CALCULATION OF HIGH CONTACT RATIO SPUR GEAR MESH STIFFNESS AND LOAD SHARING RATIO USING MATLAB & EXCEL SPREAD SHEET

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ABSTRACT

The mathematical modelling of the dynamic analysis of gears has become important with increased demand for high speed machinery. Various mathematical models have been developed for fault diagnosis/conditioning monitoring of spur gear. There are many numerical methods available for finding out the spur gear mesh stiffness like, photo elasticity methods, FEA methods and some numerical methods. In the present study, spur gear mesh stiffness is calculated using mathematical and numerical method with matlab and MS Excel spread sheet. For calculating the mesh stiffness, bending stiffness, axial compression stiffness, Hertzian contact stiffness, shear stiffness and effect of fillet-foundation deflection on mesh stiffness are considered. Also, the load sharing ratio is considered for two teeth pairs in contact and three teeth pairs in contact and the calculation of mesh stiffness for high contact ratio spur gear pair is explained.

Keywords*: *Gearmesh Stiffness, High Contact Ratio, Load Sharing Ratio, Matlab, Spur Gear

I. INTRODUCTION

Gears are the mechanical components used to transmit power and motion from one shaft to another. Gear mesh stiffness is the crucial parameter to study the gear dynamics. There are various methods to compute gear mesh stiffness. Gear mesh stiffness consists of bending deflection, axial compression deflection, shear deflection, contact stiffness and fillet-foundation deflection. Some of the researchers don't include the fillet-foundation deflection for easy computation of spur gear mesh stiffness. Many researchers have worked and are working for the analysis of gears and their faults using vibration, acoustic and finite element analysis for the determination various parameters related to the gear pair. Kar and Mohanty [1] have performed experiments on multistage gearbox to study the fault diagnosis under transient loads using fourier transform. They used the vibration signals to for fault diagnosis and concluded that vibration signals from the gear box are noisy and the signal-to-noise ratio (SNR) is so low that feature extraction of signal components is very difficult. Jafarizadeh *et al.* [2] developed a new noise canceling method, based on time-averaging method for asynchronous input signals. The proposed method is implemented on Morlet wavelet signal and real test rig of Yahama motorcycle gearbox. Gadelmawla [3] have used computer vision technology to develop a non-contact and rapid measurement system for accurate measurement and inspection of spur gear parameters. He also, developed the Gear Vision software

to analyze the captured images and to perform the measurement and inspection processes using Microsoft visual C++ software. Chen and Shao [4] proposed an analytical approach to compute the mesh stiffness with modified tooth profile. They established the relationship between the gear tooth errors and the total mesh stiffness, load sharing among different tooth pairs in mesh and loaded static transmission errors (LSTE). Wu *et al.* [5] studied the effects of tooth crack on the vibration response of a one-stage spur gearbox. The total gear mesh stiffness is affected due to the growth in a tooth crack. They used a lumped parameter model to simulate the vibration response of the pair of meshing gears and compare the calculated statistical indicators affecting the change in the vibration response caused by the tooth crack. Lin and Zuo [6] introduced an adaptive wavelet filter based on Morlet wavelet. The wavelet function was optimized by using kurtosis maximization principle. Santosh Patil *et al.* [7] analyzed and determined the shape function to define the change in contact stresses along the line of action of the gear pair. Prabhu and Muthuveerappan [8] elaborated an idea to eliminate the unbalanced maximum fillet stress which improves the load-carrying capacity of gear drives. In this improved gear drives, the maximum fillet stress is unequal in both the gears. They achieved the success in providing the uniform fillet strength of the gear drive by changing the tooth thickness on pitch line and design the spur gear drives with uniform fillet strength. They also, analysed the effect of backup ratio, cutter tip thickness and addendum modification factors on the maximum fillet stress through FE method.

Raghuwanshi and Parey [9] used the photo elasticity technique to measure spur gear mesh stiffness. The variations in the stress intensity factor (SIF) and mesh stiffness have been quantified with angular displacements of the gears. M. Divandari *et al.* [10] developed a six degree-of-freedom nonlinear dynamic model including different gear errors and defects for investigation of effects of tooth localized defect and profile modifications on overall gear dynamics. Lin and Parker [11] analytically investigated the parametric instabilities from mesh stiffness variation in multi mesh, two stage gear trains. They also, examined the effects of mesh stiffness parameters on instabilities systematically. In this paper, spur gear mesh stiffness is calculated using mathematical and numerical method with matlab and MS Excel spread sheet. For calculating the mesh stiffness, bending stiffness, axial compression stiffness, Hertzian contact stiffness, shear stiffness and effect of fillet-foundation deflection on mesh stiffness are considered. Also, the load sharing ratio is considered for two teeth pairs in contact and three teeth pairs in contact and the calculation of mesh stiffness for high contact ratio spur gear pair is explained. The same numerical simulation can be performed for low contact ratio which is much simpler than that simulation for high contact ratio spur gear pair. The mathematical equations from literatures [12] – [18] are modelled in matlab and the single pair tooth mesh stiffness of pinion and gear is calculated along pinion roll angle.

The total gear mesh stiffness is then, calculated in MS Excel spread sheet using summation function. Simpson's 3/8 rule is used to calculate bending stiffness, axial compression stiffness and shear stiffness. The gear parameters used in this study are listed in Table 1.

