Effect of Web & Flange Thickness on Nonmetallic Gear Performance

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Nomenclature

b	face width (mm)	d_p	pitch diameter (mm)
F	force (N)	K_B	rim thickness factor
t	rim thickness (mm)	T	torque (Nm)

Introduction

Gears are manufactured with thin rims for several reasons. Steel gears are manufactured with thin rims and webs where low weight is important. Nonmetallic gears, manufactured by injection molding, are designed with thin rims as part of the general design rule to maintain uniform thickness to ensure even post-mold cooling. When a thin-rimmed gear fails, the fracture is through the root of the gear, as shown in Fig. 1a, rather than the usual fillet failure shown in Fig. 1b.

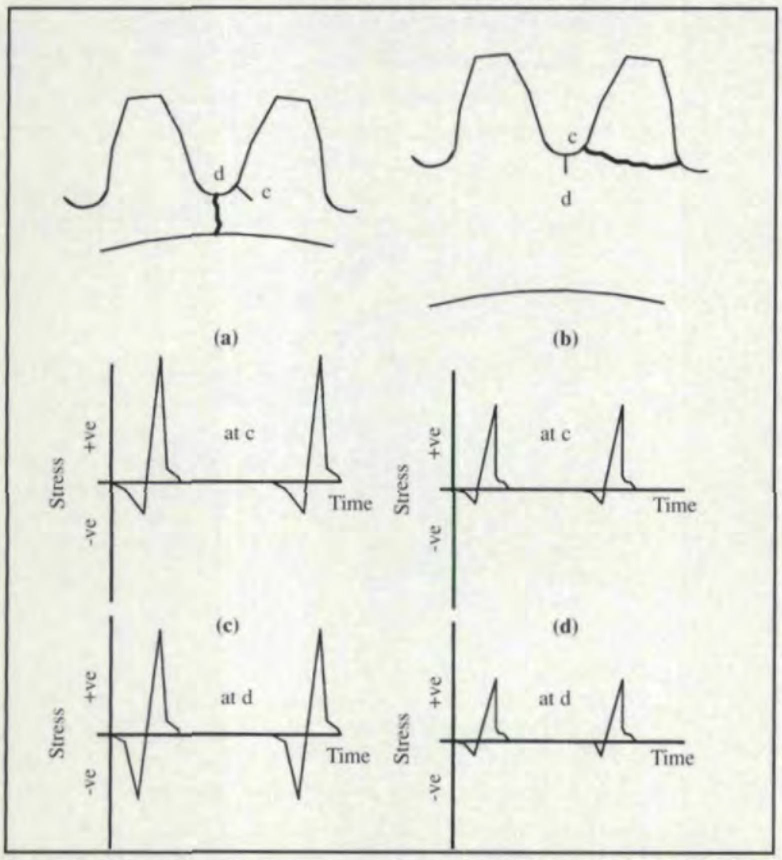

Fig. 1 — Failure due to bending stress for (a) thin-rimmed gears and (b) thick-rimmed gears. Tooth bending stresses for (c) thin-rimmed gears and (d) thick-rimmed gears.

The failure of thin-rimmed gears through root fracture was first explored by Drago (Ref. 1). He showed that for thin-rimmed gears, the bending stress varied from a larger compressive stress to an even larger maximum tensile stress (see Fig. 1c). For thick-rimmed gears, the bending stress varied between a small compressive and a large tensile stress (Fig. 1d). Comparing the root and fillet of thin-rimmed gears, it is possible to see that the higher compressive stress is found at the root of the gear; thus, the alternating stress at the root of the gear tooth is higher than at the fillet, making the root of the gear the critical section for fatigue failure. The compressive part of the stress for thin-rimmed gears is the result of compressive deflection caused by the preceding tooth pair. Drago compared his experimental result with finite element analysis, where good agreement was found. The finite element analysis involved calculating both the compressive and tensile stresses for different points from the fillet to the root to determine the maximum alternating stress and its position. Drago's later work on rim factors, which accounts for the difference in stress level between thin- and thick-rimmed gears, is now incorporated in AGMA 2001 (Ref. 2). Gulliot and Tordion (Ref. 3) have analyzed the bending stress of thin-rimmed gears with keyways using finite element methods. They plotted a nondimensional stress number, σ (defined in Eq. 1 as a function of maximum root stress, pitch diameter, normal force and face width), as a function of the ratio of the rim thickness to the pitch diameter, t/d_p.

