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Investigation of Critical Stress Combinations for Turbine Input Shaft of Transaxle Automotive Automatic Transmission

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Abstract

Hollow Turbine shaft is a starting part for torque, revolutions or energy into another component of the transmission. It for a transverse mounted four speed automotive automatic transmission whenever were introduced to study its stresses. It has two splined parts at its ends and in between a bearing support mounting on the without splined part. The engine power has been transmitted by a drive link assembly to the input gear sets shaft. The study of stressed states of splined and without splined parts of hollow turbine shaft resulting in fatigue failure under two different working operating conditions of torque has been done. Also, hollow turbine shaft is loaded by two vertical loads which are coming from torque converter, friction viscous clutch and drive link and sprocket assemblies. From that, there are many critical combinations of forces (contact, normal, twist and shearing forces) applied on hollow input turbine shaft. The critical forces can be possibility exist many types of cracks for cross section of shaft they are: inclined (torsion stress), longitudinal and transverse (shear stress), and vertical (bending stress). The torsion, shear, bending, and compound stresses are studied theoretically in this article.

Keywords

Transverse Automotive Automatic Transmission, Turbine Shaft Stresses, Torque Converter Torque Ratio, Torque Converter Weight, Shaft Crack

1. Introduction

The maximum tractive effort is a main parameter to evaluate the vehicle performance potential. One of the methods to determine the maximum tractive effort limit of a road vehicle is the vehicle power plant and its transmission. The vehicle performance is important as a source of stress and critical stress combinations of Hollow Turbine Shaft (HTS) for present transverse mounted four speed automotive automatic transmissions, which they are resulting in failure. It is common to focus as a first operation condition (*first stage*) on main loads acting on splined and no splined parts of HTS which are forces due to stall turbine torque and turbine weight on right splined side and vehicle movement resistant torque from rest, and drive link and sprocket assembly on left splined side also focus on the loads acting on no splined which are forces by resulting difference torque between stall turbine torque and vehicle resistance torque and reaction weights at bearing support point. [1]. Moreover, the study guaranties a second operation conditions (*second stage*) from that the above parts of HTS are forces due to engine torque at its maximum power (when viscous friction clutch of torque converter is active) and weight of torque converter with viscous friction clutch and flywheel on right splined side and vehicle movement resistant torque at minimum transmission reduction ratio, and drive link and sprocket assembly on left splined side also focus on the loads acting on no splined which are forces by resulting difference torque between engine at maximum power torque and new vehicle resistance torque and also, reaction weights at bearing support point.

Turbine forces are transmitted to right splined part of HTS and for easiest calculation, it is assumed that turbine transmits force and torque to splined part of shaft at middle point of its width, and study stressed is based on corresponding cross-section, although, turbine force of interaction between turbine hub and splined shaft part is distributed along length of turbine hub.

In studies of forces and stressed states of HTS resulting in fatigue failure of HTS for transverse mounted four speed automotive automatic transmission induce stresses which are of torsional, bending, and shearing focus primarily on torsion phenomena. In so doing, torsional, shear, bending and compound stresses are considered for the most parts.

2. A Transaxle Automatic Transmission Model [1-7]

The transaxle Automatic Transmission system is used for the transverse wheel drive. The transaxle system Figure 1 gives three forward reduction speed drive and one overdrive and one reverse with a Viscous Torque Converter Clutch (VTCC) however the turbine and the planetary gears input shaft are drive link chain assembly. The turbine torque variation has two stages. *The first one* is given at low speed ratio (n) of the torque converter which is turbine angular velocity (ω_T) over pump angular velocity (ω_P) as equation 1, and the ratio (R_T) between turbine torque (T_T) to pump torque (T_P) as indicated in equation 2. *The second stage* is occurred when the torque converter lock-up clutch is activated (at the design point), the turbine and pump are connected to the engine output. The angular velocities ω_T, ω_P, and ω_E (engine angular velocity) and torques T_T, T_P, and T_E (engine torque) are expressed as equation 3.

The transmission consists of compound planetary gear systems employing two simple planetary gear sets with two ring gears.

