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Bending Load on Internal Gears of Planetary Gear Sets

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TECHNICAL PAPER

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

As compared to external toothings, the design fundamentals and standards for proving the load-carrying capacity of internal toothings have been little developed. Mostly only insufficiently considered is the influence of stress in the toothed gear rim, which is essential in almost all cases. Also for the theoretical borderline case of the infinitely thick toothed gear rim some design data are too inaccurate, since they are not based on the actual tooth fillet and on precise notch-stress calculations. Even larger are the deviations in elastically designed toothed gear rims.

This paper is to present results of investigations for the more precise determination of stress in the tooth root or the toothed gear rim of internal toothings specifically for their use in planetary gearings. Proposals for the practical calculation are included. The results of calculation are confirmed by tests. At first, the considerations concern borderline cases of load in an infinitely thick toothed gear rim, whose mounting in the casing has no influence on the stresses examined here (very elastic connecting elements).

On this basis, the case of the inner toothed gear rim with a cylindrical connection contour is treated (the internal toothing is positioned only on a part of the hollow cylinder). The influence of the connection contours on the stress as compared with the version with a ground contour was successfully taken into account by approximation relations. The assumptions and results of calculation were well determined by measurements done on a planetary gearing.

The stress calculations are carried out by a locus-dependent superposition of the stress components resulting from the radial force and the tangential force. Proceeding from the moment curve and longitudinal-force curve determined in the tooth gear rim, the local stress in the tooth gullet is determined with the aid of the boundary element method. The practical application of the findings is facilitated by generalized stress concentration factors, which are newly calculated for internal toothings.

To sum up, it can be stated that the contribution gives an insight into general connections and proposes a refined method for practical use.

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BENDING LOAD ON INTERNAL GEARS OF PLANETARY GEAR SETS

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1 INTRODUCTION¹⁾

In gear units with internal gears the geometrical main dimensions can be calculated according to the same relations as for external gears. The number of teeth of the internal gears and the centre distance have to be used with negative sign [1], [2].

Compared with gear units with external gears, however, there are peculiarities. These are first the kinds or possibilities of interferences. Some do not occur in gear units with external gears. Others are possible only in extreme cases. Instead of the tooth tip thickness, as in external gears, in internal gears the space width at the tooth root is one of the limit criteria.

In gear units with internal gears the flank contact pressure is lower than in gears with external gears of the same absolute length of diameters and the same loading. This is due to the concavo-convex pairing of the tooth profiles as against the otherwise convexo-convex pairing. It often seems that the tooth root load in internal gears is generally lower than in external gears. This impression is caused by the greater tooth root thickness on internal gears.

It is overlooked here that at fatigue stressing (continuous loading) also the tooth root notches have an essential influence. The root fillets of the internal gears are usually smaller than the root fillets of the external gears of the same module and with the same absolute number of teeth. Moreover, in the toothed rim of internal gears there is a considerable stress. Its decisive share is the bending stress. The stress in the toothed rim may lead to a considerable reduction of load capacity. This contribution deals with this problem of the tooth root load or toothed rim load. Continuous loading is assumed, i. e. a number of stress cycles $N > 10^6$. First the method of the general proof is briefly presented.

1) Symbols at end of paper

This is followed by a concrete calculation for the possible limiting cases. They constitute the highest and the lowest load capacity. The maximum value of load capacity results if the stress in the toothed rim is neglected. This is tantamount to assuming a very great toothed rim thickness. A minimum value of load capacity is obtained if it is assumed that the deformation of the toothed rim with internal gears to be calculated is not hindered by the mounting of the toothed rim in the casing or on other components. The toothed rim is thus assumed to be a free ring (figure 1). The torque is applied or absorbed by an evenly distributed loading on the periphery.

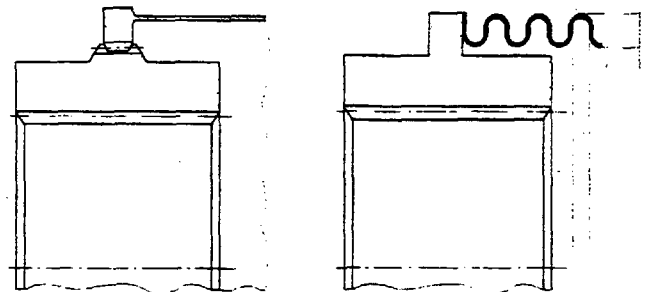


Figure 1 Elastically designed toothed rims; negligible influence of the support

The calculation of these limiting cases is followed by statements concerning the approximated consideration of the influence of the mounting of the toothed rim with adjacent cylindrical shells (figure 2). Finally, experimental investigations are referred to.