II. MESH STIFFNESS CALCULATION OF SPUR GEAR PAIR [12-18]

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In this study, single tooth contact pairs, double tooth contact pairs and three tooth contact pairs of gear system are investigated. In single pair, two teeth are meshed and share equal force. In case of double pair, two pairs share the total force and four teeth are meshed simultaneously. In case of triple pair, three pairs share the total force and six teeth are mesh simultaneously. In case of single tooth contact pair, the total effective mesh stiffness consists of gear 1 and gear 2 tooth mesh stiffness. In the same way, for double teeth contact pairs and triple teeth contact pairs, total mesh stiffness is calculated by direct sum of single tooth gear mesh stiffness because of teeth pairs are considered in parallel. But, for a single tooth pair, gear and pinion teeth are considered in series combination. The instantaneous single tooth contact pair mesh stiffness can be written as:

$$K_{e(sp)} = \frac{1}{\frac{1}{K_1} + \frac{1}{K_h} + \frac{1}{K_2}} \qquad (1)$$

Where, K_1 and K_2 are pinion and gear tooth stiffnesses respectively and K_h is the Hertzian contact stiffness.

Table 1: Parameters of the gear-pinion set

Parameter	Pinion/Gear
Tooth shape	Standard involute
Material	Steel
Number of Teeth z	30/60
Young's modulus E (GPa)	210
Poisson ratio υ	0.3
Module m (mm)	3
Pressure angle α (^{0})	20
Tip clearance coefficient c^*	0.25
Addendum coefficient $h_a{}^*$	1.5
Face width L (mm)	10*m
Hub bore radius r_{int} (mm)	17.5
Contact Ratio ε	> 2

K_1 and K_2 are the series combination of bending stiffness (K_b), axial compression stiffness (K_a), shear stiffness (K_s) and the stiffness due to fillet-foundation deflection (K_f) and can be calculated as:

$$\frac{1}{K_1} = \frac{1}{K_{b1}} + \frac{1}{K_{s1}} + \frac{1}{K_{a1}} + \frac{1}{K_{f1}} \qquad (2)$$

And,

$$\frac{1}{K_2} = \frac{1}{K_{b2}} + \frac{1}{K_{s2}} + \frac{1}{K_{a2}} + \frac{1}{K_{f2}} \qquad (3)$$

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For multi pairs, the total effective mesh stiffness is parallel combination of instantaneous single pairs mesh stiffness in the direction of force that can be written as:

$$K(t) = \sum_{i=1}^{n} \frac{1}{\frac{1}{K_h}+\frac{1}{K_{a1}}+\frac{1}{K_{b1}}+\frac{1}{K_{s1}}+\frac{1}{K_{a2}}+\frac{1}{K_{b2}}+\frac{1}{K_{s2}}+\frac{1}{K_{f1}}+\frac{1}{K_{f2}}} \quad (4)$$

The Bending, Shear, Axial compression, Contact and Fillet-foundation deflections of the gear tooth are calculated from the following equations. These equations are based on the model developed by reference [12, 13].

The Bending stiffness of the gear tooth (K_b) [18] is given by

$$\frac{1}{K_b} = \int_{-\phi_1}^{\theta_{b1}} \frac{P_b(\tau)}{Q_b(\tau)}\, d\tau \quad (5)$$

Where

$$P_b(\tau) = 3\left[\cos\phi_1 \left\{\frac{h_1}{R_{b1}} - \cos\theta_{b1} - \cos\tau + (\theta_{b1} - \tau)\sin\tau\right\} - \frac{h_1}{R_{b1}}\sin\phi_1\right]^2 [(\theta_{b1} - \tau)\cos\tau] \quad (6)$$

$$Q_b(\tau) = 2E_1 W[\sin\tau + (\theta_{b1} - \tau)\cos\tau]^3 \quad (7)$$

The Axial compression stiffness (K_a) [18] is given by

$$\frac{1}{K_a} = \int_{-\phi_1}^{\theta_{b1}} \frac{P_a(\tau)}{Q_a(\tau)}\, d\tau \quad (8)$$

Where

$$P_a(\tau) = (\theta_{b1} - \tau)(\sin\phi_1)^2 \quad (9)$$

$$Q_b(\tau) = 2E_1 W[\sin\tau + (\theta_{b1} - \tau)\cos\tau] \quad (10)$$

The Shear stiffness of the tooth (K_s)[18] is given by

$$\frac{1}{K_s} = \int_{-\phi_1}^{\theta_{b1}} \frac{P_s(\tau)}{Q_s(\tau)}\, d\tau \quad (11)$$

Where

$$P_s(\tau) = [3(1+\upsilon_1)(\cos\phi_1)^2]\left[2(\theta_{b1} - \tau)\{\sin\tau + (\theta_{b1} - \tau)\cos\tau\} * \left\{(\theta_{b1} - \tau)(1 + (\cos\theta_{b1})^2) + \cos\theta_{b1}\sin\tau - \frac{(L_1 * \sin\tau)}{R_{b1}}\right\} + 3(\theta_{b1} - \tau)\tan\tau\sin\tau\left\{\frac{L_1}{R_{b1}} - \cos\theta_{b1} + (\theta_{b1} - \tau)\sin\tau\right\}^2\right] \quad (12)$$

$$Q_b(\tau) = 5E_1 W[\sin\tau + (\theta_{b1} - \tau)\cos\tau]^3 \quad (13)$$

The parameters $\phi_1, \theta_{b1}, L_1, h_1, R_{b1}$ (Base Circle radius) are represented in Fig. 1.

