angular and axial misalignment.

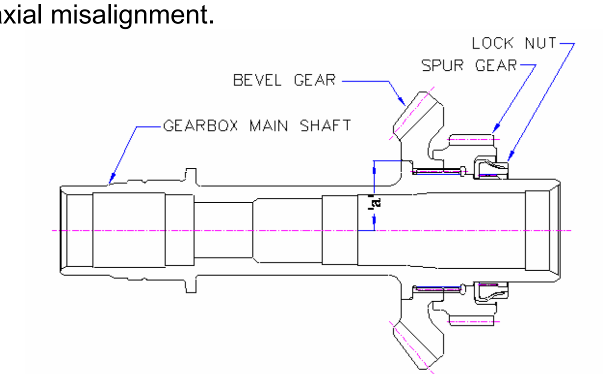

Fig 2-: Gearbox main shaft, bevel –spur gear arrangement

FAILURE IN INITIAL DESIGN

Gearbox was subjected to rigorous testing in the endurance test rig under simulated operational load conditions with power absorption through water brake dynamometers. After considerable testing hours, the gearbox was withdrawn due to deviation in vibration signatures, with respect to the initial references, corresponding to the Gearbox main shaft rotational, bevel mesh and spur gear mesh frequencies. Upon disassembly, following observations were made:

- Excessive wear out of splines
- Fretting of spigot diameters and butting faces
- Reduction in locknut clamping torque

Figure 3 (i),(ii) and (iii) presents the fretting wear and the spline wear out.

(i) **(ii)** **(iii)**

Fig 3-: (i)-shows the fretting wear on the bevel gear butting face, (ii & iii) - shows the spline wear out

The cause of failure was mainly attributed to fretting due to the relative motion among butting faces and locating diameters of the spiral bevel, spur gear and the main shaft [2]. Following reasons were identified for the fretting between the gearbox spiral bevel, spur gear and the main shaft:

a. Insufficient axial load in the assembled shaft to prevent the angular movement of the gearbox bevel gear under torque reversal conditions.
b. Insufficient 'spigot base and support' for the spiral bevel gear to react the gear tooth load on the shaft.
c. High bending elastic deflection at the butting face of Spiral bevel and spur gear interface.
d. Loss of friction locking ring effectiveness at higher speeds due to centrifugal effects.

MODIFIED DESIGN

Feedbacks obtained from testing of initially designed components were utilized to evolve a novel approach to minimize the effect of all forces that could lead to relative movement between the clamped components and to design a joint with over load torque transmission capacity.

Power transmission spline –spigot joint was designed to minimize the relative movements between mating components by following a three pronged approach. First by stiffening the components to reduce deflections second by selection of butting face with least bending elastic deflections and third by incorporating semi-permanent joint [3] offering higher interface resistive force against relative movement. A semi-permanent joint allows part to be separated with some difficulty, in comparison with temporary joint, but normally they are not dismantled. Design requirements mentioned above in a space constraint location lead to the evolution of the concept of quadruple split of input torque.

CONCEPT OF QUADRUPLE SPLIT OF INPUT TORQUE

Torque flow from and to the aero-engine gearbox is through the bevel pairs and hence it is absolutely essential to design the torque transfer path through these elements considering the torque reversals. For non integral bevel gears, this is achieved by parallel torque sharing sleeves and design with varying torsional flexibility for torque split. This is possible where adequate space is available to accommodate higher axial dimensions of such components. In the present design, the axial space was a major constraint and a novel approach was followed to utilize the axially preloaded semi permanent joint for primary torque transfer and splines for overload condition. Axially preloaded frictional butting faces, interference fitted diameters and splines were identified to transmit torque in this design.

Analytical estimation of torque transmission capacity were done for interference fit, axially preloaded frictional butting faces and spline. Tests were performed to assess the torque transmitted through axially preloaded frictional butting faces and spline capacity was proved based on tests performed on similar operational spline joints [2].

Input torque (T_{in}) to the bevel gear under normal conditions is shared by the interference fits and frictional butting faces depending on the relative capacity. The reason for this assumption of torque sharing is that both transmits torque and resist the relative movement of the components at the interface. Sharing of input torque results in triple split (T1, T2 and T3) under normal operation, of which two paths are for torque transfer to shaft (T1- through interference fit and T2 - through the bevel gear and main shaft frictional butting face) and remaining (T3-through the bevel gear and spur gear frictional butting face) is to spur gear. This is schematically shown in Fig 4.

