High Power Transmission with Case-hardened Gears and Internal Power Branching

By
Dr. Ing. J. Theissen
A. Friedr. Flender GmbH u. Co KG
Bocholt, West Germany

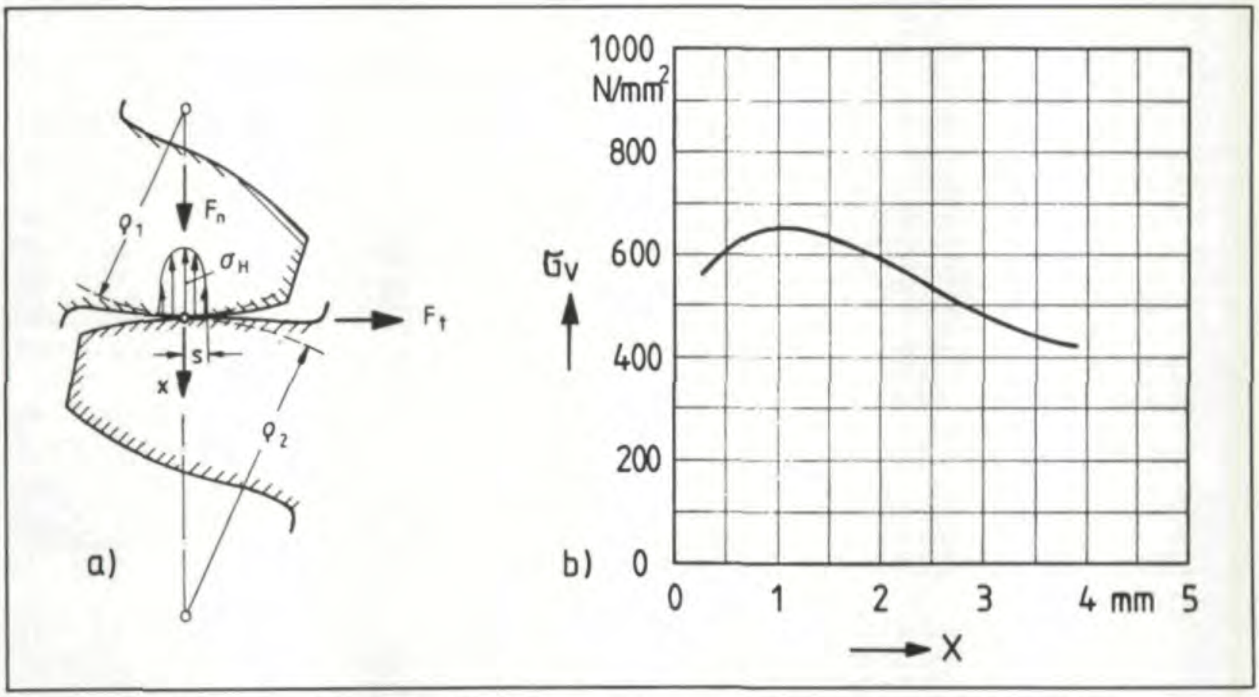

Fig. 1 a) Contact of two tooth flanks. $\rho 1$, $\rho 2$ = radii of curvature at the contact point of unloaded flanks;
Fn = tooth normal force; 2s = flattening width under load;
σH = maximum Hertzian contact stress at the tooth flank surface.
b) Dependence of equivalent stress σv on the depth x, module = 25 mm.

Introduction

In the field of large power transmission gear units for heavy machine industry, the following two development trends have been highly influential: use of case hardened gears and a branching of the power flow through two or more ways. The maximum possible torque of large gear units is limited by the machine capacity since the gear cutting machines can only manufacture gear wheels up to a maximum diameter. The highest possible output torque of gear units with case hardened gears and power branching is about ten times higher than conventional gear units with through hardened gears and without power branching.

The advantages of case hardened gears lie in their higher tooth flank load carrying capacity. The variation in the sub-surface strength matches ideally the sub-surface stress distribution. By optimising the gear geometry to balance the flank capacity to the root strength, the torque carrying capacity of case hardened gears can be four times higher than that of through hardened gears having the same diameter.

Power branching leads to a further increase in the torque carrying capacity. Such gear units have one input and one output shaft. Within the gear unit, the power at the gear of the input shaft is branched out and flows together at the out-

AUTHOR:

DR. ING. JOSEF THEISSEN *received his Bachelor's degree in engineering from the Technische Hochschule Aachen, and his Doctorate in engineering from the Ruhr-University Bochum. From 1977 through 1980, he was chief-engineer at the Institute for Machine-Elements and Gearing-Technics at Ruhr-University Bochum. Since 1981 he has been working as manager of the design department of planetary and high speed gear drives for A. Friedr.Flender GMBH and Co. Kg, a gear manufacturer in Bocholt, West Germany and in Elgin, USA/Flender Corporation.*

put shaft. To achieve equal power distribution in each branch special design features are required.