$$\sigma = \frac{\sigma d_p b}{F} \tag{1}$$

However, this shows only maximum stresses as a function of rim thickness, not the alternating stresses. Because of this, its application is limited, since it does not address the main failure mode of thin-rimmed gears, which is fatigue.

These researchers used two-dimensional finite element methods. For nonmetallic gears molded with a web and flange, the problem is three-dimensional. Considering this, Davoli *et al* (Ref. 4) tried to determine the optimum web and flange thickness based on deflection analysis. Their conclusion suggested a minimum rim thickness of about 3 times the module. Their work, however, did not consider load sharing, the running of polymer gears against steel, nor the distribution of stresses or deflection across the gear face width, and experimental work in this field appears to be nonexistent.

The objective of this work was to make a detailed assessment of the effect of web and flange thicknesses on the performance of nonmetallic gears. The analyses considered the running of polymer gears against each other and against steel. The work was achieved by the use of three-dimensional finite element analysis. Experimental work was also done to supplement and compare with computer simulations.

Finite Element Model for Nonmetallic Gears

Nonmetallic gears made by injection molding cannot have a solid body, thick sections or large differences of thickness between contiguous parts because of differential cooling and shrinking after molding. Thick sections also result in longer cycle times when molding. The maximum practical wall thickness in plastic moldings is about 12 mm. For these reasons, nonmetallic gears typically have the features shown in Fig. 2. The important parts are the teeth, the rim or flange, the web, the hub and often a metal insert. The function of the metal insert is mainly to avoid failure at the hub under load. The rim is tapered by about 2 degrees to facilitate ejection after molding. On the other hand, cut nonmetallic gears have solid bodies.

For all practical purposes, in modeling nonmetallic gears, the effect of the metal insert can be neglected, as its influence on tooth bending stress and deflection is negligibly small. Because of symmetry, it is only necessary to model half the gear (see Fig. 3a). The driver and driven gears were modeled together, making contact at the pitch point (Fig. 3b). This is because, for nonmetallic gears, the pitch point is the contact position where a single tooth pair is most likely to carry the full load and probably results in the maximum root stress (Ref. 5). To minimize computing time while retaining a reasonable degree of accuracy, the models shown in Fig. 4 were compared.

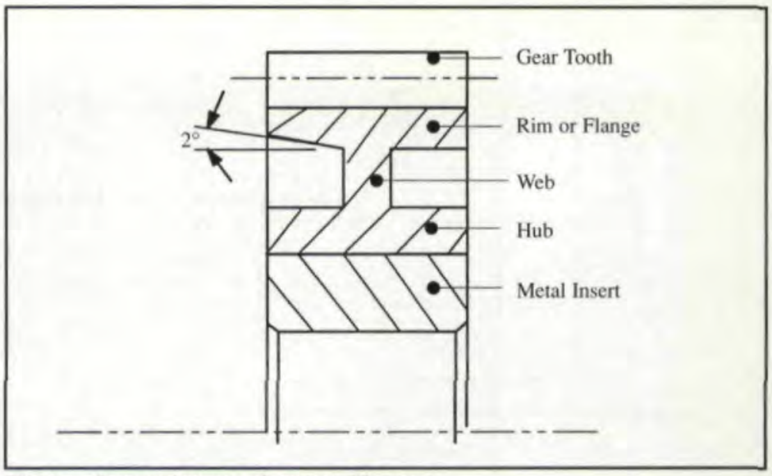

Fig. 2 — General features of nonmetallic gears.