In the transaxle system Figure 1, first starting ratio, torque from the engine is multiplied through the torque converter by 1.6, transaxle sets by 2.92 and final drive by 2.84 to the vehicle's axle shafts. In the first set, T_T is transferred to the input sun gear (S_1) as a driving member, this occurs when the input clutch is applied and forces the input sprag (outer) race to rotate. The input sprag (inner) race is forced to rotate through the holding sprag elements, and the input sun gear (S_1) which is splined into the input sprag (inner) race is also forced to rotate. The teeth of S_1, transferred the torque to the four planetary pinion gears (P_1) and reaction internal gear (R_1) which is part of the input carrier assembly (C_2). Then the reaction internal gear (R_1) transfers the torque to planetary pinion gears (P_2) through the reaction carrier assembly (C_2) and forces the planetary pinion gears (P_2) to rotate around the stationary reaction sun gear (S_2) and Reaction Sun Gear Drum (RSGD). S_2 and RSGD is held stationary by the 1-2 band brake. When the transaxle is operating in this mode, reduction through the gear sets is 2.92:1

$$n = \omega_T/\omega_P \tag{1}$$

$$R_T = T_T/T_P = (U+\omega_{up}-nUR_t^2-\omega_{ut}R_t)/(U+\omega_{up}-\omega_{ur}R_p) \tag{2}$$

$$\omega_T = \omega_P = \omega_E,\ T_T = T_P = T_E \tag{3}$$

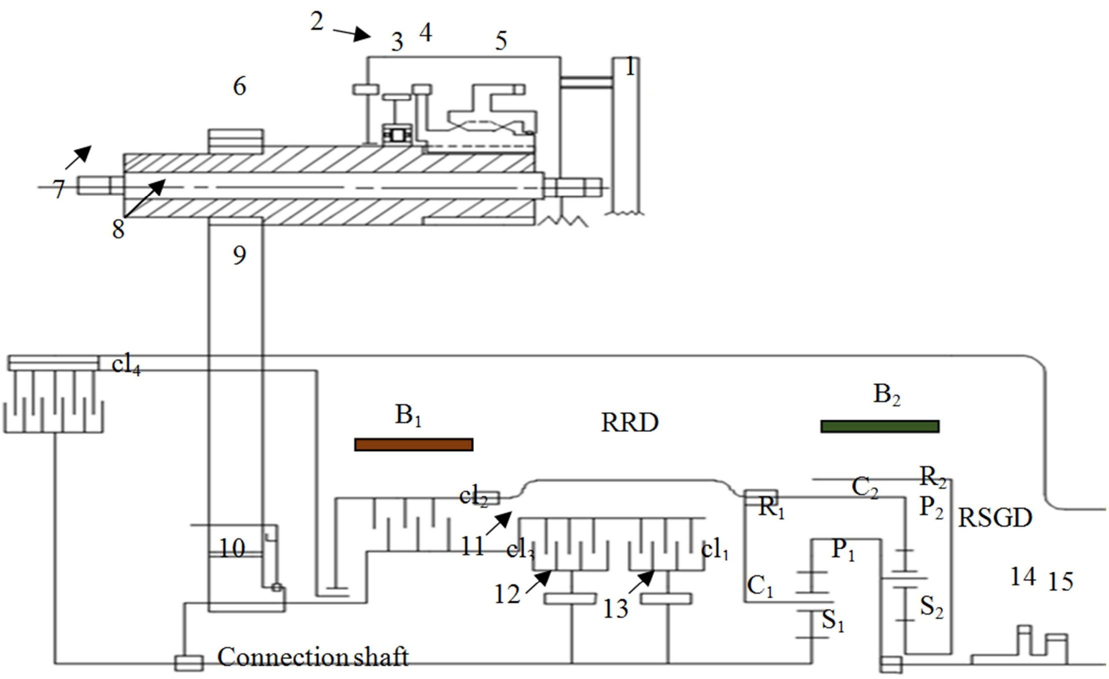

Figure 1. *Transaxle Four Speed Automatic Transmission system for the transverse wheel drive.*

Where:

1. flywheel. 2. torque converter cover with pump. 3.stator 4. turbine. 5. viscous friction clutch. 6. drive sprocket. 7. oil pump shaft. 8. Input Hollow Turbine Shaft (HTS). 9. Drive linked assembly. 10. driven sprocket. 11. Input gear sets shaft. 12. 2nd roller clutch (brakes S_1 at holding clutch). 13. Input sprang clutch (S_1 runs at holding clutch). 14. parking gear. 15. Final drive sun gear. RRD = reverse reaction drum. B_1, B_2 = 1-2band brake and reverse band brake respectively. RSGD = reaction sun gear drum. cl_1, cl_2, cl_3, = input (first), second, third clutches assembly. cl_4 = 4th clutch assembly as B_3. S_1, S_2 = sun gears. P_1, P_2 = planet gears. R_1, R_2 = ring gears. C_1, C_2 = carriers.