Firstly, the advantages of case hardened gears are shown. Then, the dependence of output torque on the gear unit size and weight is demonstrated. Also the efficiency of different power branching gear stages and simple gear stages are compared with one another. Lastly, the design of three large gear units, as given below, are presented:

- Rolling Mill gear unit with two way power distribution
- Tube Mill central gear unit with three and six way power distribution
- Planetary gear unit for ball mill drive with three way power distribution.

Gear Materials

The load capacity of gear teeth increases with a decrease in diametral pitch, i.e. decrease in number of teeth. The tooth flank load capacity is the decisive parameter for the dimensioning of a gear pair that has a minimum number of pinion teeth without undercutting.

The loading of a tooth flank along the common line of contact is calculated as the pressure load between two cylinders with contact lines having the same lengths and radii of curvature as the unloaded tooth flanks. The load on the tooth flanks is obtained approximately from the maximum Hertzian contact stress assuming that the material properties are the same.

The Hertzian contact stress alone, does not determine the load configuration. In the contact zone an elastohydrodynamic oil film pressure is developed, dependant on the rolling velocity, which varies from that of the Hertzian surface stress distribution. The tooth flanks slide and roll on one another and as a result a frictional force in the tangential direction is created. In addition, there exist residual stresses on and below the tooth flank surface. However, the maximum effective Hertzian contact stress has proved to be a useful theoretical criterium.

Stresses below the surface of tooth flanks, with due consideration to the above mentioned influences, can be computed. [1,2]. They can be treated as an equivalent stress according to maximum distortion energy theory. Fig. 1 shows the relation between calculated equivalent stress at the inner single contact point of pinion and the depth "x" from the tooth surface. The maximum equivalent stress $\sigma_{vmax} \approx 0.56\ \sigma_H$ lies approximately at a depth $x \approx 0.68$ s whereby σ_H is the maximum Hertzian contact stress and 2 s the flattening width.

A sub-surface fatigue strength distribution that has a form closely matching that of the stress distribution below the surface, is obtained with gears of case hardening steel of low carbon content. By carburizing the tooth surface and hardening, a hardened case of martensite is obtained while the core of the tooth remains soft and ductile. Correct heat treatment and a proper depth of hardness increases the root fatigue strength. Finally, grinding the tooth flanks gives a good tooth quality. There will be no strength reducing notch effect when grinding the tooth root, if the teeth of the gears are protuberance-hobbed before carburizing and hardening.[3]

Fig. 2 shows the Vickers hardness curve and the related fatigue strength σ_{sch} for repeated load cycles below the tooth surface of a gear manufactured from 17 CrNiMo 6 steel. Due to the hardened casing the equivalent stress at any depth "x" is less than the fatigue strength for repeated (non-alternating) loads i.e. $\sigma_v < \sigma_{sch}$.

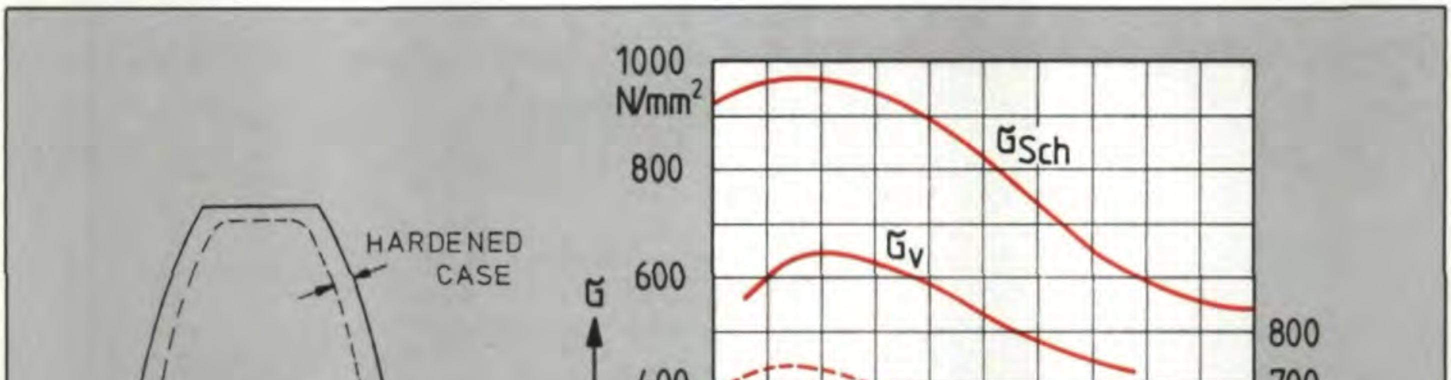

Fig. 2 a) Cross section through a tooth of case hardened gear, schematic
b) Dependence of Vickers-hardness HV (measured), σ_{sch} = endurance strength under repeated load and σ_v = equivalent stress at the depth x for a case hardened gear from 17 CrNiMo 6, module = 25 mm.