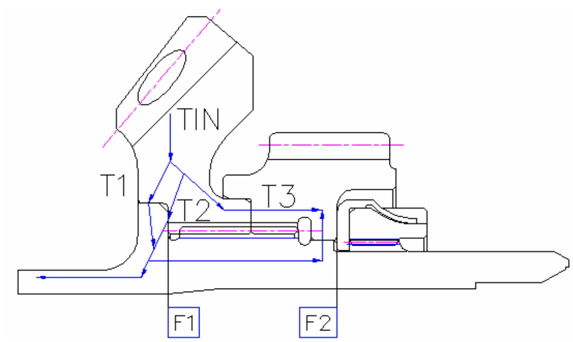

Fig 4-: Input torque triple split.

Overload torque transfer across the semi permanent joints will lead to macro or micro-level slippage. Under such conditions, a fourth torque transfer path is designed through the splines which will take up all loads in excess to the normal torque levels and this occurs under input torque quadruple split condition.ie. the design incorporate the capability to provide a fourth path for the loads when overload occurs. Figure 5 schematically shows the input torque quadruple split. Quadruple split of input torque (T1, T2, T3 and T4) occurs because of torque sharing between the torque transmitting element viz. axially preloaded frictional butting faces, interference fits and splines.

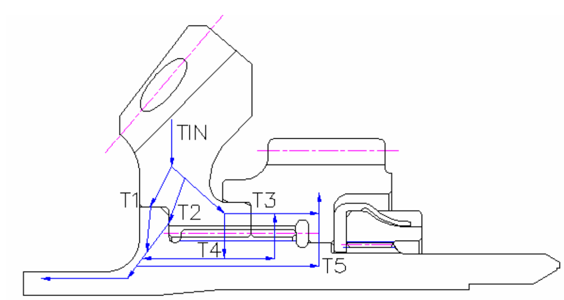

Fig 5-: Input torque quadruple split.

Torque flow path to the main shaft under overload condition is through the interference fit (T1), frictional butting face (T2) and the splines (fraction of T4). Torque flow path to the spur gear under overload condition is through the interference fit (T5), frictional butting face (T3) and the splines (fraction of T4). Spline, in this joint is designed to transmit the overload torque forcing the mechanical fuse in the aircraft gearbox to give way under extreme conditions. The focus of remaining part of the paper is primarily on the design verification of each of the elements in the spline –spigot joint contributing to transfer of torque.

All dimensions are normalized with respect to 'a' ,which is the nominal radius of fit between the bevel gear and the main shaft. Torque, horse power and stress are normalized with respect to maximum start torque (T_{cs}) , peak horse power (P_{peak}) and 0.2% proof stress (σ_y) respectively. Material properties of low alloy case carburizing steel used for the manufacturing of gears and shafts are used for the analysis. Normal operating temperature of the gearbox is 100°C and not to exceed temperature is 140°C.

Torque transmitted through friction

Transmission joint was designed to transmit the power primarily by means of friction between the butting faces through nut preload, interference fit and to utilize the splines to take up overload that could occur during the mission profile.

Value of locknut clamping torque was chosen to introduce a preload within safe thread strength and elastic stretch in the shaft between the front and rear butting face (F1 and F2 shown in fig 4) to generate the axial preload load to transmit torque through the frictional butting faces. Component clamping faces were selected such that the clamping load line of action is straight and parallel to the axis as possible. Butting faces between components were identified using FE analysis in ANSYS 8.0 where the elastic deflections are minimum.

Theoretical estimation of pre load based on [4] was carried out to predict the maximum torque that can be transmitted across the frictional butting faces for tightening torque of T_{it}. Design of assembly process performed ensures minimum loss of axial load to drive in the interference fit components.

Preload F_p introduced in the shaft corresponding to the tightening torque T_{it} is given by (1)

$$T_{it} = F_p \left(\frac{P}{2\pi} + \frac{\mu_{th} r_{tcr}}{\cos\beta} + \mu_n r_n \right) \qquad (1)$$

T_{it} is resisted by three reaction torques produced by bolt stretch component, frictional restraint between nut and bolt threads and frictional restraint between face of nut and joint.

