DESIGN AND CONTACT STRESS ANALYSIS OF HELICAL GEAR FOR LIGHT-WEIGHT CAR

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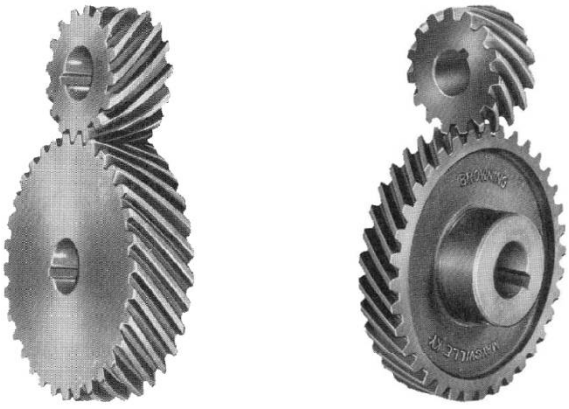
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Abstract- Gear is the most essential element of power transmission. One of the main gear tooth failure is pitting which is surface fatigue failure due to many repletion of high contact stresses occurring in the gear tooth. The work of this paper is to reduce the contact stress happing often on gear by changing the different values of face width. Contact failure in gears can be predicted by calculating the contact stress with AGMA (American Gear Manufacturing Association) equation and also analyzing with ANSYS. The model of helical gear pairs are generated by using SolidWorks software and ANSYS is used for numerical analysis. Ultimately, the results obtained from ANSYS are presented and compared with theoretical values. In this analysis, the face width 21mm which has 493.27 MPa von-Mises stress is selected because of the weight reduction.

Keywords- AGMA, ANSYS, contact stress, face width, helical gear

I. INTRODUCTION

Gearing is one of the most effective methods transmitting power and rotary motion from the source to its application with or without change of speed or direction. In automobile, highly reliable and lightweight gears are essential. Furthermore the best way to diminution of noise in engine requires the fabrication of silence gear system. Noise reduction in gear pairs is especially critical in the rapidly growing today's technology since the working environment is badly influenced by noise. The most successful way of gear noise reduction is attained by decreasing of vibration related with them. Helical gears are currently being used increasingly as a power transmitting gear owing to their relatively smooth and silent operation, large load carrying capacity and higher operating speed. Designing highly loaded helical gears for power transmission systems that are good in strength and low level in noise necessitate suitable analysis methods [10]. There are two basic types of helical gears parallel and crossed as shown in Figure 1. Helical gears are the modified forms of spur gears, in which all the teeth are cut at a constant angle, known as helix angle. Helical gears are also employed to transmit power between two shafts parallel to the axis [4]. Sarfraz Ali N.Quadri [1] have done his research work on "Contact Stress Analysis of Involute Spur Gear under Static Loading". In this paper, they have analysed a spur gear pair from lathe machine for contact stresses under static loading conditions using Hertz theory with the effect of various modules. The analytical method of calculating gear contact stresses are compared with calculated contact stresses using ANSYS. Finally, the higher the module, the lower the contact stress. S.K. Sureshkumar (2015) [2] investigated a research in which analysis of helical gear pairs. The author carried out an attempt to study the contact stresses among the helical gear pairs of different materials such as Steel, Cast Iron, and Aluminium under static conditions by using a 3D finite element method. J.Venkatesh [3] is analysed the helical gear based on structural analysis of high speed. In this paper, he modified the number of teeth to reduce the bending stress by using AGMA. In contact stress, the different value of helix angle is changed to minimize the contact stress. And then, two methods bending and contact stress results are compared with each other. Finally, the more the number of teeth, the more the bending stress. In contact stress, the more the helix angle, the less the stress. Puttapaka Nagaraju (2014) [4] presented a paper a suggest that, the bending stress of helical gear are analysed. In this study, bending stress were calculated with different face width by using Lewis equation and these results are compared with the results obtained in static analysis in ANSYS. The author concluded that the maximum bending stress decreases with increasing the face width of the gear.