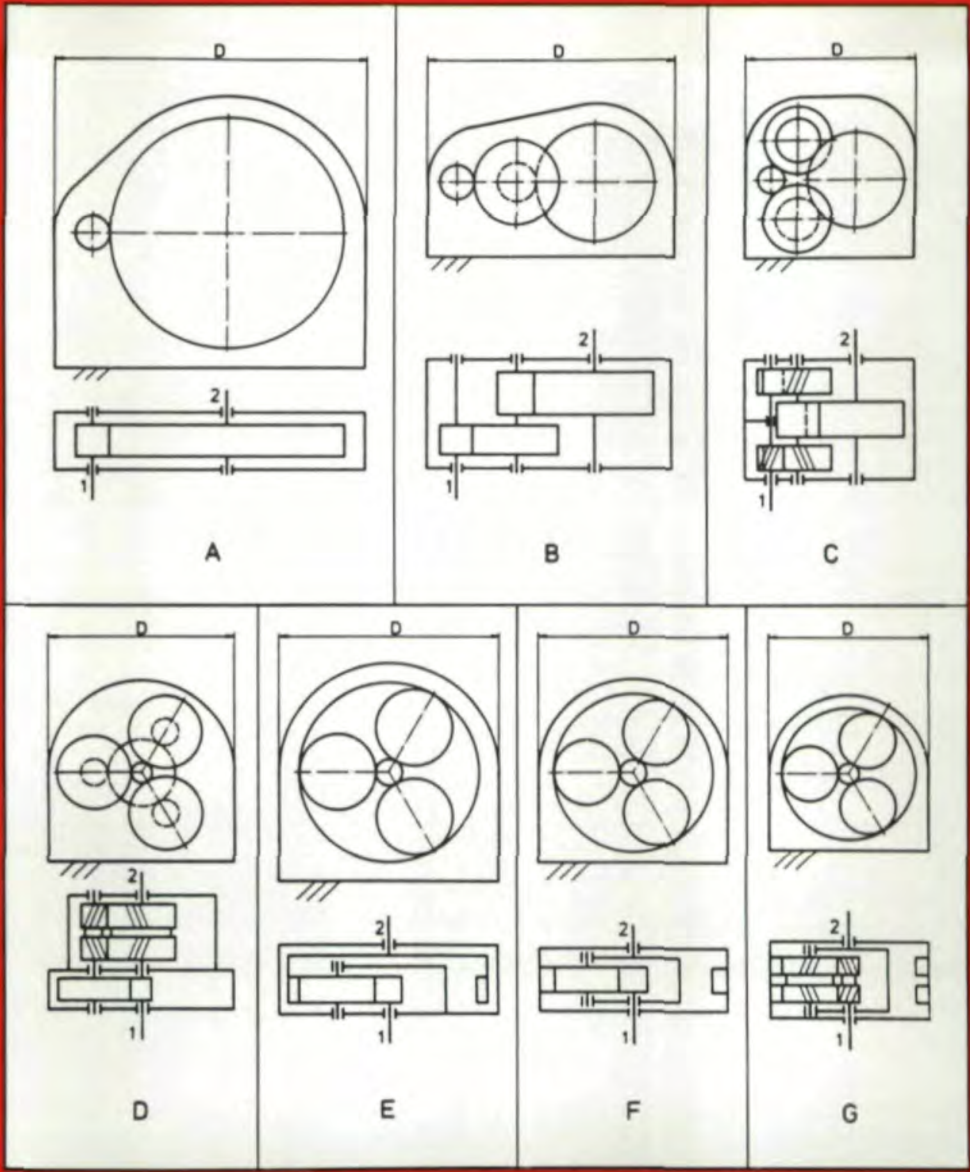

Fig. 3 – Schematic diagrams and size comparison of gear units with same shaft torques and transmission ratio $i = 7$ with and without power branching

Gear units with case hardened and ground gears from case-hardening steel have the following advantages over gear units with heat treated steel and manufactured by cutting tools:

- more compact in size and lighter in weight thus reducing manufacturing costs
- higher wear-resistance and a lower susceptibility to shock
- higher operating reliability through higher and more balanced capacity of the tooth root and the tooth flank
- lower rolling velocities and a lower tooth meshing frequency
- lower internal dynamic additional forces and a reduced noise level due to better tooth quality
- increased efficiency at part and full load operating conditions.

Case hardening and grinding of tooth flanks to increase the tooth flank capacity of gears has proved its usefulness, and has been successfully applied for several decades on small gears in Automobile Industries and for the last years on large gears for industrial gear units. Computation according to DIN 3990 or ISO/DP 3663, based on the experimental and theoretical investigations of is a safe method, pre-calculation of the capacity of case hardened gears.

Size, Weight and Efficiency

Fig. 3 shows schematically, gear units with and without power branching. The diameter ratio of the gears corresponds to an overall transmission ratio of 7. Shafts 1 and 2 are respectively the high and low speed shafts. The gear units A, B and C have parallel shafts and gear units D, E, F and G coaxial shafts.

Gear units A and B, respectively, are a single stage and a two stage unit. Neither has power branching.