2.1. First Stage of Torque Converter Operation [1,4]

When there is a significant difference in speed between torque converter turbine and its pump, it is stall torque stage for torque converter. The maximum resistance torque (T_{R1}) on Hollow Turbine Shaft (HTS) based on equation 4 is appeared when the maximum drive line torque applied to move the vehicle from the rest. The maximum drive line torque (T_{max}) based on equation 4, comes from torque converter stall torque (T_1) formed on equation 5. T_{max} occurs from turbine torque over pump torque ratio Symbolizes him R_{Tstall} and formulate by R_T as setup in equation 2. Also, T_{max} equal to engine torque at maximum power ($T_{eatmax-p}$) multiplying by both highest reduction ratio value of transmission (i_{trmax}) and final drive ratio (i_{fd}). The applied torque (T_1) and T_{R1}, and loads (W_1 = weights of turbine (W_T) + viscous friction clutch (W_{vfc}), and W_2 = weights of drive link assembly (W_{dla}) + drive sprocket (W_{ds})) are exhibited on HTS through this stage instituted on Figure 2.

Through the stall torque stage, oil pump drive shaft has torque ($T_{p\text{-}shaft}$) and speed ($N_{p\text{-}shaft}$) equal to engine torque and speed because pump shaft is splined to torque converter cover. The other side of oil pump shaft has resistance torque ($T_{P\text{-}R}$) which is advent from rotational torque of oil pump rotor and pump vanes however rotor is splined to pump shaft. $T_{P\text{-}R}$ value is depending on required line pressure which becomes main supply of fluid to various transmission shift components and hydraulic circuits in the transaxle.

$$T_{R1} = T_{max} = R_{Tstall}*T_{eatmax-p}*i_{trmax}*i_{fd} \quad (4)$$

$$T_1 = R_{Tstall}*T_{eatmax-p} \quad (5)$$

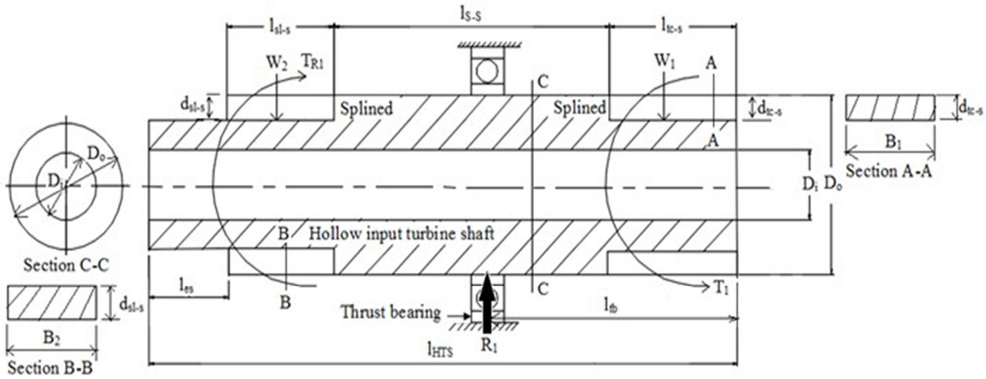

Figure 2. HTS Model at Torque Converter Stall and Maximum Vehicle Drive Line Torques

2.2. Second Stage of Torque Converter Operation [1, 4]

The *second stage* is started at the design point of torque converter ((ω_T/ω_P) ≈ 0.4). Through this period torque converter lock-up clutch is activated, then turbine and pump are connected to torque converter cover which bolted to engine flywheel. The angular velocities ((ω_T/ω_P) = 1), torques of both torque converter pump (T_P), turbine (T_T), applied torque (T_2) and $T_{eatmax-p}$ are equaled as formed in equation 6. The resistance torque (T_{R2}) on HTS are manifested in equation 7 when the vehicle doesn't need high torque to move. Minimum drive line torque (T_{min}) comes from engine torque at pump and turbine speeds are equals ($\omega_E = \omega_P = \omega_T$) multiplying by lowest reduction ratio value of transmission (i_{trmin}) and final drive reduction ratio (i_{fd}). T_2, T_{R2}, and loads (W_3 = weights of torque converter (W_{tc}) + viscous friction clutch (W_{vfc}) + flywheel (W_{fw}), and W_2) are modeled on HTS through this stage as is evidenced in Figure 3.

$$T_2 = T_P = T_T = T_{eatmax-p} \quad \text{at } R_T = 1 \quad (6)$$

$$T_{R2} = T_{min} = T_{eatmax-p}*i_{trmin}*i_{fd} \quad \text{at } R_T = 1 \quad (7)$$