Helical gear with parallel shaft

Cross helical gear at right angle

Figure 1 Gear tooth terminology [11]

In 2014, at IJERA, Raghava Krishna Saneer.B [5] presented contact stress analysis of modified helical gear. The analytical study is based on Hertz's equation. This paper have analysed by varying the geometrical profile of the teeth and to find the change

in contact stresses between gears. Pravin M.kinge (2012) [6] presented a paper to suggest that, three modifications in the design of gear were done and after that again stress analysis of the modified gear carried out. The three design modifications were done the edges of the gear teeth were tapered by an angle of 20, making groove in the gear wheel and making holes at the roots of the gear teeth. Gear tooth terminology is shown in Figure 1.

II. DESIGN CALCULATION OF THE HELICAL GEAR PAIR

The design calculation of the helical gear pair has the following steps. The material for the gear pairs are taken as AISI 5160 OQT400.

Table 1 shows the parameters considered for design a helical gear.

Table 1: Parameters considered for design a helical gear.

Design Parameter	Value	Unit
Vehicle gross weight	700	kg
Power (P)	20	kW
R.P.M (N_p)	6500	rpm
Helix angle (ψ)	23	degree
Pressure angle (ϕ)	20	degree
Modulus of Elasticity (E)	207	GPa
Ultimate Strength	2220	MPa
Yield Strength	1790	MPa
Brinell Hardness (BHN)	627	-
Number of teeth of pinion	11	mm
Number of teeth of gear	34	mm

When two mating gears are to be made of the same material, the smaller gear (pinion) will be the weaker. Both pinion and gear are same material.

1. Calculation of Torque (M_t)

The torque of gears in (Nm) can be calculated as

$$M_t = \frac{9550 \times P}{N_p} \quad (1)$$

2. Calculation of pitch line velocity (V)

The pitch line velocity can be calculated by

$$V = \frac{\pi \times D_p \times N_p}{60} \quad (2)$$

3. Unknown diameter case

The actual induced stress can be calculated by using Lewis equation.

$$S_{ind} = \frac{2M_t}{m^3 k\pi^2 y_p n_p \cos\psi} \quad (3)$$

$$k = 6, n_f = \frac{n}{\cos^3\psi} \rightarrow y_p = 0.088 \text{ (from the table)}$$

4. Calculation of allowable stress,

Allowable stress can be calculated by

$$S_{all} = S_0 \times \left[\frac{5.6}{5.6 + \sqrt{V}}\right] \quad (4)$$

5. Calculation of number of teeth,

The number of teeth can be calculated as

$$n = \frac{D}{m} \quad (5)$$

6. Strength Check,

Compare S_{all} and S_{ind} (6)

If $S_{all} > S_{ind}$, Design is satisfactory.

If not so, keeping on calculating by increasing the module until it is satisfied need to be done

7. Calculation of the face width of helical gear, b,

The face width of helical gear in (mm) can be calculated as

$$b_{min} = k_{red} \times \pi \times m \quad (7)$$

$$b_{max} = k \times \pi \times m \quad (8)$$

$$k_{red} = k_{max} \times \frac{S_{ind}}{S_{all}}$$

9. Dynamic Check,

a. Calculation of transmitted load, F_t

The transmitted load in (N) can be calculated as

$$F_t = \frac{2M_t}{D_p} \quad (9)$$

b. Calculation of dynamic load, F_d

The dynamic load can be determined as

$$F_d = F_t + \frac{21V(bC\cos^2\psi + F_t)\cos\psi}{21V + \sqrt{(bC\cos^2\psi + F_t)}} \quad (10)$$

c. Calculation of limiting endurance load, F_0

The limiting endurance load can be determined as

$$F_0 = S_0 b y_p \pi m \cos(\psi) \quad (11)$$

d. Calculation of limiting wear load, F_w

The limiting wear load can be determined as

$$F_w = \frac{D_p \times b \times K \times Q}{\cos^2\psi} \quad (12)$$

where $S_{es} = (2.75BHN - 70)\ MN/m^2$

$$K = \frac{S_{es}^2 \times \sin\phi_n}{1.4} \times \left[\frac{2}{E}\right]$$

$$Q = \frac{2 \times D_g}{D_g + D_p}$$

$\tan\phi_n = \tan\phi\cos\psi$

The required condition to satisfy the dynamic check is $F_0, F_w > F_d$.

If not so, keeping on calculating by increasing the module until it is satisfied need to be done.