Gear units C, D, E, F and G all have two stages and power branching. The gears on the intermediate shaft of the gear units C and D have different diameters, however, the intermediate gears on one intermediate shaft of E, F and G have been reduced to a single gear, hence, the latter are treated as single stage units.

Gear unit C has two way power branching. Equal power

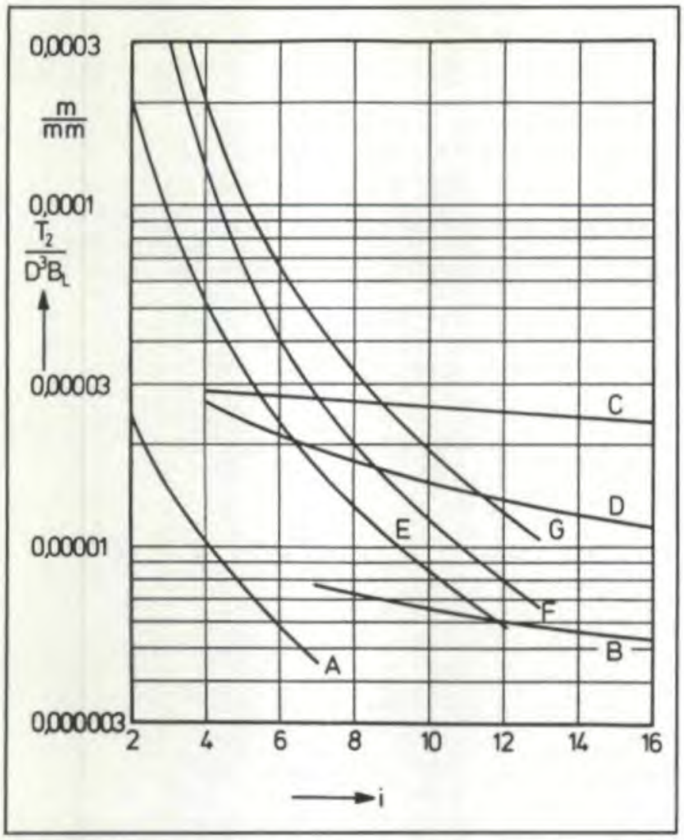

Fig. 4 – Dependence of relative (to size) torque of gear units according to Fig. 3 on the transmission ratio i. T_2 = torque of shaft 2 in Nm; D = design length or diameter in m; B_L = load value in N/mm^2

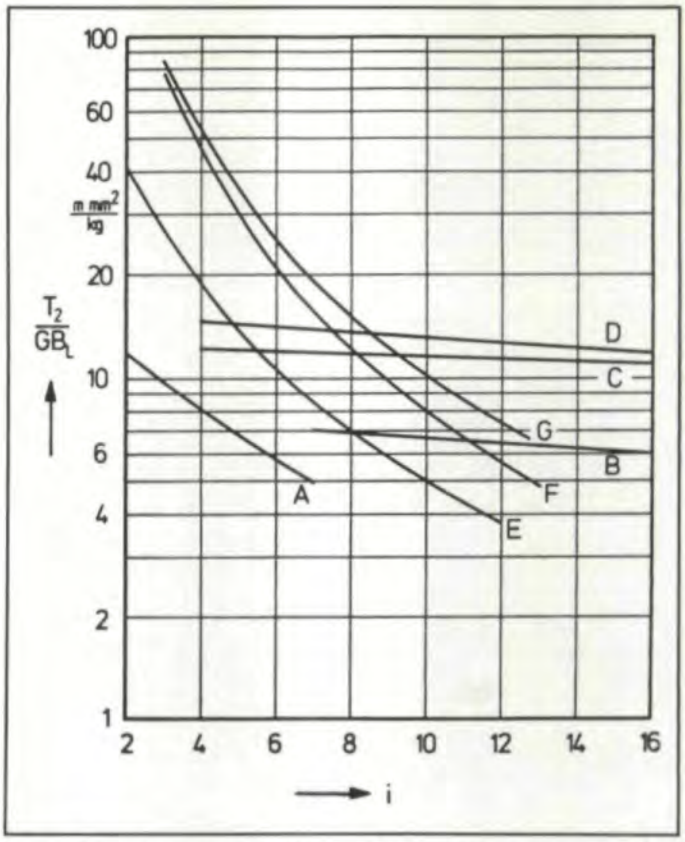

Fig. 5 – Dependence of relative (to weight) torque of gear units according to Fig. 3 on the transmission ratio i. T_2 = torque of shaft 2 in Nm; G = gear unit weight in kg; B_L = load value in N/mm^2

distribution is achieved by the herringbone gears of the high speed stage and the axial positioning of shaft 1.

In the case of gear unit D, the power of the high speed stage is equally distributed to three intermediate gears by the radial positioning of the small central gear on shaft 1. The power in the low speed stage is equally distributed through the three herringbone gears by the axial positioning of the three intermediate shafts.

To achieve equal power distribution on the three intermediate gears in the case of E, F and G, the small central gear on the shaft 1 of large units should be radially positionable.