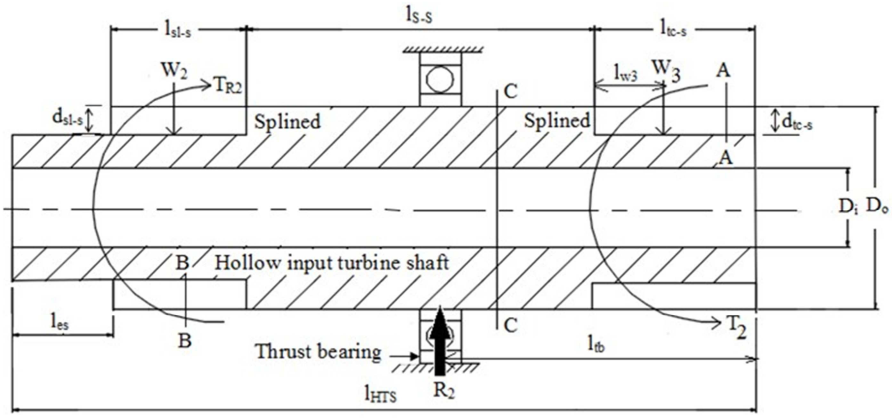

Figure 3. HTS Model at the Equality of Torque Converter and Engine Torques, and Minimum Vehicle Drive Line Torque

3. Critical Stress Combinations for HTS [8-10]

Figure 2 and Figure 3 show two models of HTS with different loads and torques. The models are both sides overhanging beams with one support. In Figure 2, the support reaction R_1 is equal the summation of weights W_1 and W_2, however, support reaction R_2 in Figure 3 is equal the summation of weights W_2 and W_3. The both two models of HTS consist of two parts have spline and two others without spline.

In the first stage of variation in turbine torque, the front spline of HTS ($l_{tc\text{-}s}$) Figure 2 affected manly torsion moment (no bending) which comes from torque converter stall torque and the rear spline influenced manly by T_{R1} (no bending) which moves vehicle from rest. W_1 cause bending moment (M_{b1}) from front till support point of HTS and W_2 occasion bending moment (M_{b2}) from front till support point of HTS.

Also, from W_1 to W_2 shear force ($F_{s\text{-}s\text{-}1}$) take place and on the part between front and rear spline parts torsion moment (T_3) happens. While In the second stage of variation in turbine torque, the front spline of HTS ($l_{tc\text{-}s}$) Figure 3 affected manly torsion moment (no bending) which comes from engine torque at maximum power and the rear spline influenced manly by T_{R2} (no bending) which moves vehicle at lowest need torque. W_3 cause bending moment (M_{b3}) from front until support point of HTS and W_2 occasion bending moment (M_{b2}) from front to support point of HTS. Moreover, from W_3 to W_2 shear force ($F_{s\text{-}s\text{-}2}$) take place and on the part between front and rear spline parts torsion moment (T_4) happens.

3.1. Torsion Stress (Inclined Crack)

The computing of torsional stress for HTS means that: a. torsion stresses $\tau_{tc\text{-}s}$, and $\tau_{sl\text{-}s}$ for two splined parts which can be obtained from equation (8) for $l_{tc\text{-}s}$ and equation (9) for $l_{sl\text{-}s}$, with considering $d_{tc\text{-}s}$ and $d_{sl\text{-}s}$ are equaled, b. torsion stress $\tau_{s\text{-}s}$ for $l_{s\text{-}s}$ part from end of $l_{tc\text{-}s}$ to end of $l_{sl\text{-}s}$ can be got from equation (10).

$$\tau_{tc\text{-}s1} = K_{sf} * T1/W_{tc\text{-}s} \quad \text{stage one}$$

$$F_{s1} = 2T_1/(D_m * n_{tc\text{-}s}) \quad \text{stage one}$$

$$\tau_{tc\text{-}s2} = K_{sf} * T_2/W_{tc\text{-}s} \quad \text{stage two}$$

$$F_{s2} = 2T_2/(D_m * n_{tc\text{-}s}) \quad \text{stage two} \qquad (8)$$

$$\tau_{sl\text{-}s1} = K_{sf} * T_{R1}/W_{sl\text{-}s} \quad \text{stage one}$$

$$F_{sR1} = 2T_{R1}/(D_m * n_{sl\text{-}s}) \quad \text{stage one}$$

$$\tau_{sl\text{-}s2} = K_{sf} * T_{R2}/W_{sl\text{-}s} \quad \text{stage two}$$

$$F_{sR2} = 2T_{R2}/(D_m * n_{sl\text{-}s}) \quad \text{stage two} \qquad (9)$$

$$\tau_{s\text{-}s1} = K_{sf} * (T_{R1} - T_1)/W_{s\text{-}s} \quad \text{stage one}$$

$$F_{s\text{-}s\text{-}1} = 2(T_{R1} - T_1)/D_{m1} \quad \text{stage one}$$

$$\tau_{s\text{-}s2} = K_{sf} * (T_{R2} - T_2)/W_{s\text{-}s} \quad \text{stage two}$$

$$F_{s\text{-}s\text{-}2} = 2(T_{R2} - T_2)/D_{m1} \quad \text{stage two} \qquad (10)$$