Table 2: Design result data for helical gear.

	Symbol	Pinion	Gear	Unit
No. of teeth	n	11	34	-
Pitch circle diameter	D	27.5	85	mm
Outside diameter	D_0	32	89	mm
Root diameter	D_R	21	78	mm
Face width	b	21	21	mm
module	m	2.5	2.5	mm
Speed	N	6500	5000	rpm

The design result data for helical gear is shown in Table 2.

Contact stresses are analysed by increasing face width which are obtained from the result data.

III. AGMA EQUATION USED TO CALCULATED CONTACT STRESSES

3.1. Contact Stress Calculation by Using AGMA Equation

$$\sigma_c = C_p \sqrt{\frac{F_t\left(\frac{\cos\psi}{0.95CR}\right)K_v K_0 (0.93K_m)}{bdI}} \qquad (12)$$

Where,

$$I = \frac{\sin\phi\cos\phi}{2} \times \frac{i}{i+1}$$

$$C_p = \sqrt{\frac{1}{\pi\left(\frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}\right)}}$$

$$CR = \left(\frac{\sqrt{(r_1+a)^2 - r_{b1}^2} + \sqrt{(r_2+a)^2 - r_{b2}^2} + (r_1+r_2)\sin\phi}{\pi m\cos\phi}\right)$$

Table 3: Theory result for contact stress by using AGMA equation.

Face Width (mm)	Contact stress (MPa)
21	493.27
22	481.93
23	471.34
24	461.41
25	452.09
26	443.31
27	435.02
28	427.18

The theory results for contact stress by using AGMA equation is shown in Table 3.

3.2. Calculation of von-Mises Stress

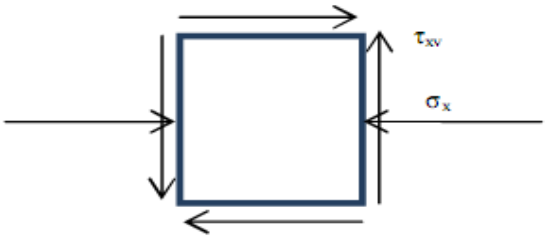


Figure 2. Stresses in the x-y plane

To get shear stress τ_{xy},

$$\tau_{xy} = \frac{T \times R}{J} \qquad (13)$$

$$= \frac{37.054}{\left(\frac{\pi}{32}\right) \times 27.5^4} \times \frac{27.5}{2} = 0.0091 \text{MPa}$$

$$\sigma_{1,2} = \frac{1}{2}(\sigma_x + \sigma_y) \pm \frac{1}{2}\left[(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2\right]^{\frac{1}{2}} \qquad (14)$$

By using von-Mises Criterion equation,

$$\bar{\sigma} = \frac{1}{\sqrt{2}}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]^{\frac{1}{2}} \qquad (15)$$

Table 4: Theoretical result of von-Mises stresses.

Face Width (mm)	von-Mises stress for AGMA (MPa)
21	493.27
22	481.93
23	471.34
24	461.41
25	452.09
26	443.31
27	435.02
28	427.18

The theoretical results of von-Mises stresses is shown in Table 4.

IV. MODELLING OF GEAR

This gears are modelled by the results obtained from theory in SolidWorks software. The material for the gear is AISI 5160 OQT400. The gears are modelled with the combination of the all above mentioned parameter, other gear set are modelled in same way. The helical gear was generated by using SolidWorks software.

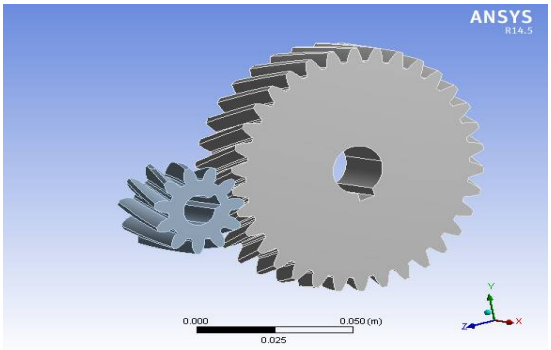


Fig. 3 Solid model of helical gear pair

The solid model of helical gear pair is shown in Figure 3.