Torque transmitted (T_f) through the frictional butting face for an axial load of F_p is given by (2)

$$T_f = F_p \, \mu_n r_n \tag{2}$$

Tightening torque of $1.85T_{cs}$, introduces a preload that transmits a peak torque of $0.6T_{cs}$ between the bevel gear and shaft towards the front and between the bevel gear and spur gear towards the rear with a factor of 1.5. This is proved by rig testing the frictional torque transmission through main shaft assembly upto $0.9T_{cs}$ without interference fit and relieving the splines. Fig 6 shows the bevel and spur gear part of the main shaft assembly after the frictional torque transmission testing.

Calculations with minimum material condition and assumed coefficient of friction of 0.1 at the butting face show that the selected clamping torque can safely transmit complete normal operating torque and a major fraction of the peak start torque T_{cs} through the frictional surfaces.

Fig 6-: Components after testing

Torque transmitted by interference fitted spigots

Torque transmission through interference fit is estimated based on the Lame's Equation [5,6]. Radial interference fit design [7] for torque transmission across mating components was done considering centrifugal, thermal, surface finish and tolerance stack up effect. The edges of the gear hub beyond the interference zone were designed to decrease stress concentration [7]. Both stepped and raised diameter design of shaft at interference location was followed to meet the functional and geometrical requirements.

Ratio of the inner diameter of the bevel and spur gear to their respective thickness place them in the range of thick wall cylinders. For a thick wall approach, the radial stress (σ_r) and circumferential stress (σ_θ) generated on a cylinder subjected to external and internal pressure is given by:

$$\sigma_r = \left(\frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2}\right) - \left[\left(\frac{r_i^2 r_o^2}{r^2}\right)\left(\frac{p_i - p_o}{r_o^2 - r_i^2}\right)\right] \tag{3}$$

$$\sigma_\theta = \left(\frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2}\right) + \left[\left(\frac{r_i^2 r_o^2}{r^2}\right)\left(\frac{p_i - p_o}{r_o^2 - r_i^2}\right)\right] \tag{4}$$

Radial stress, circumferential stress and radial displacement of a hub with a hole of radius r_{ih} and rotating at angular velocity of ω is given by

$$\sigma_{rh} = \left(\frac{3+\nu}{8}\rho\omega^2\right)\left(r_{oh}^2 + r_{ih}^2 - \frac{r_{oh}^2 r_{ih}^2}{r^2} - r^2\right) \tag{5}$$

$$\sigma_{\theta h} = \left(\frac{3+\nu}{8}\rho\omega^2\right)\left(r_{oh}^2 + r_{ih}^2 + \frac{r_{oh}^2 r_{ih}^2}{r^2} - \frac{1+3\nu}{3+\nu} r^2\right) \tag{6}$$

$$u_r = \frac{r}{E}\left(\sigma_{\theta h} - \nu \sigma_{rh}\right) \tag{7}$$

Maximum torque (T_{cs}) transmitted through the bevel gear is under the starting condition and this occurs in the range of 5 to 6% of the rated main shaft speed n_{rt}. At this speed, the torque that can be transmitted through the interference fit is $0.197T_{cs}$ and torque that can be transmitted across the butting face through friction is $0.6T_{cs}$. Maximum interference of 0.00319a is maintained between bevel gear and the main shaft to ensure that the peak stresses in the interference zone is within safe limits. Interference fit maintained between spur gear and the main shaft is 0.001278a. Figure 7 shows the location and maximum interference fit for both the gears with the main shaft.

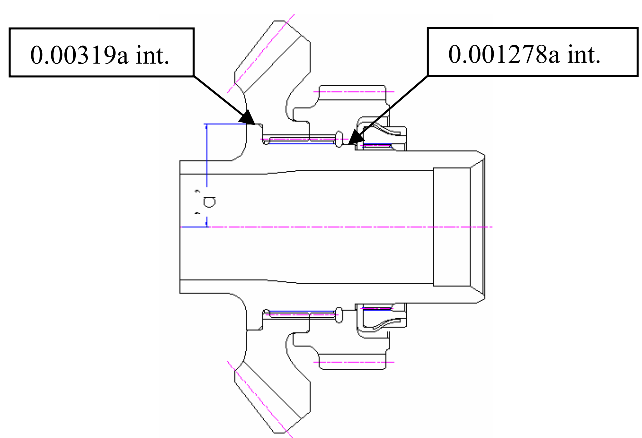

Fig 7-: Interference fit at the collars of the assembly

Fig 8 and fig 9 shows the variation of torque transmission capacity of interference fit with rotational speed. With the increase in speed, the torque transmission capacity through the bevel gear and main shaft interference fit decreases and there is an increase in the torque transmission requirement through friction between the bevel and spur gear butting face.