Where:

$W_{tc\text{-}s} = W_{sl\text{-}s} = 2*(I_X + I_Y)/D = 4 * I_x/D$ at $D = (D_o - 2d_{tc\text{-}s})$, $I_x = (\pi/64)(D^4 - D_i^4)$, $I_X = I_Y$ $W_{tc\text{-}s} = ((\pi/16)(D^4 - D_i^4))/D = 0.2(D^4 - D_i^4)/D$	Polar moment of inertia for splined parts
$W_{s\text{-}s} = 0.2(D_o^4 - D_i^4)/D_o$	Polar moment of inertia for part from front spline to rear spline
D_o	Outer diameter of HTS
D_i	Inner diameter of HTS
D	Minor diameter of splined parts
$D_m = (D_o + D)/2$	Mean diameter for cross section of spline
$D_{m1} = (D_o + Di)/2$	Mean diameter for cross section of $l_{s\text{-}s}$
F_{s1}, F_{s2}, F_{sR1}, F_{sR2}	Contact forces from torsion (splined parts) for one tooth
$F_{s\text{-}s\text{-}1}$, $F_{s\text{-}s\text{-}2}$	Tangential forces from torsion
$K_{sf} = 3.5$ at $I_{hub} > I_{shaft}$	Fatigue stress concentration factor for torsion
I_{hub}	Mass moment of inertia for hub
I_{shaft}	Mass moment of inertia for shaft

3.2. Shear Stress (Longitudinal and Transverse Cracks)

Moreover, when HTS is loaded by twist force, shearing force at section always parallel to tangential force (circumference force or contact force from torsion). For spline parts of HTS equations 8 and 9 is studied the contact forces which can be caused inclined crack for spline tooth however, equations 11 and 12 are deliberate resultant shearing stress (τ_{sl}) at tooth section to check longitudinal crack of HTS at splined parts (torque converter mounting part $\tau_{sl(tc\text{-}s)}$ and drive sprocket mounting part $\tau_{sl(sl\text{-}s)}$). Also, transverse crack ($\tau_{st(tc\text{-}s)}$ torque converter mounting part and $\tau_{st(sl\text{-}s)}$ drive sprocket mounting part) can be checked for splined parts which is coming from direct shearing force because of weights W_1, W_2, and W_3, and $\tau_{st(s\text{-}s)}$ transverse crack for $l_{s\text{-}s}$ because of reactions R_1 and R_2 by equations 13 and 14.

$$\tau_{sl(tc\text{-}s)1} = F_{ts11}/B_1 * l_{tc\text{-}s} \quad \text{stage one}$$

$$F_{ts11} = 2T_1/(D * n_{tc\text{-}s}) \quad \text{stage one}$$

$$\tau_{sl(sl\text{-}s)1} = F_{ts12}/B_2 * l_{sl\text{-}s} \quad \text{stage one}$$

$$F_{ts12} = 2T_{r1}/(D * n_{sl\text{-}s}) \quad \text{stage one} \qquad (11)$$

$\tau_{sl(tc\text{-}s)2} = F_{ts21} / B_1 * l_{tc\text{-}s}$ stage two

$F_{ts21} = 2T_2 / (D * n_{tc\text{-}s})$ stage two

$\tau_{sl(sl\text{-}s)2} = F_{ts22} / B_2 * l_{sl\text{-}s}$ stage two

$F_{ts22} = 2T_{r2} / (D * n_{sl\text{-}s})$ stage two (12)

$\tau_{st(tc\text{-}s)1} = F_1 / (B_1 * d_{tc\text{-}s} * n_{tc\text{-}s})$ stage one

$\tau_{st(sl\text{-}s)1} = F_2 / (B_2 * d_{sl\text{-}s} * n_{sl\text{-}s})$ stage one & two

$\tau_{st(s\text{-}s)1} = 4F_{R1} / \pi * D^2_{m1}$ stage one (13)

$\tau_{st(tc\text{-}s)2} = F_3 / (B_1 * d_{tc\text{-}s} * n_{tc\text{-}s})$ stage two

$\tau_{st(s\text{-}s)2} = 4F_{R2} / \pi * D^2_{m1}$ stage two (14)

Where:

F_{ts11}, F_{ts12}, F_{ts21}, F_{ts22}	Twist forces from torque for one tooth of splined parts
B_1, B_2	Splined tooth width
$d_{tc\text{-}s} = d_{tc\text{-}s}$	Splined tooth depth