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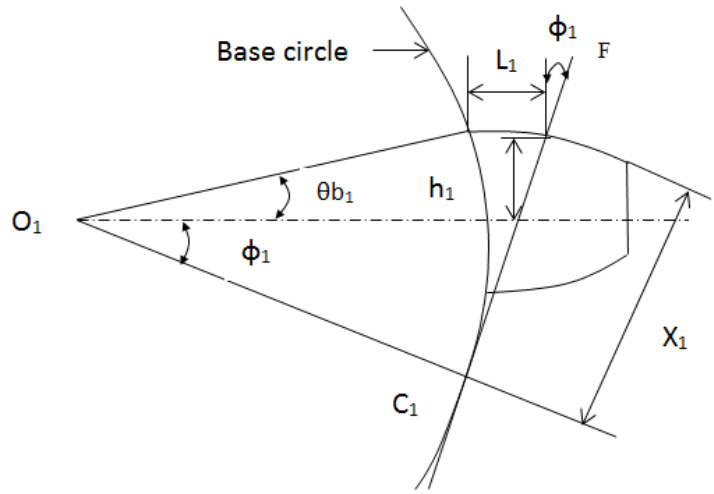

Fig. 1: Spur Gear Tooth Model (As A Non-Uniform Cantilever Beam) [18]

The fillet-foundation deflection also influences the stiffness of gear tooth. Sainsot et al. [14] derived the fillet-foundation deflection of the gear based on the theory of Muskhelishvili [15], and then, they applied it to circular elastic rings to derive an analytical formula reflecting the gear body-induced tooth deflections by assuming linear and constant stress variations at root circle. It can be calculated as [14]

$$\delta_f = \frac{(\cos\alpha_m)^2}{WE}\left[L^*\left(\frac{u_f}{S_f}\right)^2 + M^*\left(\frac{u_f}{S_f}\right) + P^*\{1 + Q^*(\tan\alpha_m)^2\}\right] \quad (14)$$

Where, W is the tooth width. u_f and S_f are given in Fig. 2 The coefficients L^*, M^*, P^*, Q^* can be approached by polynomial functions [17].

$$X_i^*(h_{fi}, \theta_f) = \frac{A_i}{\theta_f^2} + B_i h_{fi}^2 + C_i\frac{h_{fi}}{\theta_f} + \frac{D_i}{\theta_f} + E_i h_{fi} + F_i \quad (15)$$

X_i^* denotes the coefficients L*, M*, P*, and Q*. $h_{fi} = {}^{r_f}/_{r_{int}}$, r_f, r_{int} and θ_f are defined in Fig. 2, the values of A_i, B_i, C_i, D_i, E_i and F_i are given in Table 2.

The stiffness with consideration of gear fillet-foundation deflection can be obtained by

$$\frac{1}{K_f} = \frac{\delta_f}{F} \quad (16)$$

Table 2: Values of The Coefficients L*, M*, P*, And Q*. [17]

	A_i	B_i	C_i	D_i	E_i	F_i

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L*(h_{f1}, θ_f)	-5.574×10^{-5}	-1.9986×10^{-3}	-2.3015×10^{-4}	4.7702×10^{-3}	0.0271	6.8045
M*(h_{f1}, θ_f)	60.111×10^{-5}	28.1000×10^{-3}	-83.431×10^{-4}	-9.9256×10^{-3}	0.1624	0.9086
P*(h_{f1}, θ_f)	-50.952×10^{-5}	185.50×10^{-3}	0.0538×10^{-4}	53.3×10^{-3}	0.2895	0.9236
Q*(h_{f1}, θ_f)	-6.2042×10^{-5}	-9.0889×10^{-3}	-4.0964×10^{-4}	7.8297×10^{-3}	-0.1472	0.6904

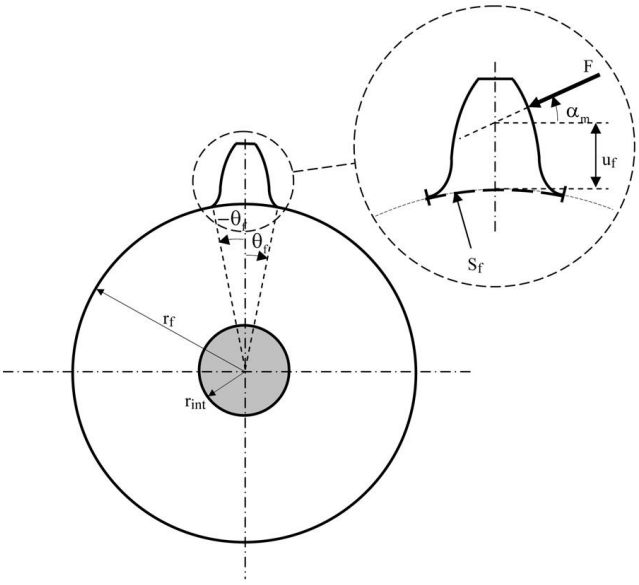

Fig. 2: Geometrical parameters for the fillet-foundation deflection [16,17]

From the results derived by Yang and Sun [12], the stiffness of Hertzian contact of two meshing teeth is constant along the entire line of action. It is independent of the contact position and the interpenetration depth between meshing teeth. The Hertzian contact stiffness is given by

$$K_h = \frac{\pi E_1 L}{4(1-\upsilon_1^2)} \qquad (17)$$

Where

- E is Modulus of Elasticity
- υ is Poisson Ratio

The locations and sizes of the two and three pairs of teeth contact zones can be determined from the contact ratio and base pitch of the gear pair. These zones are shown in Fig. 3. A_1-A_2, A_3-A_4, A_5-A_6 are the three pairs of teeth contact zones A_2-A_3, A_4-A_5 are two pairs of teeth contact zones.

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After determining the positions and widths of these contact zones, the corresponding pinion or gear roll angles are determined from the gear geometry, since each point on the common normal line can be mapped to a corresponding point on the base circle, using the involute properties of the spur gear tooth.