The large outer gear is an internal gear and is connected to shaft 2, in the case of E, and to the gear unit casing in the case of F and G respectively. Gear units F and G both have the planet carrier and shaft 2 forming one unit.

The intermediate gears rotate about the central axis as planets. Herringbone gears and axial positioning of the intermediate gears give an equal power distribution through six branches in the case of gear unit G.

Power branching influences the size and weight of a gear unit. Figs. 4 and 5 show these relationships as a function of the total transmission ratio

$$i = \frac{n_1}{n_2} . \qquad (1)$$

n_1 and n_2 are the rotational speeds of shafts 1 and 2 respectively.

Figs. 4 and 5 have plotted T_2/D^3B_L and T_2/GB_L against the transmission ratio i whereby T_2 = torque of low speed shaft 2, D = gear unit size, G = gear unit weight and B_L = load value. B_L is calculated from the following equation.[4]

$$B_L = \frac{Fu}{d \bullet b} \qquad (2)$$

with F_u = tangenital force, d = diameter of pinion and b = facewidth.

Equation (3) gives the allowable load value B_L^* approximately.

$$B_L^* = \frac{f_W}{K_A} B_o \qquad (3)$$

K_A is the application factor and f_W is the load factor. With repeated load on the gears $f_W = 1$ and with alternating load $f_W = 0.7$.

Gear units E, F and G have intermediate gears with an alternating load. For gear units dimensioned for an infinite fatigue life the value of B_o is approximately 4 . . . 5 N/mm^2 for case hardened gears and 1 . . . 1.3 N/mm^2 for through hardened gears. Gear size D and gear weight G can be approximately determined for known values of torque T_2 with the help of Figs. 4 and 5 and equations (2) and (3).

The gear weight G includes the weight of solid gears, shafts and case. The gear case forms shown schematically in Fig. 3, wall thicknesses = 0.02 D and the density of steel are the basis for the weight calculation.

Gear units G and F have the largest relative torques for low transmission ratios and, therefore, are preferred. A transmission ratio limit of $i \approx 8$ is obtained with $f_W = 0.7$ for gear units G and F.

The relative torque of units G and F can be favourable even for higher transmission ratios when two gear units are connected in series. For example, two F type gear units having transmission ratios $i_1 = i_2 = 4$ and are connected in series. The total transmission ratio is $i = i_1 \bullet i_2 = 16$. The total weight of these two units is about 1.8 times less than a single D type gear unit with $i = 16$.

Fig. 6 shows the relation between the efficiency n of gear units according to Fig. 3, and the transmission ratio i according to equation (1). Only gear tooth meshing losses are considered since this loss is significantly larger than all the other losses for gear units under full loading.

For the one stage gear unit, A the efficiency corresponds to the efficiency n_z of one gear meshing, i.e.

$$n = n_Z . \tag{4}$$

But in the case of B, C, D and E the total efficiency is the product of the slow and high speed stage efficiencies, i.e.

$$n = n_{Z1}\, n_{Z2}. \tag{5}$$

Gear units F and G transmit part of the power directly without any loss. Here, the efficiency with shaft 1 as the input shaft is

$$n = \frac{1}{i}\,\{1 + (i - 1)\, n_{Z1}\, n_{Z2}\}. \tag{6}$$

The tooth meshing efficiency n_Z of a gear pair is obtained from the following equation

$$n_Z = 1 - f_Z\, \mu_Z. \tag{7}$$

The geometrical factor f_Z is calculated[5] from equation (8) with the numbers of teeth z_1 and z_2 respectively of pinion 1 and gear 2

$$f_Z = \pi \left(\frac{1}{z_1} + \frac{1}{z_2} \right) \quad (E_1 + E_2) . \tag{8}$$

z_2 is negative for internal gears. The values for E1 and E2 are given by partial transverse contact ratios ϵ_1 and ϵ_2 of pinion 1 and gear 2 respectively.

$$E_{1,2} = 0.5 - \epsilon_{1,2} + \epsilon^2_{1,2} \quad \text{for } 0 \leqq \epsilon_{1,2} \leqq 1 \tag{9}$$

$$E_{1,2} = \epsilon_{1,2} - 0.5 \quad \text{for } \epsilon_{1,2} > 1 \tag{10}$$

$$E_{1,2} = 0.5 - \epsilon_{1,2} \quad \text{for } \epsilon_{1,2} < 0 \tag{11}$$

The equations (9) to (11) derived for spur gearing in[6] are also approximately valid for helical gearing.

It is assumed that the tooth flank coefficient of friction $\mu_Z = 0.06$ for all the tooth meashings of the gear units in Fig. 3. Internal gears enhance the formation of a good elastohydrodynamic oil film due to the complementary profiles of the meshing flanks.

However, the sliding properties of internal gearings are poorer due to higher tooth flank roughness and, thus, the assumption of the same coefficient of friction for external and internal gearing approximately holds good. The gearing geometry for Fig. 6 is according to DIN 3960 for gears having no addendum modification and number of teeth of the pinion $z_1 = 17$ for all the gear meshings.