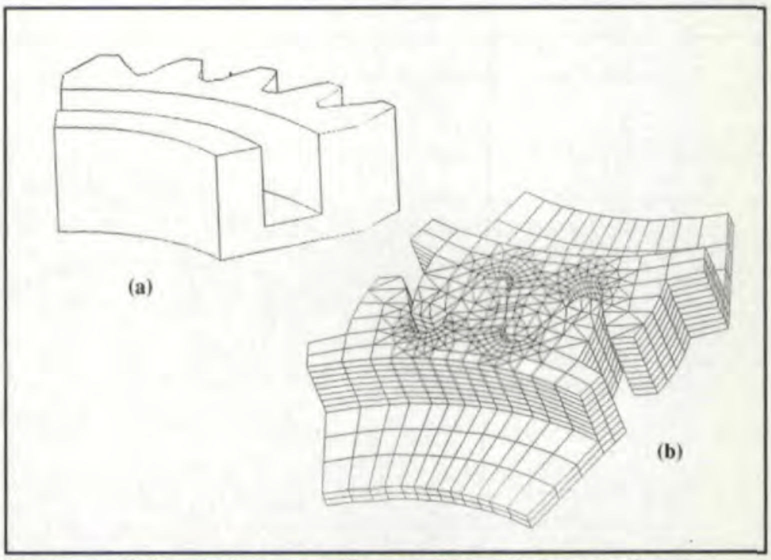

Fig. 3 — (a) Half model for nonmetallic gears; (b) model of gear tooth contact.

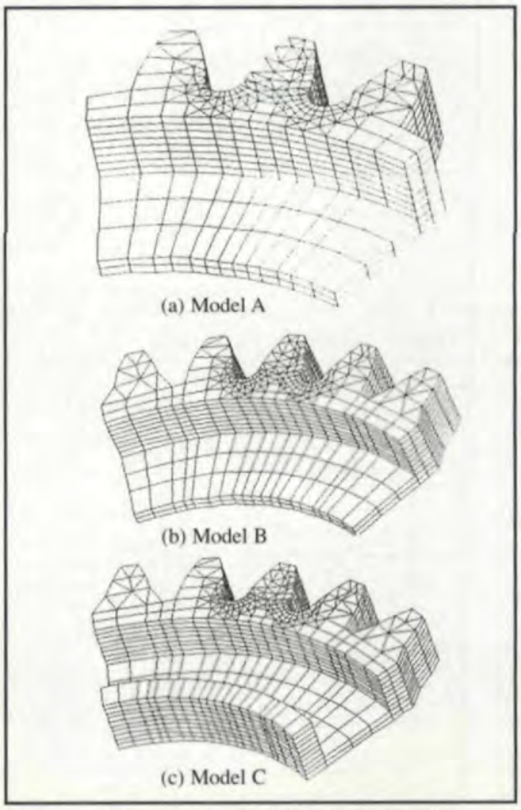

Fig. 4 — Various models of finite element mesh.

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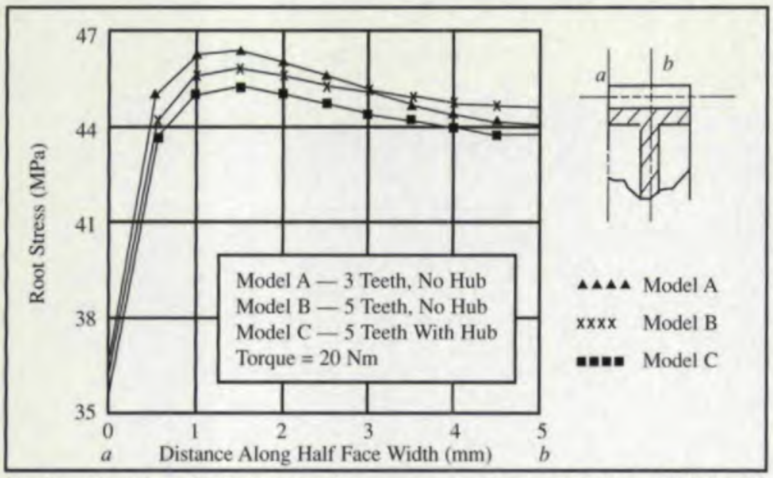


Fig. 5 — Variation of principal bending stress across the gear half face width for different finite element models.