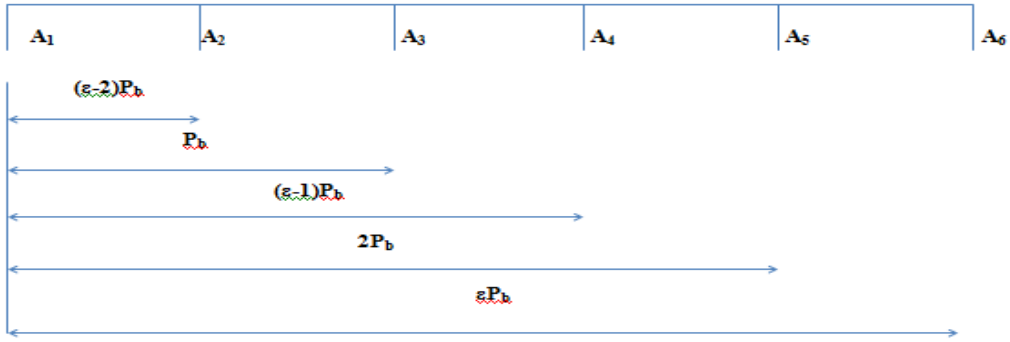

Fig. 3: Alteration of number of contact pairs

By using Fig. 4, The angle between the symmetrical line of tooth and line $\mathbf{O_1C_1}$ is given by

$$\phi_1 = \alpha - (\psi + \theta_{b1}) \qquad (18)$$

This is also known as inclination angle of load line with the normal to the symmetrical line of the tooth.
Where, α is pitch circle pressure angle. θ_{b1} is Half of the tooth thickness angle measured on the base circle of pinion which is calculated as:

$$\theta_{b1} = \frac{\pi}{2z_1} + \mathrm{inv}(\alpha) \qquad (19)$$

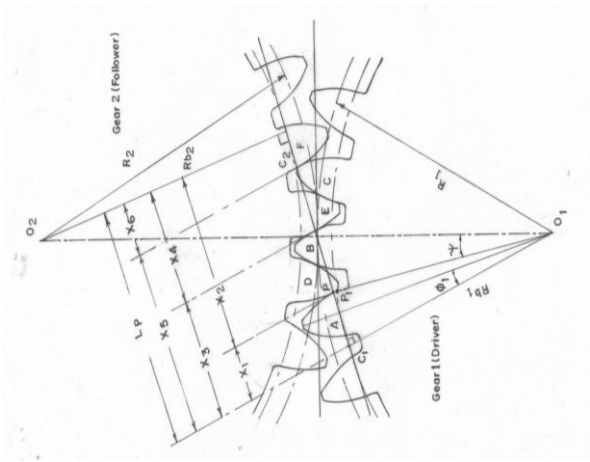

Fig. 4 Teeth meshing in high contact ratio gears [18]

And, ψ is the angle between the line joining gear centers O_1, O_2 and the line O_1P_1 where P_1 is the corresponding point of contact P is mapped on the base circle. The angle ψ for the reference points A_1, A_2, A_3, A_4, A_5 and A_6

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representing the locations of two and three pairs of teeth contact zones on the common normal line and they are represented as under.

$$\psi_{a1} = -[\alpha - (\angle C_1 O_1 P_1)] \quad (20)$$

$$\psi_{a2} = \psi_{a1} + (\varepsilon - 2) * {}^{2\pi}/_{z_1} \quad (21)$$

$$\psi_{a3} = \psi_{a1} + {}^{2\pi}/_{z_1}$$

(22)

$$\psi_{a4} = \psi_{a1} + (\varepsilon - 1) * {}^{2\pi}/_{z_1} \quad (23)$$

$$\psi_{a5} = \psi_{a1} + {}^{4\pi}/_{z_1} \quad (24)$$

$$\psi_{a6} = \psi_{a1} + \varepsilon * {}^{2\pi}/_{z_1} \quad (25)$$

The angle $\angle C_1 O_1 P_1$ is calculated as:

$$\angle C_1 O_1 P_1 = \tan\alpha - \left(\frac{(R_{a2} * \sin\alpha_{a2} - R_{b2} * \tan\alpha)}{R_{b1}}\right) \quad (26)$$

To find stiffness of the corresponding tooth of the gear 2, the subscripts in the above equations are to be changed from 1 to 2. $\boldsymbol{\phi_2}$ is calculated as

$$\phi_2 = \frac{X_2}{R_{b2}} - \theta_{b2} \quad (27)$$

$$X_2 = L_p - X_1 \quad (28)$$

$$X_1 = \frac{\pi}{2Z_2} + \mathrm{inv}(\alpha_0) \quad (29)$$

Similarly, for the second pair of teeth $\mathbf{X_1}$, $\mathbf{X_2}$, ϕ_1 and ϕ_2 are to be replaced by $\mathbf{X_3}$, $\mathbf{X_4}$, ϕ_3 and ϕ_4 and for the third pair by $\mathbf{X_5}$, $\mathbf{X_6}$, ϕ_5 and ϕ_6 respectively, where

$$X_3 = X_1 + P_b \quad (30)$$

$$X_4 = L_p - X_3 \quad (31)$$

$$X_5 = X_1 + 2P_b \quad (32)$$

$$X_6 = L_p - X_5 \quad (33)$$

$$\phi_3 = \frac{X_3}{R_{b1}} - \theta_{b1} \quad (34)$$

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$$\phi_4 = \frac{X_4}{R_{b2}} - \theta_{b2} \quad (35)$$

$$\phi_5 = \frac{X_5}{R_{b1}} - \theta_{b1} \quad (36)$$

$$\phi_6 = \frac{X_6}{R_{b2}} - \theta_{b2} \quad (37)$$

To calculate the individual tooth load, the profile modification and tooth error are not considered. When three pairs of teeth are in contact, then the load shared by first teeth is given by equation (38), load shared by second teeth is given by equation (39) and load shared by third teeth is given by equation (40).

$$F_1 = F\frac{K_{e1}}{K_{e1}+K_{e2}+K_{e3}} \quad (38)$$

$$F_2 = F\frac{K_{e2}}{K_{e1}+K_{e2}+K_{e3}} \quad (39)$$

$$F_3 = F\frac{K_{e3}}{K_{e1}+K_{e2}+K_{e3}} \quad (40)$$

When two pairs are in contact, for example, teeth **A** and **B** on the driving teeth remain in contact with teeth **D** and **E** on the driven, then the load shared by first teeth is given by equation (35) and lad shared by second teeth is given by equation (36).