4.1. FEM Analysis

The structural analysis of helical gear tooth model is carried out in SolidWorks and imported in ANSYS software. The fixed support is given at the inner rim of the pinion and torque which the value is 37.054Nm is applied to the helical pinion in clockwise direction. The frictionless support is given at the inner rim of the pinion.

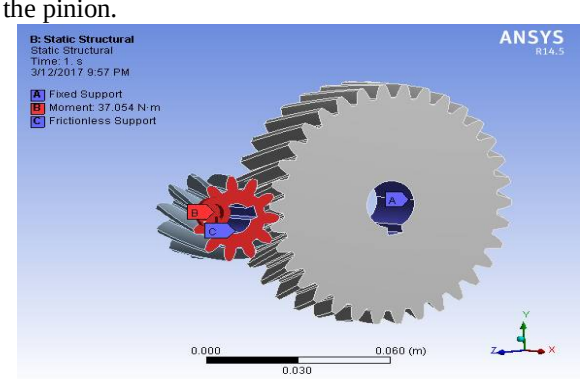


Figure 4 Boundary condition of helical gear pair

The boundary condition and fixed support of helical gear pair is shown in Figure 4.

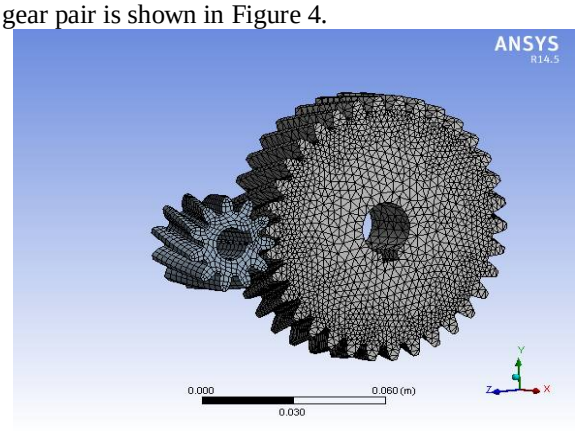


Figure 5 Meshed gear model

The generated mesh of helical gear pair is shown in Figure 5. The generated mesh is done by fine position to obtain the good quality of mesh.

V. FEM EQUATION USED TO CALCULATED CONTACT STRESSES

The model was analysed by using ANSYS software. The different value of face width are used to find out the contact stress.

5.1 Contact Stress Calculation by using ANSYS

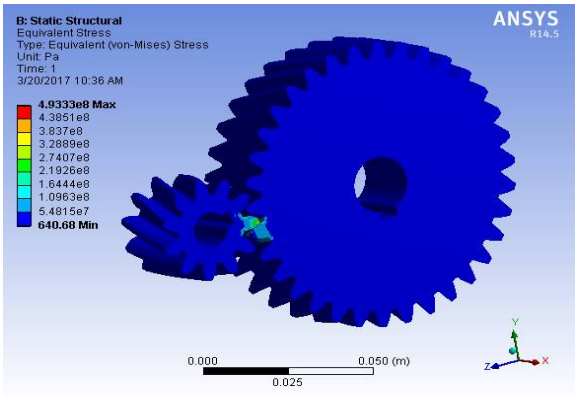


Figure 6 von-Mises stress of 21 mm face width modelled gear

The von-Mises stress of 21 mm face width modelled gear is shown in Figure 6. The magnitude of maximum stress is 493.33MPa which occur at the contact point.

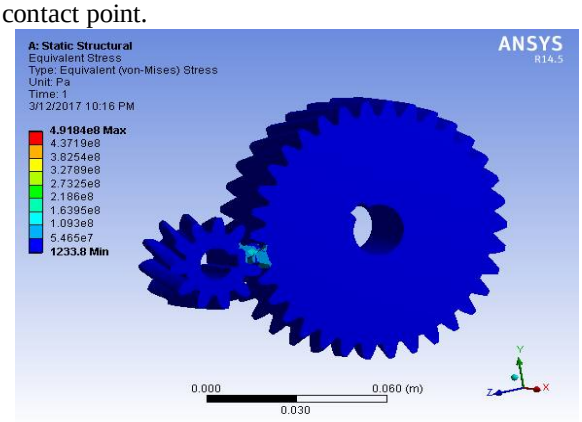


Figure 7 von-Mises stress of 22 mm face width modelled gear

The von-Mises stress of 22 mm face width modelled gear is shown in Figure 7. The magnitude of maximum stress is 491.84MPa which occur at the contact point.