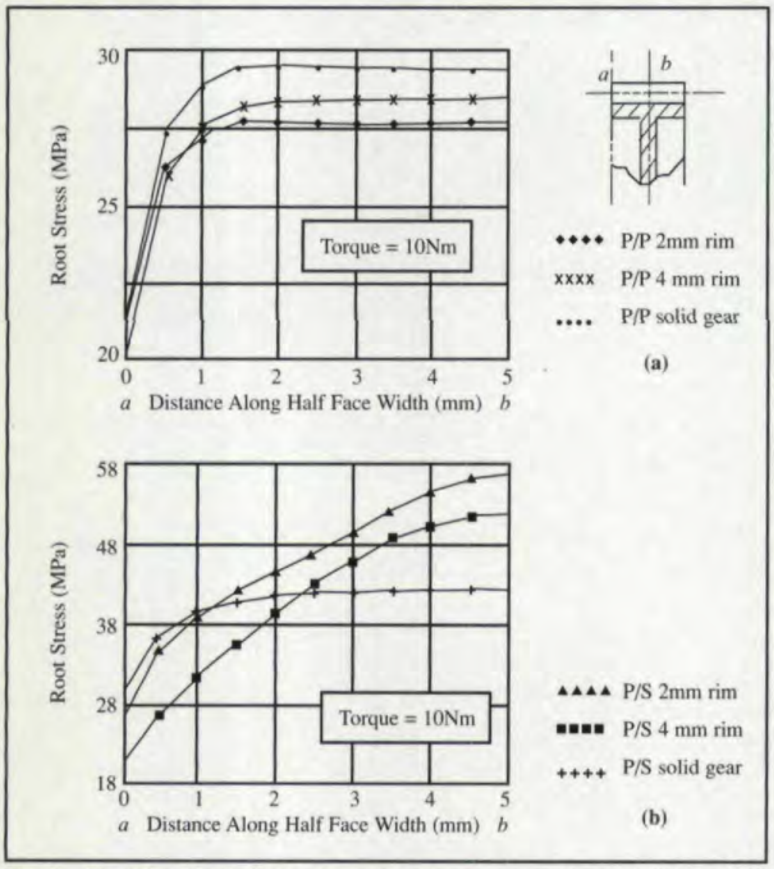


Fig. 6 — Variation of principal root bending stresses across the gear half face width for (a) polymer/polymer gear pairs and (b) polymer/steel gear pairs.

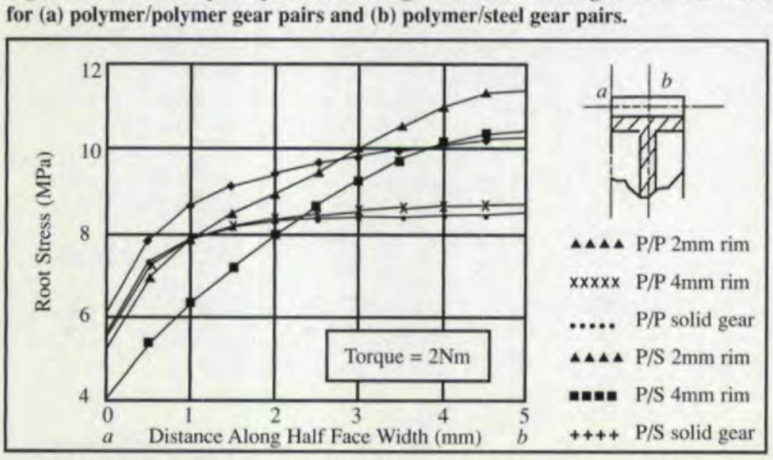


Fig. 7 — Variation of principal root bending stresses across the gear half face width for 2 Nm.

Linear brick and wedge elements were used for the gear mesh. The wedge elements were used only to smooth the transition from fine to coarse mesh. Model A was made up of 3 teeth with no hub. The web of the gear was assumed to end on top of the hub. Model B was the same as Model A, except that it was made up of 5 teeth. Model C consists of all the features of nonmetallic gears, except the metal insert, and was also made up of 5 teeth. All these models were loaded to the same torque. The stress distributions for all three models are shown in Fig. 5. As seen in this diagram, the difference in stress levels between the three models was small. For computing efficiency Model A was chosen and all analyses in this report were based on this model.