$$F_1 = F\frac{K_{e1}}{K_{e1}+K_{e2}} \quad (41)$$

$$F_2 = F\frac{K_{e2}}{K_{e1}+K_{e2}} \quad (42)$$

The calculation of contact ratio [19] for a pair of two external spur gears is

$$\varepsilon = \frac{z_1}{2\pi} * (\tan\alpha_{a1} - \tan\alpha) + \frac{z_2}{2\pi} * (\tan\alpha_{a2} - \tan\alpha) \quad (43)$$

Where,

$$\cos\alpha_{a1} = \frac{z_1 * \cos\alpha}{2R_{a1}} \quad (44)$$

And

$$\cos\alpha_{a2} = \frac{z_2 * \cos\alpha}{2R_{a2}} \quad (45)$$

III. METHODOLOGY

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A program is created for calculating the mesh stiffness using Matlab programming code. The input parameters for the program are pressure angle, module, addendum coefficient, number of teeth on pinion, gear ratio, modulus of elasticity, Poisson's ratio and gear hub diameter. By using these basic parameters, the other parameters such as base circle diameter, pitch circle diameter, addendum circle diameter, base pitch, contact ratio etc. are calculated. The contact ratio is calculated to know the meshing nature of spur gears during transmission operations. The pinion roll angle, calculated using the contact ratio is the angle which constitutes the one mesh cycle of gear transmission. The locations and sizes of contact zone are calculated using contact ratio and base pitch. This information is used for deciding the contact zones for different conditions. The conditions are:

- Single tooth contact zone
- Double tooth contact zone
- Triple tooth contact zone

The position of starting point from tooth middle axis is calculated using equation (18) to equation (29) for first teeth contact zone for both the gears. The positions of next zones are calculated by equation (30) to equation (37). These equations are developed upto triple tooth contact zones. A large number of points are imagined along the total contact zone for one mesh cycle. These points are taken to know spur gear mesh stiffness behavior along the complete mesh cycle. The pinion roll angle is used as a quantity for selecting these points for which mesh stiffness is to be calculated. The formulas for first three stiffnesses equation (5), equation (8) and equation (11)] depend on tooth geometry. These values are difficult to solve by the general integration method. So, Simpson's 3/8 rule is selected to solve these integrating quantities. The contact stiffness and fillet-foundation deflection stiffness are constant throughout the complete rotation and easy to calculate by general mathematics. Finally, the four stiffnesses are calculated by using equation (1), equation (2), equation (3) and equation (4). These stiffnesses are as follows:

- Pinion tooth stiffness
- Gear tooth stiffness
- Single tooth mesh stiffness
- Total effective mesh stiffness

The load sharing ratio is calculated using equation (38) – equation (40) for triple tooth contact zone and equation (41) – equation (42) double tooth contact zone. The Fig. 5 shows the command window for entering the input parameters. After entering these parameters, the program is run to calculate the parameters such as base pitch, contact ratio, stiffnesses as shown in Fig. 6. P_b is result for base pitch. e is result for contact ratio. K_p is result for pinion tooth stiffness. K_g is result for gear tooth stiffness. K is result for single tooth mesh stiffness. To calculate the total effective mesh stiffness, these results are exported to MS excel spread sheet. The Fig. 7 shows the exported results in MS excel spread sheet.

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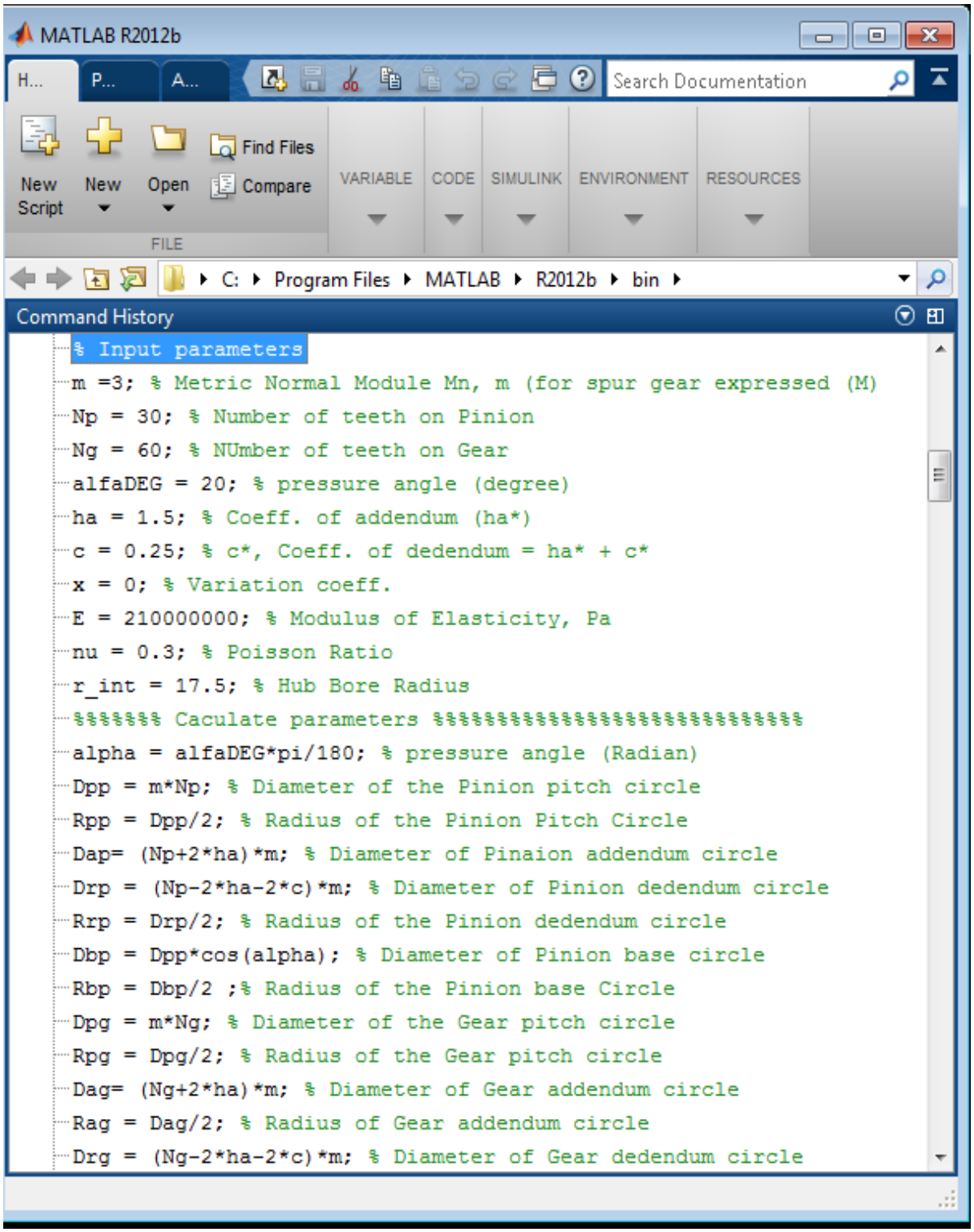