Choice of minimum value of interference fit for the bevel gear and the spur gear was based to ensure that fit with respect to the main shaft does not enter into the clearance zone under the centrifugal load upto the maximum permitted over speed. This is evident from the Fig 8 and 9 for both the bevel and spur gear joint.

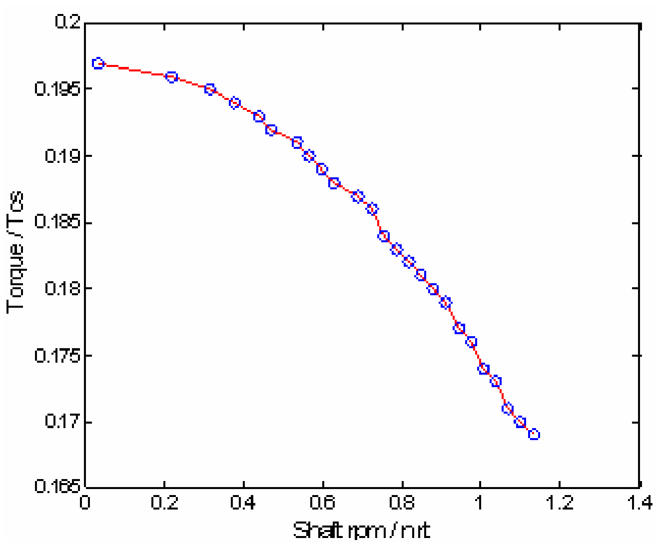

Fig 8-: Variation in Non-dimensional torque transfer with speed through the bevel gear side interference fit.

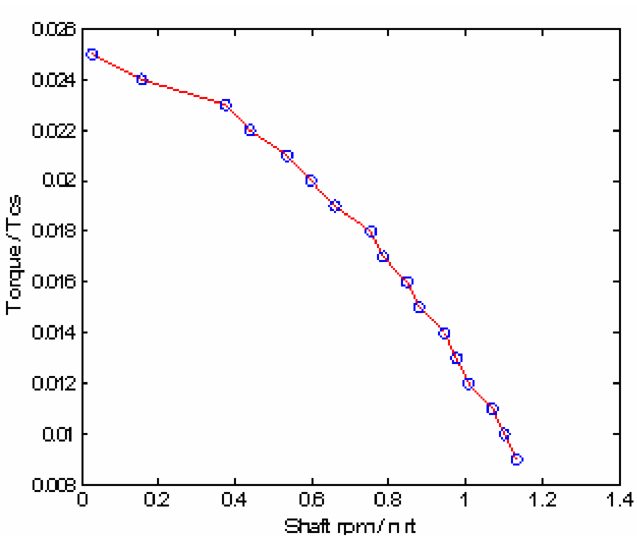

Fig 9-: Variation in Non-dimensional torque transfer with speed through the spur gear side interference fit.

At peak speed, torque transmission through the bevel gear and main shaft interference fit is $0.17T_{cs}$.

Assessment of torque transmission through the interference fit by testing was not performed as the fraction of torque transmitted through interference is relatively low.

Torque transmitted through spline

Frictional butting faces of the bevel gear, spur gear and the main shaft and the interference fit between the gears and the main shaft is designed to transmit the normal operating torque Transient overload torque in-excess to the capacity of the normal transmission path is designed to be transmitted through the splines. Under overload condition, gear and the shaft undergoes relative slip on the face and diameter corresponding to spline clearance and the splines get engaged.

Fillet root side fit involute splines of Class 2 fit to BS 3550 standard [8] was selected for the transmission. Spline parameters as given in Table-1 are chosen such that the capacity of the spline is sufficient enough to transmit torque corresponding to the failure torque of the mechanical fuse thus ensuring that this joint is safe.

Table-1: General spline parameters

Spline	
Diametral Pitch	20/40
Pitch Dia. (ref.)	1.704a
Pressure angle	30°
Class of fit	2

High performance spline couplings in aero applications are bound to fret under severe load conditions to which it is continuously subjected [9], specifically in the case for flexible splines.