All the analyses in this work are for a module 2 mm gear with 30 teeth and a 20° pressure angle, with standard gear tooth proportions to BS 436:1970 (Ref. 6). A relatively low face width of 10 mm was used to reduce the computing time involved. Web and flange thicknesses were made variable.

Stress Distributions for Polymer/Polymer and Polymer/Steel Gear Pairs

The effect of web and flange thicknesses on the performance of polymer gears was investigated by plotting stress distributions across the gear face width (see Fig. 6), showing both polymer gears running against each other (P/P) and with steel (P/S). An elastic modulus of 207 and 3 GPa was taken for steel and polymers, respectively. A modulus of 3 GPa is representative of acetal and nylon. In each case, the thicknesses of the rim and the web were made equal, and only the values of the rim thickness are shown in the figures. The applied torque of 10 Nm represents a typical value when running polymer gears against steel for the dimensions of the gears described above.

Fig. 6a shows that the stress distribution across the face width is nearly the same for all values of web and flange thicknesses investigated. The slight increase in maximum stress with increasing web and flange thicknesses is caused by the load sharing effect; i.e, these sections result in larger tooth deformations, leading to increased load sharing. However, the increase in load sharing is caused by edge contact. The plot for polymer gears running against steel (Fig. 6b) shows a distinct difference between solid and webbed gears. Thin web and flange thicknesses result in an increased maximum stress and an uneven stress distribution across the face width. A solid—that is, rimless gear gives the minimum stress levels and the most uniform stress distribution.

In general, the main difference between the P/P and P/S gear pairs is that the latter increase stress and distribute it more widely across the face width for the same applied load because of the difference in flexibility between polymer and steel gears. Polymer gears deflect about 100 times more than steel gears under the same load. Making polymer gears with web and flange features also creates a difference in flexibility across the face width, where the middle part is less flexible than the outer part because of the stiffening effect of the web.

Influence of Applied Load on the Effect of Web and Flange Thicknesses

The effects of web and flange thicknesses are load dependent. Looking at Fig. 7 for P/P gear combinations, the maximum stress levels increase with decreasing rim thickness. For 4 mm web and flange thicknesses, the gear is sufficiently thick to give almost the same stress level as that of the solid, nonrimmed gear. This is in contrast to the results for a torque of 10 Nm shown in Fig. 6. For 2 Nm the load is not sufficiently high to produce any appreciable load sharing effect. This figure also shows that running polymer gears against steel produces an uneven stress distribution across the face width.

Influence of Face Width on the Effect of Web and Flange Thicknesses

Gears with narrow face widths can experience buckling and misalignment, while too large a face width may be subjected to non-uniform load distributions because of twisting. This effect is shown in Fig. 8. A torque of 1 Nm/mm face width was applied for 10 mm and 17 mm face width gears. In both cases, the web and flange thicknesses of the polymer gears were kept to 4 mm. Both of the models were meshed with a solid nonrimmed steel gear. The plot for the 17 mm face width polymer gear shows a high maximum root stress and a wider variation in stress distribution across the gear face width compared to the narrower gear. This figure indicates that there is a dependence between web and flange thicknesses and face width. Wide-faced polymer gears should have a thicker rim (or reinforcing axial webs) than narrow face width polymer gears for the same load per unit face width.

Effect of Web Thickness on Stress Distribution

An analysis was made by varying the thickness of the web for a fixed flange thickness and face width; the results are shown in Fig. 9. The stress distribution for the 2 mm flange and 4 mm web thicknesses is between the 2 and 4 mm web and flange thicknesses. The implication of this result is that changes in web and flange thicknesses have much the same effect on stress distributions.