Fig. 5: Input Command window

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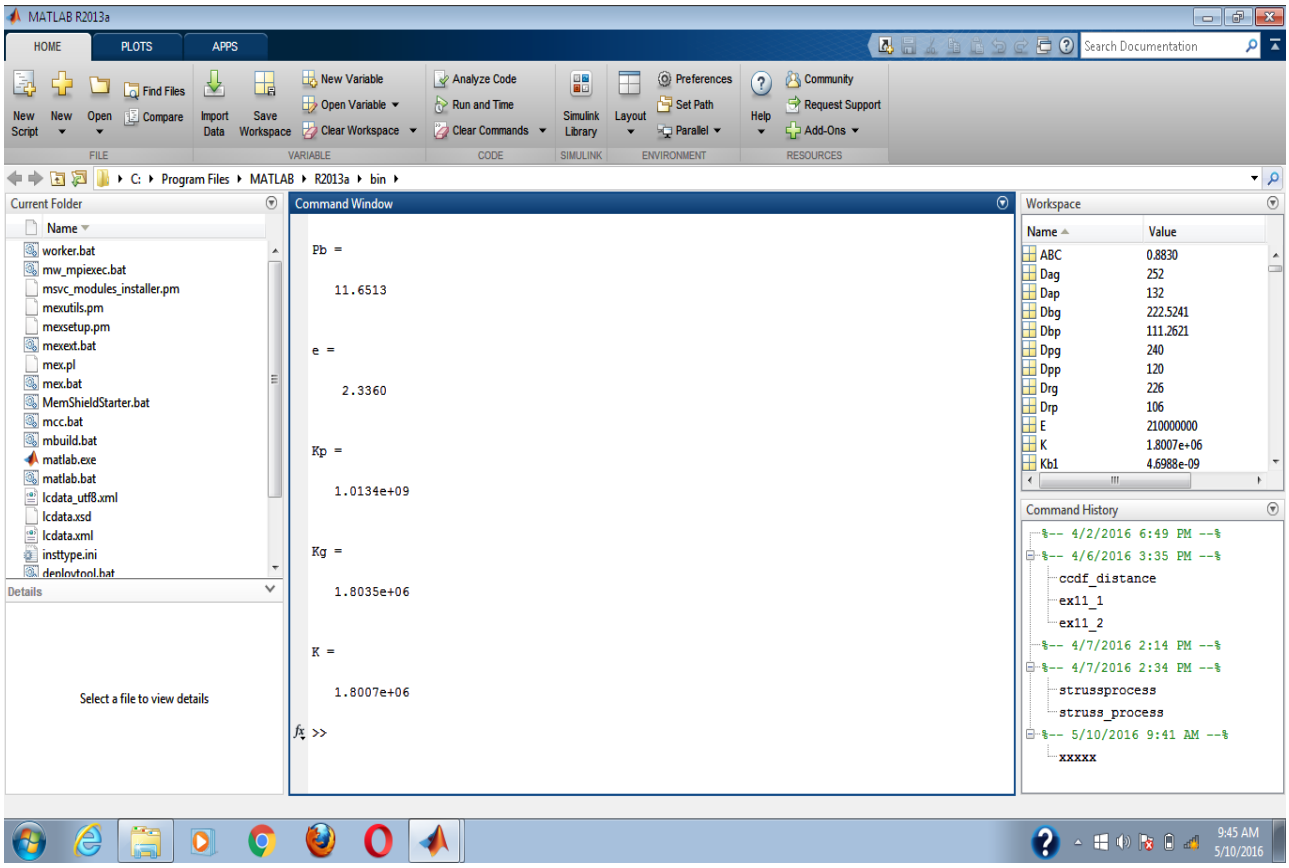