$F_1 = W_1 * g$, $F_2 = W_2 * g$, $F_3 = W_3 * g$, $F_{R1} = R_1 * g$, $F_{R2} = R_2 * g$

g	Acceleration of gravity = 9.81 m/s^2
$n_{tc\text{-}s}$, $n_{sl\text{-}s}$	Number of splined teeth for torque converter, and drive sprocket mounting parts on HTS

3.3. Bending Stress (Vertical Crack)

3.3.1. Portion of HTS Without Splined

The level of bending stress through portion $l_{s\text{-}s}$ of HTS certainly at middle point of the bearing is measured by two forces F_1 and F_2 multiply by their distances starting from forces applied points to HTS bearing support point as mentioned in equation (15) and equation (16) for stall torque stage (stage one) of torque converter. However, equation (16) and equation (17) is measured level of bending stress through same portion from two forces F_2 and F_3 multiply by their distances starting from forces applied points to middle of bearing this bending stress when torque converter lock-up clutch stage (second stage). Also, modulus of shaft section effects on bending stress value.

$б_{b1} = F_1 * (l_{tb} - (l_{tc\text{-}s}/2))/W_b$ stage one (15)

$б_{b2} = F_2 * (l_{HTS} - l_{tb} - l_{es} - (l_{sl\text{-}s}/2))/W_b$ stage one and two (16)

$б_{b3} = F_3 * (l_{tb} - (l_{tc\text{-}s}/2))/W_b$ stage two (17)

Where:

$W_b = 0.1(D_o^4 - D_i^4)/D_o$	Bending section modulus for portion without splined of HTS

3.3.2. Portion of HTS with Splined

The level of bending stresses $б_{bs1}$ and $б_{bs1,2}$ for $l_{tc\text{-}s}$ and $l_{sl\text{-}s}$ splined parts respectively is calculated slightly different for $l_{s\text{-}s}$ without splined part from that two forces $W_1 * g$ and $W_2 * g$ multiply by their distances starting from forces applied points to the end of $l_{tc\text{-}s}$ and $l_{sl\text{-}s}$ splined parts (stage one) as in equations 18 and 19. This is repeated for stage two by multiply forces $W_2 * g$ and $W_3 * g$ by stage one distances as in equation (19) and stage two bending stress $б_{bs2}$ for $l_{tc\text{-}s}$ equation (20). Also, modulus of shaft section effects on bending stress value and it doesn't same in stages HTS without splined and with splined.

$б_{bs1} = F_1 * l_{tc\text{-}s}/(2 * W_{bs1})$ stage one (18)

$б_{bs1,2} = F_2 * l_{sl\text{-}s}/(2 * W_{bs2})$ stage one and two (19)

$б_{bs2} = F_3 * l_{tc\text{-}s}/(2 * W_{bs1})$ stage two (20)

where:

$W_{bs1} = d_{tc\text{-}s} * B_1^2/6$	Bending section modulus for portion splined $l_{tc\text{-}s}$
$W_{bs2} = d_{sl\text{-}s} * B_2^2/6$	Bending section modulus for portion splined $l_{sl\text{-}s}$

4. Compound Stress [8-10]

4.1. Portion of HTS without splined

The reliable study of power transmitting shafts stress sources and critical stress combinations are supported on three steps of study for same cross section without splined. First; torsion stress characteristics were studied for given shaft section 3.1. Second; shear stresses were calculated section 3.2. Third; bending stresses were established section 3.3 Finally; compound stresses ($б_{eq\text{-}s\text{-}s1}$) stage one and compound stresses ($б_{eq\text{-}s\text{-}s2}$) stage two into without splined part of HTS can be calculated by equation (21) and equation (22) respectively.

$$б_{eq-s-s1} = \sqrt{(б_{b1} + б_{b2})^2 + 4((\tau_{s-s1})^2 + (\tau_{st(s-s)1})^2)} \quad \text{stage one} \qquad (21)$$

$$б_{eq-s-s2} = \sqrt{(б_{b2} + б_{b3})^2 + 4((\tau_{s-s2})^2 + (\tau_{st(s-s)2})^2)} \quad \text{stage two} \qquad (22)$$

4.2. Portion of HTS with Splined

Compound stresses ($б_{eq\text{-}tc\text{-}s1}$ and $б_{eq\text{-}sl\text{-}s1}$) can be calculated for two cross sections (A-A) and (B-B) stage one and Compound stresses ($б_{eq\text{-}tc\text{-}s2}$ and $б_{eq\text{-}sl\text{-}s2}$) can be calculated for two cross sections (A-A) and (B-B) stage two of HTS splined parts by equations (23 and 25) and equations (24 and 26) respectively after applying the same four steps in section 4.1.