Gear unit A with a single stage has the best efficiency, see Fig. 6. Since there are two gear meshings in gear units B, C, D, E, F and G, the efficiency curves lie below that of case A. In the case of gear units E, F and G, the internal gear provides a favourable geometrical factor f_Z, thus, a better efficiency compared to gear units B, C and D which have only external gearings. The power component transmitted without loss for gear units F and G gives a further improvement in efficiency.

Planetary gear units F and G, in view of their size, weight and efficiency, are the most favourable choice. Considering the high manufacturing cost to obtain a good tooth surface finish on internal gears, planetary gear units have a reduced advantage over gear units with power branching and external gears only.

Design of Large Gear Units

Three large gear units with power branching, which have proved to be useful in practice, are dealt with here. The rolling mill drive shown in Fig. 7 has a two way power branch similar to gear unit C of Fig. 3. It has four stages to attain

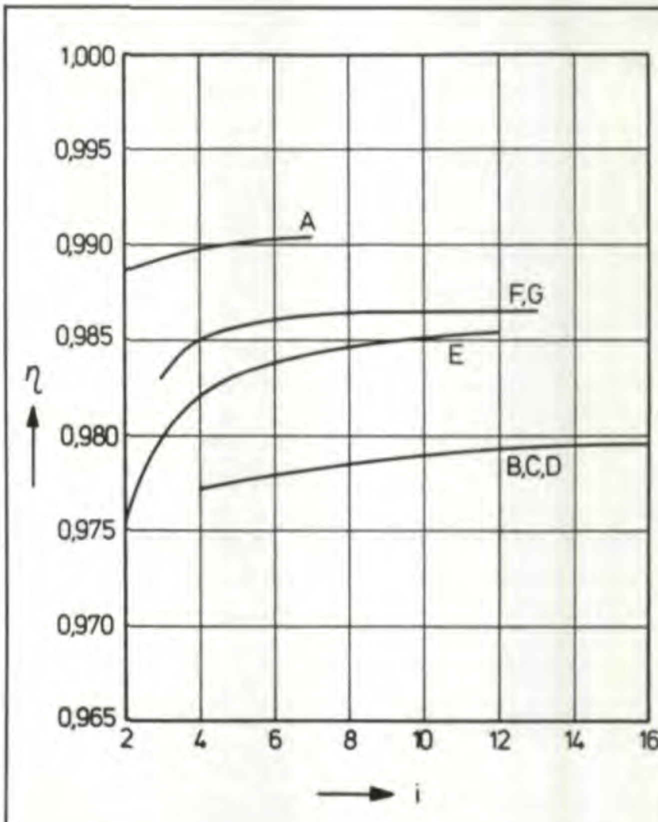

Fig. 6 – Dependence of efficiency η on the transmission ratio i for gear units according to Fig. 3.