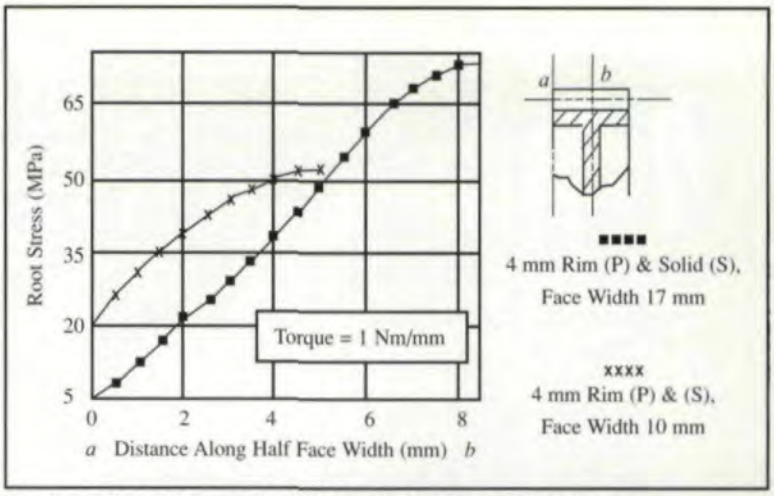


Fig. 8 — Effect of face width relative to rim thickness on bending stress.

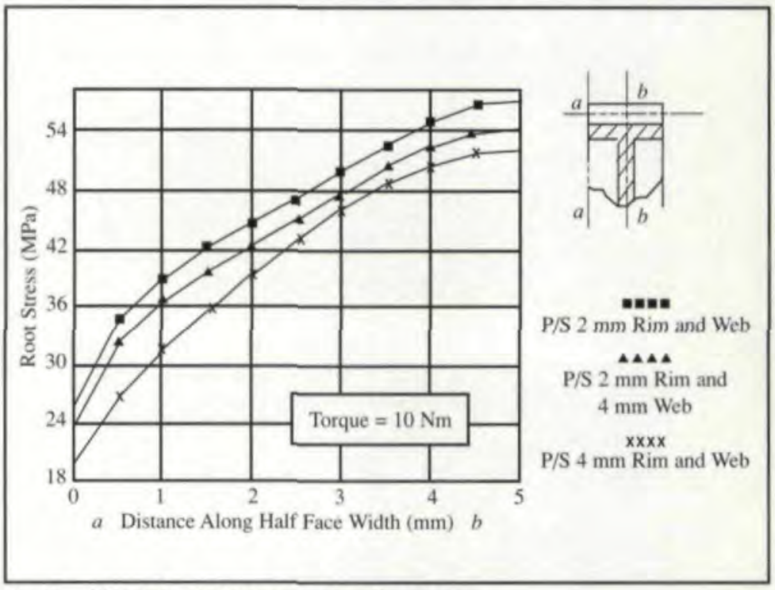


Fig. 9 — Effect of web thickness on bending stress.

Modes of Rim Failures in Nonmetallic Gears

Fatigue tests were carried out on 2 and 4 mm web and flange nylon gears running against steel. The modes of rim failures observed are shown in Fig. 10. Fig. 10a shows a typical failure mode for solid gears. Fig. 10b shows the typical failure mode of thin-rimmed gears with a fatigue crack running from the tooth fillet through to the rim of the gear, as discussed in Section 1. Fig. 10c shows a similar rim failure as in Fig. 10b, but where the fracture has extended from the tooth root to an area of high stress between the flange and the web.

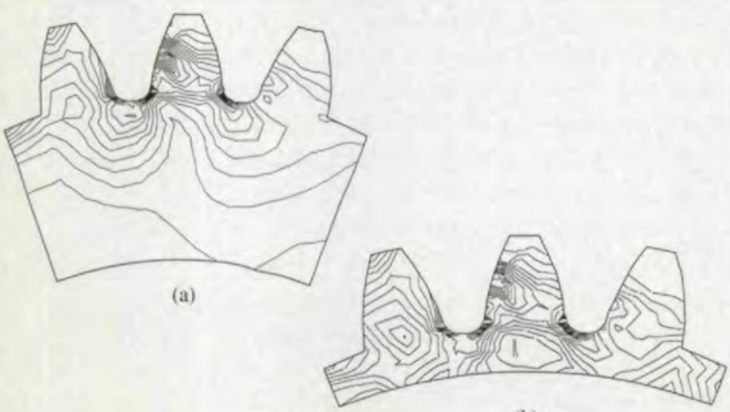
As previously mentioned, Drago explained the failure at the root of thin-rimmed gears rather than at the fillet. On the other hand, experiments have shown that cracks can initiate at the fillet and propagate through the rim of the gear (Fig. 10b). To find an explanation for the failure of thin-rimmed gears in this way, analysis of the stress distribution pattern for thin- and thick-rimmed