The stress analysis of the external and internal spline of gears and main shaft was carried out for the fixed condition of the splines [10,11].Fixed splines permit no relative or rocking motion between internal and external teeth and can be shrink fitted or loosely fitted together. Splines were designed based on the following criteria:

I. Shear stress in spline teeth
II. Compressive stress in spline teeth
III. Bursting stress in internal spline parts

The induced shear stress on the spline teeth due to the transmitted torque T is given by

$$\sigma_s = \frac{4TK_m}{DNF_e\, t_c} \qquad (8)$$

Calculated shear stress is modified by the factors K_a and L_f and compared against the allowable shear stress σ'_S based on the material and its hardness.

$$\sigma'_s \geq \sigma_s \frac{K_a}{L_f} \quad (9)$$

Compressive stress on the spline teeth is given by:

$$\sigma_c = \frac{2TK_m}{DNF_e\, h} \quad (10)$$

Computed compressive stress is modified by the factors K_a and L_f and compared against the allowable stress σ'_c based on the material and its hardness.

$$\sigma'_c \geq \sigma_c \frac{K_a}{9L_f} \quad (11)$$

Internally toothed spline part tends to burst due to three different kinds of tensile stresses:

- Bursting stresses caused by the radial force component at the pitch line.

$$\sigma_{t1} = \frac{T \tan \Phi}{\pi D t_w F} \quad (12)$$

- Bursting stress caused by centrifugal force.

$$\sigma_{t2} = 0.828(10)^{-6}(n^2)(2D_{oi}^2 + 0.424 D_{ri}^2) \quad (13)$$

- Tensile stress due to beam loading of spline teeth .

$$\sigma_3 = \frac{4T}{D^2 F_e Y} \quad (14)$$

All of the above stresses (11, 12, and 13) are tensile stresses at the root diameter of the internally toothed part. The total stress tending to burst the rim modified by the factors K_a , K_m and L_f is compared against the allowable tensile stress σ'_t .

$$\sigma'_t \geq \frac{K_a K_m(\sigma_{t1} + \sigma_{t3}) + \sigma_{t2}}{L_f} \quad (15)$$

The spline stress analysis details for the spiral bevel and spur gear, based on the fixed spline design approach, is given in table 2.

Table 2 -Non dimensional stresses on bevel and spur gear splines for starting and shear neck failure condition

Condition / Stress	Starting stress (Bevel Gear)	Shear neck failure	
		Bevel gear	Spur gear
Shear stress in teeth	0.03 σ_y	0.0914 σ_y	0.0386 σ_y
Compressive stress	0.0027 σ_y	0.0083 σ_y	0.0035 σ_y
Bursting stress	0.080 σ_y	0.1967 σ_y	0.1039 σ_y

Allowable shear stress σ'_s = $0.281\sigma_y$

Allowable compressive stress σ'_c = $0.021\sigma_y$

Allowable tensile stress σ'_t = $0.317\sigma_y$

In the design presented in this paper, the torque transmission through the splines is minimum under normal conditions and at the same time the theoretical factor of safety of spline is 1.6 under shear neck failure condition assuming that the entire torque is transmitted through the splines. Spline capacity was proved based on tests performed on similar operational spline joints.

Brief of the design validation test performed on the main shaft assembly for the starting and normal operating condition is given below.

TESTING OF THE SPLINE SPIGOT JOINT

Main shaft assembly with the bevel and spur gear was subjected to extensive testing to prove the design approach. For this, two test setups were used (i) a four square back to back test set up to prove the start sequence, where the torque transmission is through the bevel and (ii) power absorption set up to prove the normal operating sequence, where both the bevel and spur gear are loaded.

A four square back to back test set up is used to test gears in a closed loop.Torque and rotation is simulated simultaneously through a torquing unit and a motor programmed to follow the start sequence cycle to the required torque and speed levels. Torquing unit and the driving motor is connected at separate locations in the loop and the gearbox bevel gear mounted on the main shaft assembly closes the torque flow loop. Torque is introduced in the loop by an angular twist imparted by the torque unit and the gear train is rotated independently by an electric drive. Advantage of this set up is that the electric drive will be only rated to overcome the mechanical losses and impart the required speed. This is schematically shown in fig 10 and test set up is shown in fig 11. Gearbox bevel gear was subjected to simulated starting torque and speed for cycles several times more than the starting cycles in engine life. Main shaft assembly was subsequently tested to $1.15T_{cs}$.