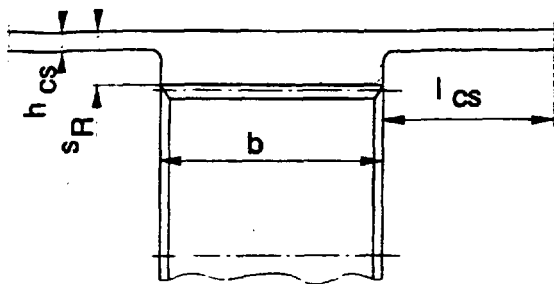


Figure 2 Elastic toothed rim with cylindrical adjacent contour; supporting effect

The objective of this paper is to point out influences and dependencies and to present an efficient approximation method for the calculation using PCs.

2 GENERAL METHOD FOR PROVING THE LOAD CAPACITY

Due to the greater elasticity of internal gears, in general the stress distribution is considerably different from that for external gears (figure 3). As it is shown in principle in figure 3, there is an essential stress not only during the tooth contact. On internal gears the whole gear periphery or the whole cycle ($|\phi| = 120^\circ$ with 3 planets) has to be considered. Another special feature is that the radial component of the tooth force F_r cannot be neglected. It effects a considerable reduction of stresses at the location of the tooth which is designated as tensile side and a considerable increase of stress at the location of the tooth which designated as compression side. Moreover, outside the tooth contact range a considerable tensile stress occurs, which is determined by the radial force component to a decisive degree.

The ratio of the tensile stress occurring in the tooth bottom to the compressive stress is not constant. Therefore, the influence of there-spective occurring mean stress on the strength value has to be taken into account.

The stresses to be considered are shown in the figures 4 and 5 (in connection with figure 3). For the tensile side (t) of the tooth bottom the stress varies between the value occurring at the load (I) of the tooth contacting the mating tooth ($\sigma_I^{(0)}$) and the compressive stress (c) which is present during the loading of the neighboring tooth (II) ($\sigma_{II}^{(0)}$). In cases of thin toothed rims and large numbers of teeth the highest tensile stress often occurs in the tooth bottom outside the zone of contact (spot III, figure 4; cf. also figure 3). Then it is the decisive tensile stress. During loading, however, the same location receives a stress with opposite sign on the side of the compressive stress of the tooth. The difference between the stresses constitutes the range of stress variation which has to be considered. It is a peculiarity of internal gears that due to that the crack may start from a location in the tooth bottom which is usually designated as compression side (c) (cf. also [4]). However, it receives a crack-opening tensile stress outside the zone of contact.

In general, the proof of the load capacity for internal gears has to be furnished for (figures 4 and 5)

a) Tensile side (spot t)

Range of stress variation from the tensile stress ($\sigma_I^{(0)}$) of the loaded tooth and the compressive stress at loading of the neighboring tooth ($\sigma_{II}^{(0)}$)

b) Compression side (spot c)

Range of stress variation from the compressive stress ($\sigma_I^{(0)}$) of the loaded tooth and the maximum tensile stress (σ_{III}) occurring offset by a larger angle.

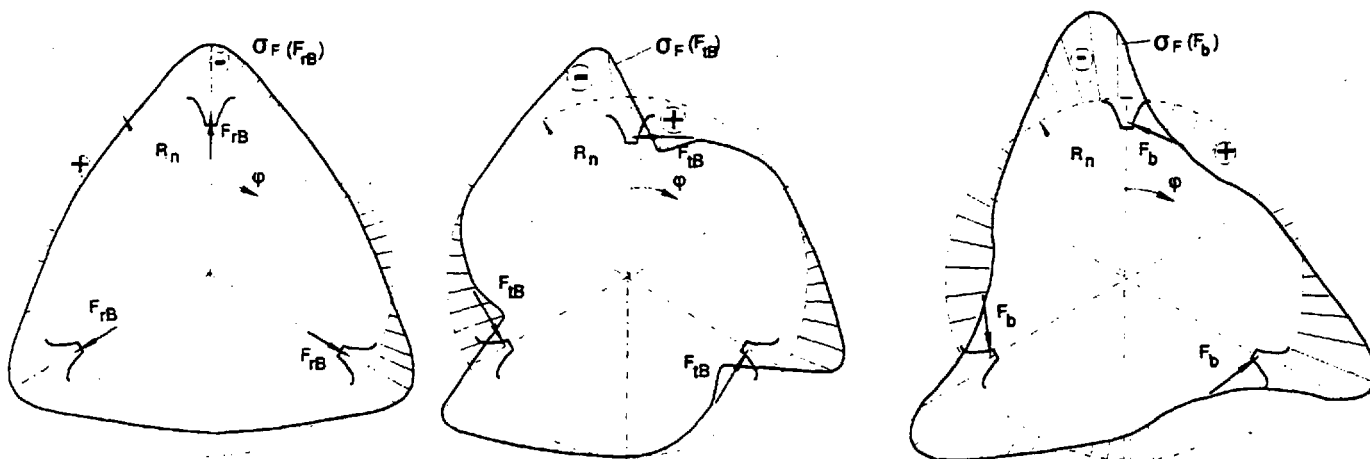


Figure 3 Schematic representation of the stress distributions on internal gears