(a)

(b)

(c)

Fig. 10 — Modes of rim failure for nonmetallic gears for (a) a solid gear and (b–c) thin-rimmed gears.



(a)

(b)

Von Mises Stress Distribution for (a) solid rim and (b) thin rim gears.

Fig. 11 — Stress distribution for (a) thick-rimmed gears and (b) thin-rimmed gears.

gears was made using the finite element method. The results are shown in Figs. 11a and b, where for solid gears, the stress intensity in the body of the gear is considerably less than at the tooth fillet, and hence fatigue cracks might not progress to the bulk of the gear. However, for thin-rimmed gears, since the stress intensity in the rim of the gear is much the same as that at the tooth fillet, there is a greater possibility for the crack to propagate across the rim (Fig. 11b).

Remedies for the Effect of Web & Flange Thicknesses

It has been shown that nonmetallic gears with thin webs and flanges result in a difference of flexibility across the gear face width. The middle part is less flexible than the outer face because of the stiffening effect of the web. The effect of this on the stress distribution across the gear face width for polymer/polymer gear combinations was found to be minimal. Web and flange thicknesses 2 to 3 times the module were found to perform the same as solid, nonrimmed, polymer/polymer gear pairs. This was confirmed by experimental tests carried out on acetal/acetal gear pairs by varying web and flange thicknesses (Ref. 7).

For polymer/steel gear pairs, thin web and flange thicknesses on the polymer gear have been shown to result in increased stress levels and a nonuniform stress distribution across the gear face width. From these results, we concluded that polymer gears intended to run against other polymer gears should be designed differently than when they are intended to be run against metal. The use of solid, nonrimmed polymer gears against steel is one solution. This solution, however, is only applicable for cut polymer gears. For molded nonmetallic gears, axially reinforcing webs can be used as an alternative. Running a flexible metallic gear against the polymer gear could be possible if materials of lower elastic modulus, such as aluminum and copper, were used. However, the use of these materials against polymers is not recommended according to BS 6168(8) because of wear-related problems. For wide-faced gears, it has been shown that metallic gears with very thin web and flange thicknesses might be flexible enough to get a near uniform stress distribution across the polymer gear face. Because of the limited number of experimental tests, the limit as to how much the steel gear can be thinned has not been investigated (Ref. 9). In general, running nonmetallic gears against thin-webbed and flanged steel gears has been shown to be advantageous. However, this potential improvement in the performance of polymer/steel gear combinations needs further investigation, as does the use of counter-crowned or double, low-helical-angle polymer gears (Ref. 10).

The effect of web and flange thicknesses on the stress distribution in polymer gears is dependent on load, face width, module and Young's modulus. For this reason, finding a simple analytical formula for the optimum web and flange thicknesses may not be possible. The optimization of a given design by using the finite element method would still appear to be the best option, although for many applications this level of analysis may not be justified. The general results of this research in recommending rim thickness are, however, useful for general applications.

Rim Factor

To account for the effect of rim thickness in calculating bending stresses for steel gears, Drago proposed a rim thickness factor which was included in AGMA 2001 (Ref. 2). The rim thickness factor proposed by Drago is based on measured stress levels at the root of a gear tooth under static conditions. This implies that the rim factor is somewhat similar to a stress concentration factor and accounts only for the geometry of the gear. More conveniently, a rim thickness factor, analogous to the strength reduction factor, which takes into account geometry, material and operating conditions, can be determined from fatigue tests using the following equation (see Fig. 12).

$$K_B(l,r_t) = \frac{T_s}{T_r}$$

Where K_B is the rim factor as a function of life, l, and rim thickness, r_t. T_s is the torque for the solid, nonrimmed gear, and T_r the torque for the rimmed gear; both for a life of l cycles.