In the power absorption test set up, the gear box was tested with the accessory loads simulated through dynamometers. Test cycle for the normal operation is based on the equivalent mission profile for accelerated test.

Health of the gearbox was monitored through online vibration measurement, bearing and scavenge oil temperature measurement. Offline monitoring was done through magnetic chip detectors and oil analysis.

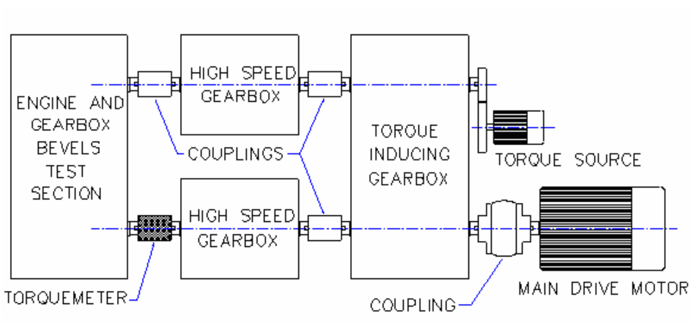

Fig 10-: Schematic diagram of four square back to back test set up

Fig 11-: Four square back to back test set up

Figure 12 and fig 13 shows start sequence cycles and normal operating cycle respectively performed in these facilities. Time in seconds is marked along the abscissa in figure 12 and 13. Speed in rpm, torque in Nm and axial load in N is marked along the ordinate in figure 12. Speed in rpm, torque in Nm and axial load in N, RMS vibration value in mm/sec at various critical locations of the test set up and lubricating oil supply pressure in pounds per square inch is marked along the ordinate in figure 13. Figure 12 shows start sequence testing for 8 successive cycles and figure 13 shows a typical test performed for normal operating sequence of one hour duration. Maximum value of T/ T_{cs} in fig.12 is at the lowest point on torque curve because starting torque is plotted along negative x-axis.

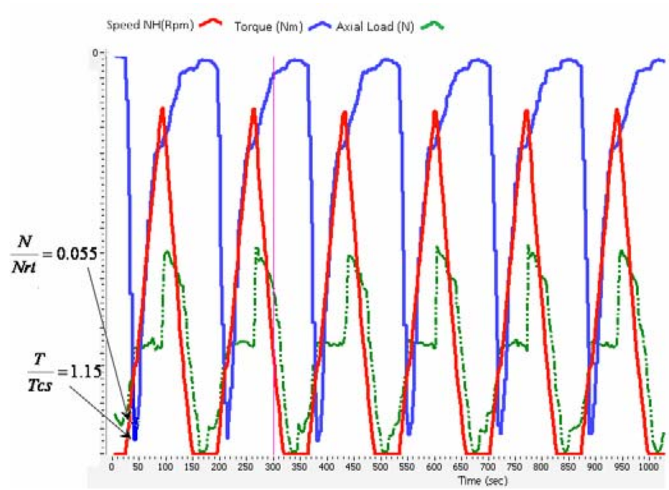

Fig 12-: Plot of 8 successive Start sequence cycles

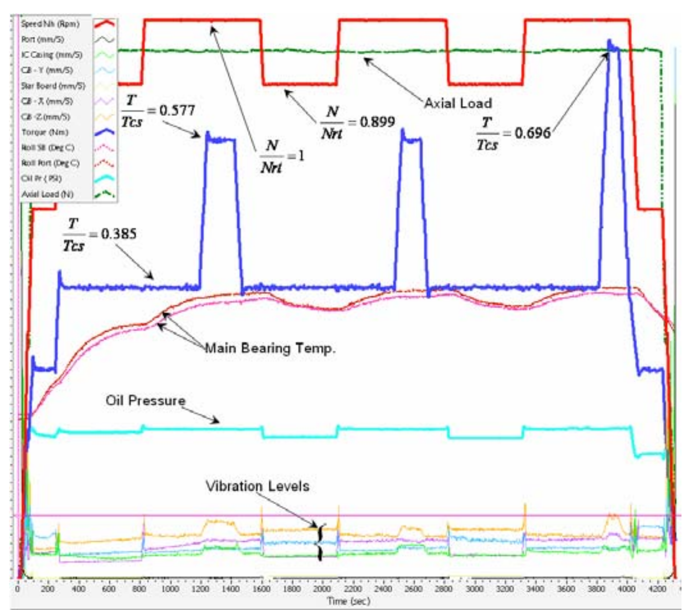

Fig 13-: Plot for the simulated normal operating condition

Figure 14 shows the bevel and spur gear part of the main shaft assembly after testing.