Fig. 6: The Matlab Output Screen

In Fig. 7, the column (A) shows the pinion roll angles corresponding to the points along the total contact zone for one mesh cycle. This also, shows the pinion tooth stiffness, gear tooth stiffness, first tooth pair mesh stiffness, second pair tooth mesh stiffness, third pair, total effective mesh stiffness and load sharing ratio in column (B), (C), (D), (E), (F), (H) and (J) respectively. The pinion roll angle 0^0 to 7.008^0 are the range for the three teeth contact zone and column (B) to column (J) are the results corresponding to this range. Further, this angle 7.008^0 to 12^0 is for two teeth contact zone and 12^0 to 19.008^0 for again three teeth contact. The pinion roll angle range 0^0 to 19.008^0 is the total pinion roll angle variation for a complete mesh cycle. The total pinion roll angle is calculated by the formula $\left(\theta_e = \varepsilon * \frac{360}{z_1}\right)$ for pinion for a complete mesh cycle. Pinion tooth stiffness and gear tooth stiffness for first pair are calculated using equation (2) and equation (3) respectively and listed in column (B) and column (C) respectively. Single tooth mesh stiffness for first pair is calculated using Eq. (1) and listed in column (D). Similarly, single tooth mesh stiffness is calculated for second and third pair by using Eq. (1) and listed in column (E) and column (F) respectively. Total effective mesh stiffness is calculated by direct summation of column (D) data, column (E) data and column (F) data and listed is column (H). Load sharing ratio is calculated by dividing column (D) data by column (H).data and listed in column (J). The flow chart (as shown in Fig. 8) shows the method used to calculate the total effective mesh stiffness and load showing ratio of the work.

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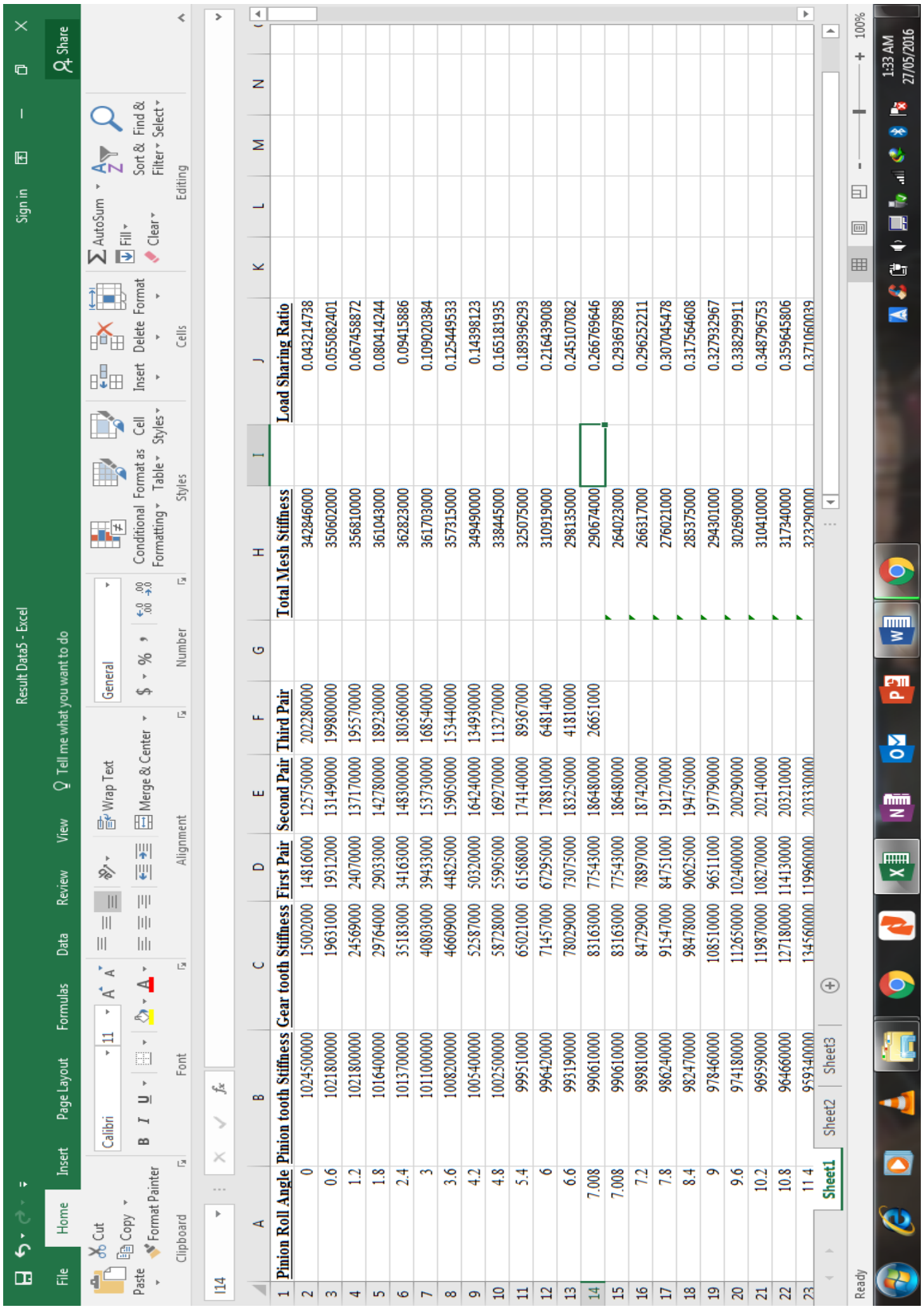