In the equation above, T_s and T_r can be replaced by corresponding bending stresses calculated using the Lewis equation. Clearly a large number of tests would need to be carried out on a range of gear rim and web thicknesses and polymer materials in order to determine K_B values for inclusion in practical designs.

Conclusions

This study shows that making nonmetallic gears with thin webs and flanges results in a difference of flexibility across the face width and, consequently, an uneven stress distribution. This effect was most pronounced when running polymer gears against steel. From this we see that the design of polymer gears to run against steels should be different from their design when they are intended to run against other polymer gears. The influence of lead and face width relative to web and flange thicknesses has been investigated where the effect of web thickness was found to be the same as flange thickness.

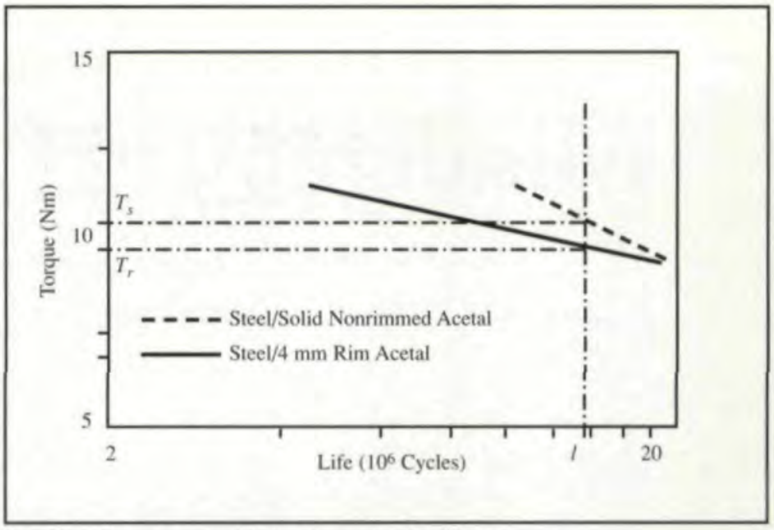


Fig. 12 — Torque versus life for rimmed and solid gears.

References:

1. Drago, R. J. "Recent Advances in the Evaluation of Stresses in Lightweight, High-Speed, Heavily Loaded Gearing." *International Conference on Reliability, Stress Analysis and Failure Prevention*, San Francisco, CA, 1980, pp. 225–235.
2. ANSI/AGMA 2001-B88, 1988. *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.*
3. Gulliot, M. & G. V. Tordion. "Stress Analysis of Thin-Rim Spur Gears by Finite Element Method." *Proceedings of the 1989 International Power Transmission and Gearing Conference*, Vol. 1, 1989, pp. 167–174.
4. Davoli, P. *et al.* "Deformations in Thin-Rimmed Gears." *Europower Transmission*, 3, 1990, pp. 64–68.
5. Walton, D. *et al.* "Load Sharing in Metallic and Nonmetallic Gears." *J. Eng. Sci., Proc. I Mech E, Part C*, 1994, pp. 81–87.
6. BS 436: Part 2: 1970. *British Standard Specification for Spur and Helical Gears.*
7. Tessema, A. A. & D. Walton. "Effect of Web and Flange Thicknesses on the Bending Stresses of Nonmetallic Gears." *KOLA ZEBATE KZ '93 Wytwarzanie, Pomiary, eksploatacja*, Poznan, 1993, pp. 214–220.
8. BS 6168: 1987 *British Standard Specification for Nonmetallic Gears.*
9. Tessema, A. A. "Effects of Flexibility in Nonmetallic Gears." PhD. thesis, The University of Birmingham, 1993.
10. Tsukamoto, N. & K. Terashima. "Development of Plastic Gears for Power Transmission." *Bulletin of the JSME*, 29/247, 1986, pp. 246–255.

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