Fig 14-: Component after testing

After completion of stipulated duration of testing, following inspections were performed on the components:

- Visual check – The components were intact without brownish iron oxide paste which indicated absence of fretting.
- Profile inspection of bevel gear tooth and spur gear –Profiles were within acceptable levels
- Magnetic particle inspection – Components were free of surface cracks
- Lock nut retention torque – Lock nut retained the tightening torque
- Profile check of spline- Inspection of splines revealed contact on the working profile, which is indicative of torque transmission and the profile was well within acceptable limits.

Results of inspection revealed that all components are intact with no signs of distress.

CONCLUSION

It is an engineering practice to make integral gears and shafts wherever possible since this reduces eccentricities and eliminates fretting problems at abutment faces, location diameters and drive splines. When such integral gears are not possible, number of components in clamped stacks should be kept to a minimum and multiple torque transmission path has to be designed, tested and proved.

Design of aero gearbox power transmission spline spigot joint presented in this paper based on torque split, due to torque sharing, was analytically verified and tested to 1.15 T_{cs}.

Tightening torque of $1.85T_{cs}$, introduces a preload that transmits a peak torque of $0.6T_{cs}$ between the bevel gear and shaft towards the front and between the bevel gear and spur gear towards the rear with a factor of 1.5 proved by rig testing.

Spline capacity was proved based on tests performed on similar operational spline joints upto accessory overload conditions. The theoretical factor of safety of spline is 1.6 under shear neck failure condition assuming that the entire torque is transmitted through the splines.

The concept of quadruple split of input torque ensures that this joint is safe under all operational and overload conditions.

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NOMENCLATURE

a	Shaft/bevel gear spigot radius, mm
D	Pitch Diameter of Spline, mm
D_{oi}	Outside diameter of internally splined part, mm
D_{ri}	Major diameter of internal tooth, mm
E	Young's modulus, MPa
F	Full face width, mm
F_e	Effective face width, mm
F_p	Preload, N
h	Radial height of tooth in contact, mm
K_a	Application factor
K_m	Misalignment factor
L_f	Life factor limited by fatigue
N	No. of spline teeth
n	Shaft speed, rpm
n_{rt}	Rated main shaft speed,rpm
P	Pitch, mm
p_i	Internal pressure on cylinder, MPa
p_o	External pressure on cylinder. MPa
r	radius, mm
r_i	Internal radius of cylinder, mm
r_{ih}	Internal radius of the hub, mm
r_n	Radius of contact between nut and joint surfaces, mm
r_o	External radius of cylinder, mm
r_{oh}	External radius of hub, mm
r_{tcr}	Effective thread contact radius, mm
T	Torque, Nmm
T_{cs}	Maximum start torque, Nmm

T_f	Frictional torque, Nmm
t_c	Chordal tooth thickness, mm
T	Torque, Nmm
T_{it}	Torque applied to the fastener, Nmm
t_w	Wall thickness of shaft, mm
u_r	Radial displacement of circular disk, mm
Y	Lewis form factor
μ_{th}	Coefficient of friction between male and female threads
μ_n	Coefficient of friction between the surfaces of the nut and corresponding butting face of joint
ω	Angular velocity, rad s^{-1}
Φ	Spline pressure angle, deg
ν	Poisson's ratio
ρ	Density, Kg (mm)-3
σ	Stress, Mpa
σ'	Allowable stress, Mpa
σ_{t1}	Bursting stress due to radial force, MPa
σ_{t2}	Bursting stress due to centrifugal force, MPa
σ_{t3}	Tensile stress due to beam loading of spline teeth, MPa
σ_y	0.2% proof stress, MPa
σ_{rh}	Radial stress in hub, MPa
$\sigma_{\theta h}$	Circumferential stress in hub, MPa
β	Half angle of thread, deg

Subscripts

c	Compressive
r	Radial direction
s	Shear
t	Tensile
θ	Circumferential direction

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