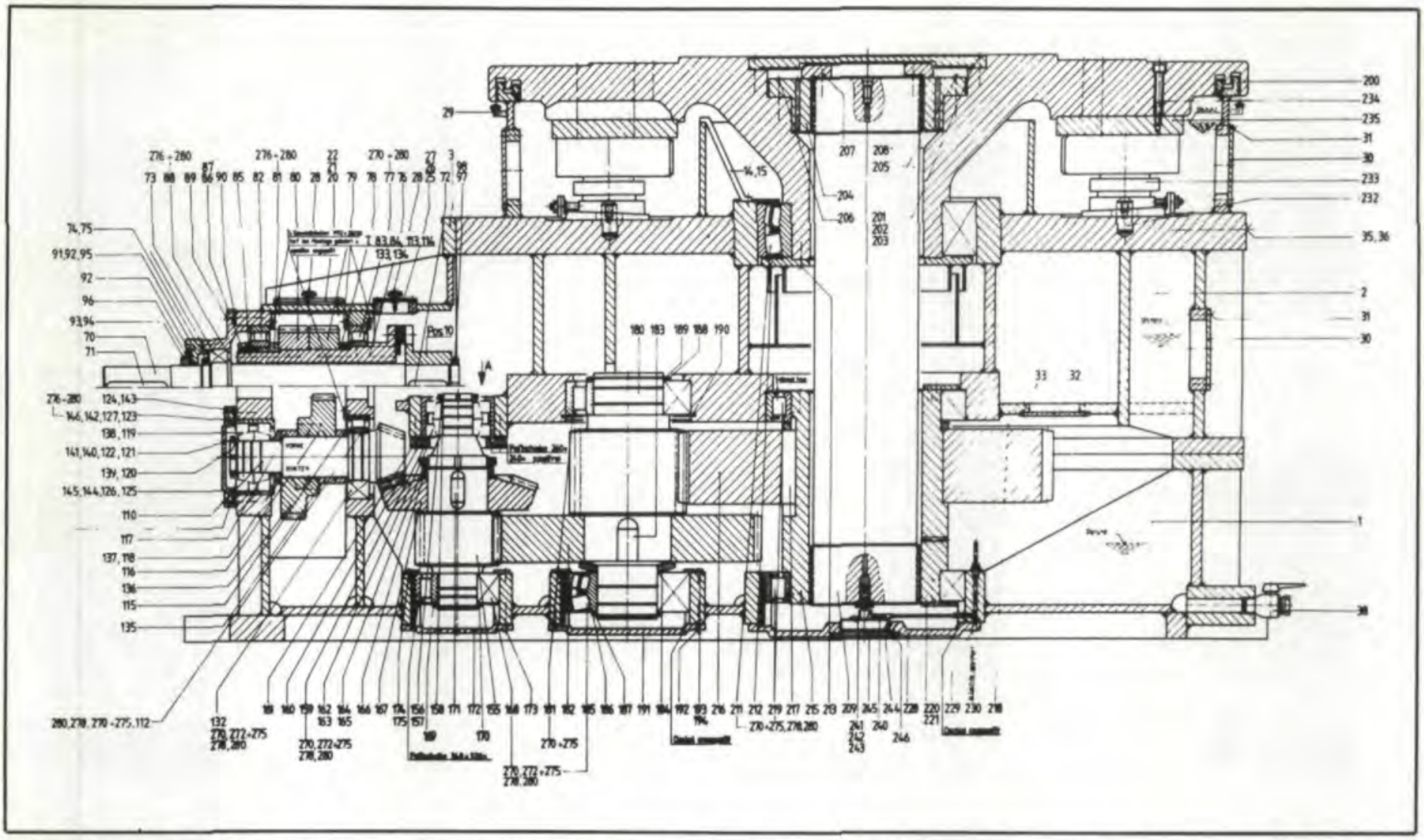

Fig. 7—Rolling mill gear unit with two way power branching (see scheme in Fig. 8). Power P = 1350 kW; shaft speeds n_1 = 980 min^{-1}, n_2 = 24.5 min^{-1}. Total weight including flange disk G = 82 t. (Flender-Kienast-Conception)

the high transmission ratio, whereby, the second stage is a bevel gear stage as the output shaft is in the vertical position. Fig. 8 shows the gearing system.

The input shaft 1 is connected to an axially free hollow shaft 2 via an axially moveable coupling made of steel lamellas. Herringbone gear 4, on the hollow shaft, branches the power equally due to its axial positioning. Bevel gears 5 and 6, with cyclo-palloid-spiral teeth and longitudinally crowned flanks, are case hardened and are cut after heat treatment "High Power Gear-teeth" (HPG). The power is transmitted onward to the output shaft via the helical gears 7, 8, 9 and 10. All the helical gears are case hardened, and the tooth flanks are ground. A stub shaft, with coupling teeth at the shaft end, connects the output gear with the flange disk, shown in Fig. 7.

Hydrostatically lubricated axial sliding bearings with tilting segments carry the large forces from the milling operation and the weight of the output shaft, flange disk and mill pan (not shown) and transmits the same to a strongly designed gear case. Each sliding bearing segment has a temperature feeler gauge to monitor bearing load. No inspection during operation is necessary due to an adequate oil supply.

Fig. 9 shows the central gear unit of a tube mill. The gear unit has, in the first stage, no power branching and has two input shafts. One of the two input shafts serves as an auxiliary drive, and is driven at a greatly reduced speed via an overrunning clutch and auxiliary gear unit (not shown). When the main input shaft is driving at a higher speed, the auxiliary drive will be disengaged due to the overrunning clutch. Both the main stages of the gear unit in Fig. 9 distribute the power similar to gear unit D in Fig. 3. The first main stage with spur gearing branches the power three ways. Equal power branching is achieved by the radial positioning of a central pinion on a shaft designed for elastic deflection. The second main stage, with the herringbone gear, branches out the power six times. The eccentric intermediate shaft has axial freedom ensuring their equal loading.

The large intermediate gears belonging to the first main stage are mounted onto the intermediate shaft by oil hydraulic shrink fitting. Thus, an exact alignment of the gearing of the

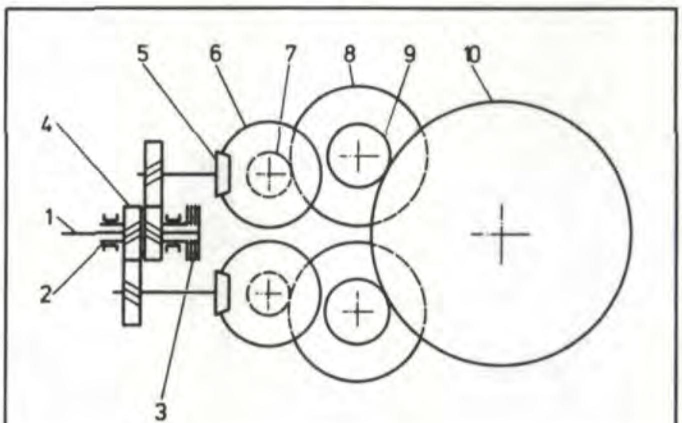

Fig. 8—Gearing scheme of rolling mill gear drive shown in Fig. 7. 1 = input shaft; 2 = hollow shaft; 3 = steel lamella disk coupling; 4 = herringbone gear; 5, 6 = bevel gear pair; 7, 8, 9 = spur gears; 10 = gear on output shaft