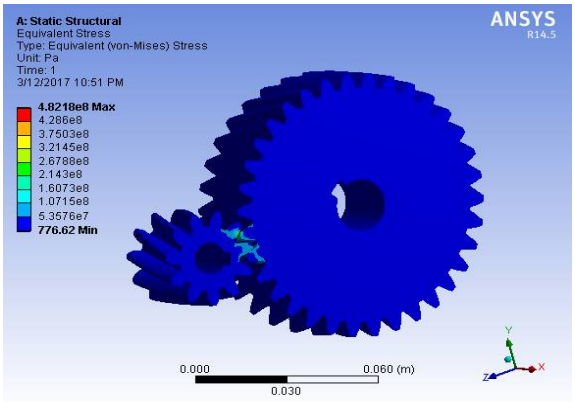


Figure 8 von-Mises stress of 23 mm face width modelled gear

The von-Mises stress of 23 mm face width modelled gear is shown in Figure 8. The magnitude of maximum stress is 482.18MPa which occur at the contact point.

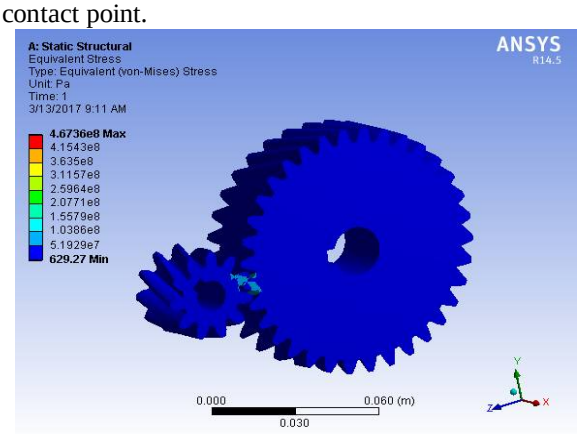


Figure 9 von-Mises Stress of 24 mm face width modelled gear

The von-Mises stress of 24 mm face width modelled gear is shown in Figure 9. The magnitude of maximum stress is 467.36MPa which occur at the contact point.

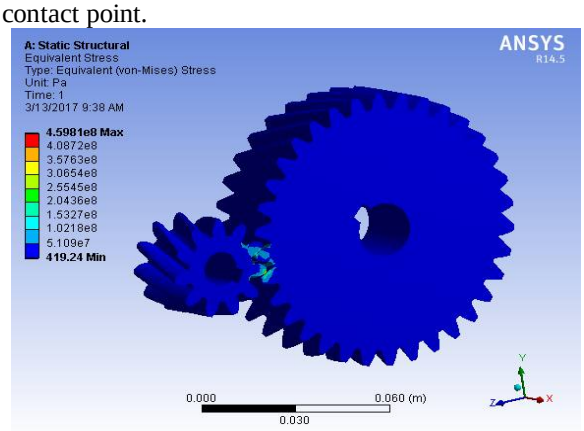


Figure 10 von-Mises stress of 25 mm face width modelled gear

The von-Mises stress of 25 mm face width modelled gear is shown in Figure 10. The magnitude of

maximum stress is 459.81MPa which occur at the contact point.

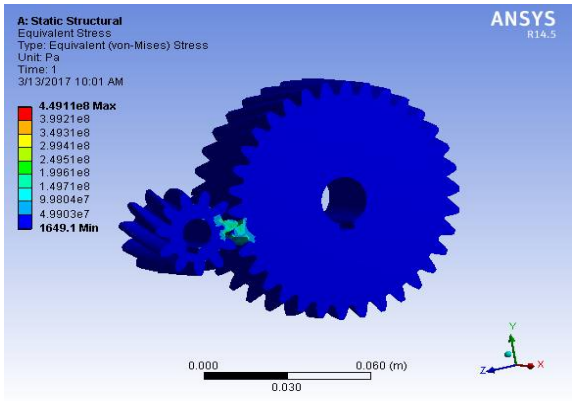


Figure 11 von-Mises stress of 26 mm face width modelled gear

The von-Mises stress of 26 mm face width modelled gear is shown in Figure 11. The magnitude of maximum stress is 449.11MPa which occur at the contact point.