$$б_{eq-tc-s1} = \sqrt{(б_{bs1})^2 + 4((\tau_{tc-s1})^2 + (\tau_{s1(tc-s)1})^2 + (\tau_{st(tc-s)1})^2)} \quad \text{stage one} \tag{23}$$

$$б_{eq-tc-s2} = \sqrt{(б_{bs2})^2 + 4((\tau_{tc-s2})^2 + \left(\tau_{s1(tc-s)2}\right)^2 + \left(\tau_{st(tc-s)2}\right)^2)} \text{ stage two} \tag{24}$$

$$б_{eq-sl-s1} = \sqrt{(б_{bs1,2})^2 + 4((\tau_{sl-s1})^2 + \left(\tau_{s1(sl-s)1}\right)^2 + \left(\tau_{st(sl-s)1}\right)^2)} \text{ stage one} \tag{25}$$

$$б_{eq-sl-s2} = \sqrt{(б_{bs1,2})^2 + 4((\tau_{sl-s2})^2 + \left(\tau_{sl(sl-s)2}\right)^2} \text{ stage two} \tag{26}$$

5. Basic Data for Study HTS Stresses [1-3, 9, 10]

Many data were collected to study HTS stresses. Table 1 has required data for the paper' case study. The data contents valuable useful engine, transmission, and final drive specifications. Also, diameters, lengths, number teeth of splined parts, teeth size of splined parts, weight of the effective components such as torque converter parts, drive link and sprocket assembly, section modulus for torsion, section modulus for bending all of them putted into table 1.

Table 1. *Required Data for Paper' Case Study.*

Symbol	Value	Symbol	Value
R_{Tstall}	1.6	D_{m1}	4.5 cm
$T_{eatmax-p} = T_2$	150.6 N.m	D_{ds}	8.6 cm
i_{trmax}	2.921	$l_{tc-s} = l_{dl-s} = l_{es}$	3.175 cm
i_{trmix}	0.705	n_{tc-s}	35
i_{fd}	2.385	n_{sl-s}	40
$T_{R1} = T_{max}$	1678.67 N.m.	n_{ds} in mesh	10
T_1	240.96 N.m.	$d_{tc-s} = d_{dl-s}$	4 mm
$T_{R2} = T_{min}$	253.22 N.m.	l_{HTS}	15.24 cm
W_1	5 kg	$W_{tc-s} = W_{sl-s}$	25 cm^3
W_2	8 kg	W_{s-s}	40.5 cm^3
W_3	18 kg	W_b	20.5 cm^3
R_1	13 kg	W_{bs1}	6 cm^3
R_2	26 kg	W_{bs2}	10.67 cm^3
K_{sf}	3.5	B_1	3 mm
D_o	6 cm	B_2	4 mm
D_i	3 cm	l_{s-s}	5.715 cm
D	5.2 cm	l_{t-b}	6.033 cm
D_m	5.6 cm	$[б_{eq}]$	900 Mpa

6. Results and Discussions

The turbine input shaft specifications of transaxle automotive automatic transmission from view of shape, dimension, number of its splined parts, hollow or solid, material, place of its support points, weight of mounted on it pats …. etcetera is very important to give correct models as are given in Figures 2 and 3. HTS models have many torques (T_1, T_2, T_{R1}, T_{R2}), normal loads (W_1, W_2, W_3) and two reaction forces (R_1, R_2) which gives many stresses and crack. Two stages of operation conditions for HTS were investigated. Both two stages of operation condition resulted many stresses which are written as equations in section 3. Torsion stress equations (8-10) study the attitudes of HTS under splined contact forces (F_{s1}, F_{s2}, F_{sR1}, F_{sR2}) and no splined tangential forces (F_{s-s-1}, F_{s-s-2}) which are resulted from turbine torques and vehicle operation conditions resistance torques. The contact and tangential forces examining to avoid limitation of cross sections inclined crack. Twist forces (F_{ts11}, F_{ts12}, F_{ts21}, F_{ts22}) parallel to tangential forces based on equations 11 and 12 and normal forces (F_1, F_2, F_{R1}, F_3, F_{R2}) based on equations 13 and 14 are from shearing stresses. Twist and normal forces examining to avoid limitation of cross sections longitudinal and transverse cracks. Contact, tangential, twist and normal forces for two stages of HTS summarized in table 2. The highest value of forces is F_{sR1} and F_{ts12} per tooth which resulted from resistant torque T_{R1}.