Pinion Roll Angle	Pinion tooth Stiffness	Gear tooth Stiffness	First Pair	Second Pair	Third Pair	Total Mesh Stiffness	Load Sharing Ratio
0	1024500000	15002000	14816000	125750000	202280000	342846000	0.043214738
0.6	1021800000	19631000	19312000	131490000	199800000	350602000	0.055082401
1.2	1021800000	24569000	24070000	137170000	195570000	356810000	0.067458872
1.8	1016400000	29764000	29033000	142780000	189230000	361043000	0.080414244
2.4	1013700000	35183000	34163000	148300000	180360000	362823000	0.09415886
3	1011000000	40803000	39433000	153730000	168540000	361703000	0.109020384
3.6	1008200000	46609000	44825000	159050000	153440000	357315000	0.125449533
4.2	1005400000	52587000	50320000	164240000	134930000	349490000	0.14398123
4.8	1002500000	58728000	55905000	169270000	113270000	338445000	0.165181935
5.4	999510000	65021000	61568000	174140000	89367000	325075000	0.189396293
6	996420000	71457000	67295000	178810000	64814000	310919000	0.216439008
6.6	993190000	78029000	73075000	183250000	41810000	298135000	0.245107082
7.008	990610000	83163000	77543000	186480000	26651000	290674000	0.266769646
7.008	990610000	83163000	77543000	186480000		264023000	0.293697898
7.2	989810000	84729000	78897000	187420000		266317000	0.296252211
7.8	986240000	91547000	84751000	191270000		276021000	0.307045478
8.4	982470000	98478000	90625000	194750000		285375000	0.317564608
9	978460000	108510000	96511000	197790000		294301000	0.327932967
9.6	974180000	112650000	102400000	200290000		302690000	0.338299911
10.2	969590000	119870000	108270000	202140000		310410000	0.348796753
10.8	964660000	127180000	114130000	203210000		317340000	0.359645806
11.4	959340000	134560000	119960000	203330000		323290000	0.371060039

Fig. 7: MS Excel Spread Sheet Output Screen

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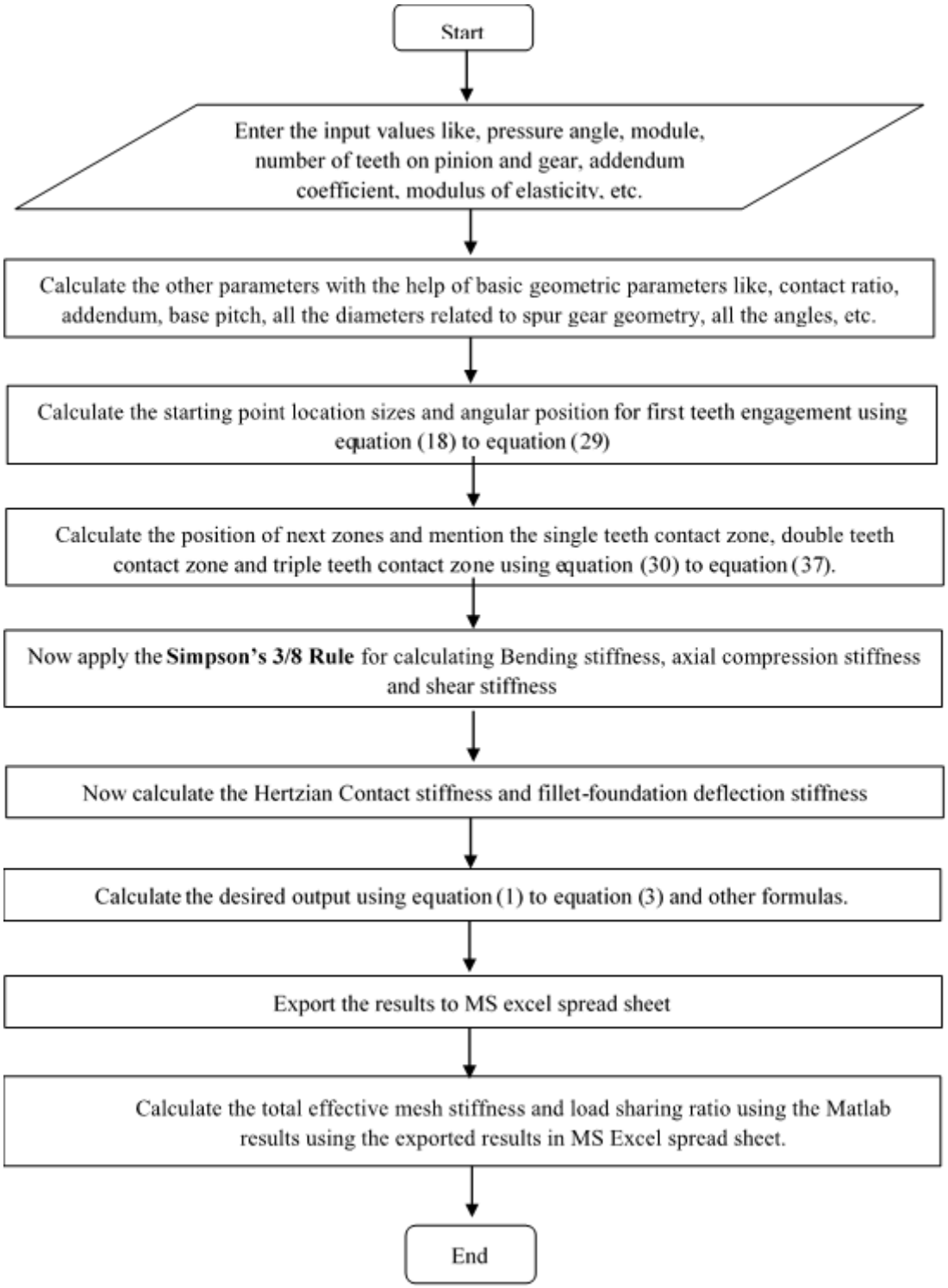

Fig. 8: The flow chart of the method used

IV. CONCLUSION

In this work, an analytical approach is used. Pinion tooth stiffness and gear tooth. Pinion tooth stiffness, gear tooth stiffness and single tooth mesh stiffness are calculated by using a computer program in matlab. In the matlab program, Simpson 3/8 rule is used for calculating bending stiffness, shear stiffness and axial compression stiffness. Total effective mesh stiffness and load sharing ratio are calculated by using MS Excel spread sheet. This is performed for the ideal spur gear profile. It can be used by incorporating spur gear profile defects such as – tooth crack, pitch error etc. it can also be used for modified spur gear profile by considering the modified geometric values.

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