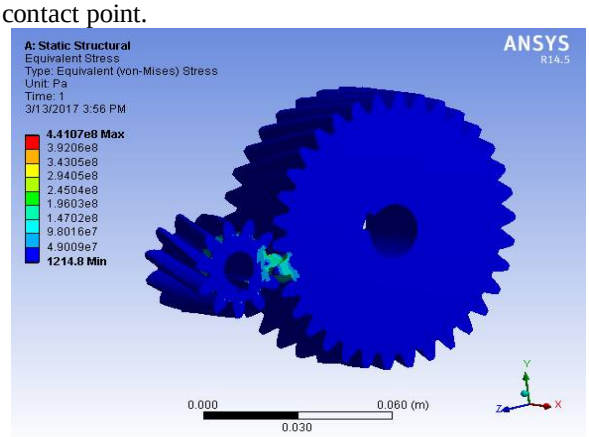


Figure 12 von-Mises stress of 27 mm face width modelled gear

The von-Mises stress of 27 mm face width modelled gear is shown in Figure 12. The magnitude of maximum stress is 441.07MPa which occur at the contact point.

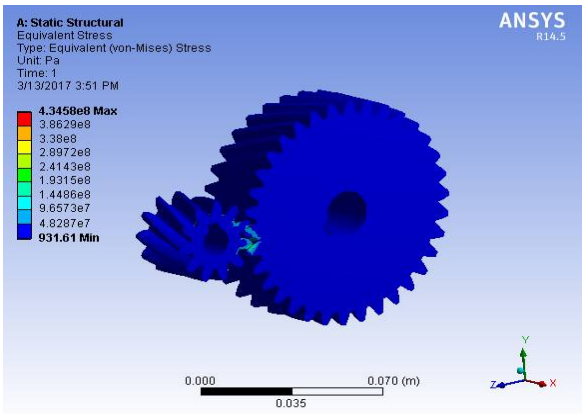


Figure 13 von-Mises stress of 28 mm face width modelled gear

The von-Mises stress of 28 mm face width modelled gear is shown in Figure 13. The magnitude of maximum stress is 434.58MPa which occur at the root of the teeth which contact between pinion and gear.

VI. RESULTS AND DISCUSSIONS

Table 5: Comparison of AGMA Theory and FEM

Face Width (mm)	Theoretical Stress (MPa)	Simulation Results (MPa)	Allowable Stress (MPa)
21	493.27	493.33	493.46
22	481.93	491.84	
23	471.34	482.18	
24	461.41	467.36	
25	452.09	459.81	
26	443.31	449.11	
27	435.02	441.07	
28	427.18	434.58	

The result data for theory and FEM is shown Table 5. The above table shows that the von-Mises stresses is inversely proportional to the face width of the helical gear, as the face width increases the contact stresses decreases. The effect of face width on contact stress is studied by varying the face width. It can be observed that the variation in the magnitude of stress to face width. When comparing, AGMA values are a little lower than the ANSYS values. Error percentages are around 3 %.

Table 6: Weight of helical gear pairs

Face Width (mm)	Weight of helical gear pairs (N)
21	10.2152
22	10.7615
23	11.2392
24	11.7279
25	12.2165
26	12.7052
27	13.1939
28	13.6825

The weight of helical gear pairs is shown in Table 6. Weight reduction is very important criterion in automobile. Therefore, face width 21 mm is selected for reducing the weight of car.

CONCLUSION

Face width is an important geometrical parameters in determining the state of stresses during the design of gears. The effect of face width on von-Mises stress is studied by varying the face width which are from 21mm to 28mm calculated from theory. Both AGMA theory of contact stresses as well as FEM method of contact stresses gives almost identical results with a mere percentage difference. The results are indicated that as the face width increases contact stress decreases. By observing the analysis results, the stress values obtained are less than the allowable

stress (493.46 MPa). So, it can conclude that the design is safe under working condition. And then, weight reduction is a very important criterion in the helical gears. So, face width 21mm is preferred.

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