Calculation results of torsion stress equations 8 to 10 presented in table 3. It has total and per tooth torsion stress for three different cross sections (A-A, B-B, C-C) of HTS. The elevated value resulted from resistant torque T_{R1} on left side splined part. Results for total and per tooth shear stress equations 11 to 14 plotted in table 4 over sections (A-A, B-B, C-C) while total and per tooth shear stress equations 15 to 20 collected in table 5. Same resistance torque T_{R1} occurred highest shear stress and bending stress at cross section B-B. The compound stresses calculated from equations 21 to 26 and putted in table 6. As mentioned above, highest compound stress occurred at cross section B-B.

Table 2. *Contact, Tangential, Twist and Normal Forces.*

Contact, Tangential, Normal, Twist and Shearing Forces N		
Name	Total	Per tooth
F_{s1}	8605.8	245.88
F_{s2}	5378.45	153.67
F_{sR1}	56352.4	1498.81
F_{sR2}	9043.6	226.09
F_{s-s-1}	63898.22	--------
F_{s-s-2}	4560.89	--------
F_{ts11}	9267.69	264.79
F_{ts12}	64564.23	1614.1
F_{ts21}	5792.3	165.49
F_{ts22}	9739.23	243.48
F_1	49.05	1.5
F_2	78.48	1.962
F_{R1}	127.53	------
F_3	176.58	5.05
F_{R2}	255.06	-------

Table 3. *Torsion Stress for Three Cross-Sections.*

Torsion Stress N/mm^2		
Name	**Total**	**Per tooth**
τ_{tc-s1}	33.73	0.96
τ_{tc-s2}	21.08	0.6
τ_{sl-s1}	235.01	5.875
τ_{sl-s2}	35.45	0.886
τ_{s-s1}	124.25	-------
τ_{s-s1}	8.87	-------

Table 4. *Shear Stress for Three Cross-Sections.*

Shear Stress N/mm^2		
Name	**Total**	**Per tooth**
$\tau_{sl(tc-s)1}$	97.3	2.78
$\tau_{sl(sl-s)1}$	508.38	12.71
$\tau_{sl(tc-s)2}$	60.18	1.74
$\tau_{sl(sl-s)2}$	76.69	1.92
$\tau_{st(tc-s)1}$	4.09	0.117
$\tau_{st(sl-s)1}$	4.91	0.123
$\tau_{st(s-s)1}$	0.08	-----
$\tau_{st(tc-s)2}$	14.715	0.42
$\tau_{st(s-s)2}$	0.16	------

Table 5. *Bending Stress for Three Cross-Sections.*

Bending Stress N/mm^2			Bending Moment N mm	Section Modulus mm^3
$б_{b1}$	0.12		2180.52	20250
$б_{b2}$	0.17		3488.04	20250
$б_{b3}$	0.39		7849.86	20250
Name	Total	Per tooth	Total	Per tooth
$б_{bs1}$	129.78	3.71	1557.34	6
$б_{bs1,2}$	116.76	2.92	2491.74	10.67
$б_{bs2}$	467.2	13.35	5606.42	6

Table 6. *Compound Stress for Three Cross-Sections.*

Compound Stress N/mm^2		
Name	**Total**	**Per tooth**
$б_{eq-s-s1}$	248.5	---------
$б_{eq-s-s2}$	243.43	---------
$б_{eq-tc-s1}$	243.58	6.96
$б_{eq-tc-s2}$	285.19	8.15
$б_{eq-sl-s1}$	1126.25	28.16
$б_{eq-sl-s2}$	205.39	5.13

7. Conclusion

1. This paper focused on studying of stresses caused by the influence of vehicle operation conditions on transaxle automotive automatic transmission's turbine input shaft. Turbine input shaft is a hollow shaft has two splined parts one on his front and other at its end and in between no splined part. It translates the torque converter torque to transmission planetary gear sets by drive link and drive sprocket assemblies.
2. Two vehicle operation conditions investigated resulting different normal forces and torques on splined and un-splined parts cross sections of turbine shaft. *First operation condition* is at vehicle starts motion from rest. *Second operation condition* is at maximum vehicle speed. The torque converter torque is input torque applied on front splined part and vehicle traction torque simulates resistance torque on the end splined part, these torque changes at each operation condition. Also, normal force on front splined comes from turbine and viscous friction clutch once and in other comes from torque converter as a packed with engine flywheel while normal force on end splined part doesn't change and it comes from drive link and drive sprocket assemblies. So, reaction on turbine shaft's bearing support change according vehicle operation conditions.
3. All turbine shaft length has torsion, shearing, and bending stresses. These stresses studied for three cross sections (A-A, B-B, C-C). Finally, combined action of compound torsion, shearing, and bending has been considered in studying the stressed state of splined and un-splined parts of turbine input shaft.
4. By using real data for the paper cases, the concluded is the highest value of torsion, shear, bending, and compound stresses occurs for end splined cross section which can result all types of crack (inclined-longitudinal-transverse and vertical).

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