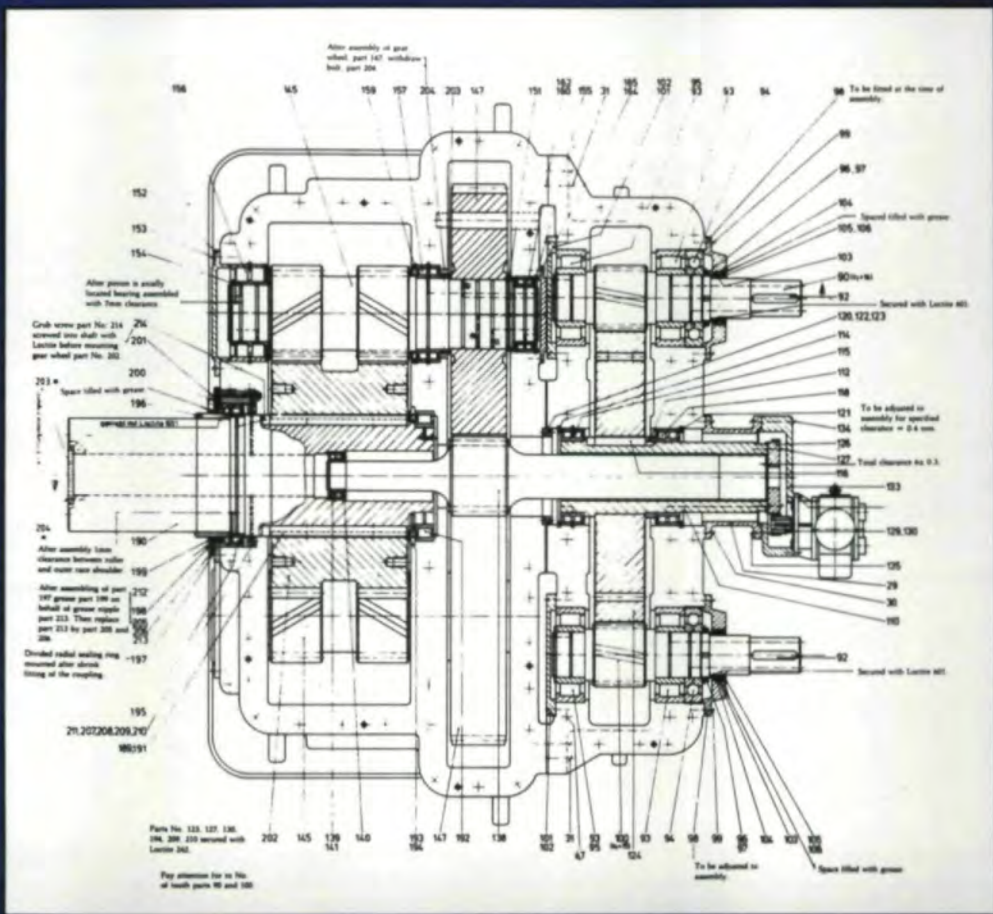

Fig. 9 – Tube mill central gear unit with two power branching stages. Transmission power $P = 4500$ kW; shaft speeds $n_1 = 490$ min^{-1}; $n_2 = 12.6$ min^{-1}; gear unit weight $G = 100$ t

two main stages during assembly is possible requiring only a small radial displacement of the central pinion of the first main stage. Measurements have shown that the radial displacement is not more than 0.2 mm and the variation of torque distribution is 4% maximum.[7]

All the gears of the gear unit, shown in Fig. 9, are case hardened and ground. It is worth noting that the resultant radial forces on the output shaft are theoretically non-existant, thus, the output shaft has relatively small roller bearings.

A two-stage planetary gear unit for a ball mill drive is shown in Fig. 10. Each stage has spur gearing and three way power branching and corresponds to the gear unit F of Fig. 3 with the internal gear fixed to the gear case.

A uniform force distribution on the three planet gears is achieved by the free pivoting configuration of the central pinion of each of the two planet stages. The sun gear centers itself in such a way that the three radial meshing forces of the planets are equal, thus, ensuring an equal transmission of the tangential forces.

Due to the equilibrium conditions, the three meshing points of the planet gears with the annulus also have equal tangential and centering forces. The sun gear which is centered by the planet wheel teeth has radial freedom due to the backlash of the teeth and the double jointed clutch in each stage. The ring gears of both the stages are of quenched and tempered steel. They are hobbed or cut and stress relieved after the final machining process.

Summary

Gear units, with case hardened gears and internal power branching, are relatively small in size and weight. This type is particularly suited for large gear units to transmit high powers. Case hardened and ground gears, due to their higher flank wear resistance and strength, give a higher load carrying capacity than through hardened gears. The strength characteristic below the tooth flank surface of a case hardened gear matches well with the stress characteristic.

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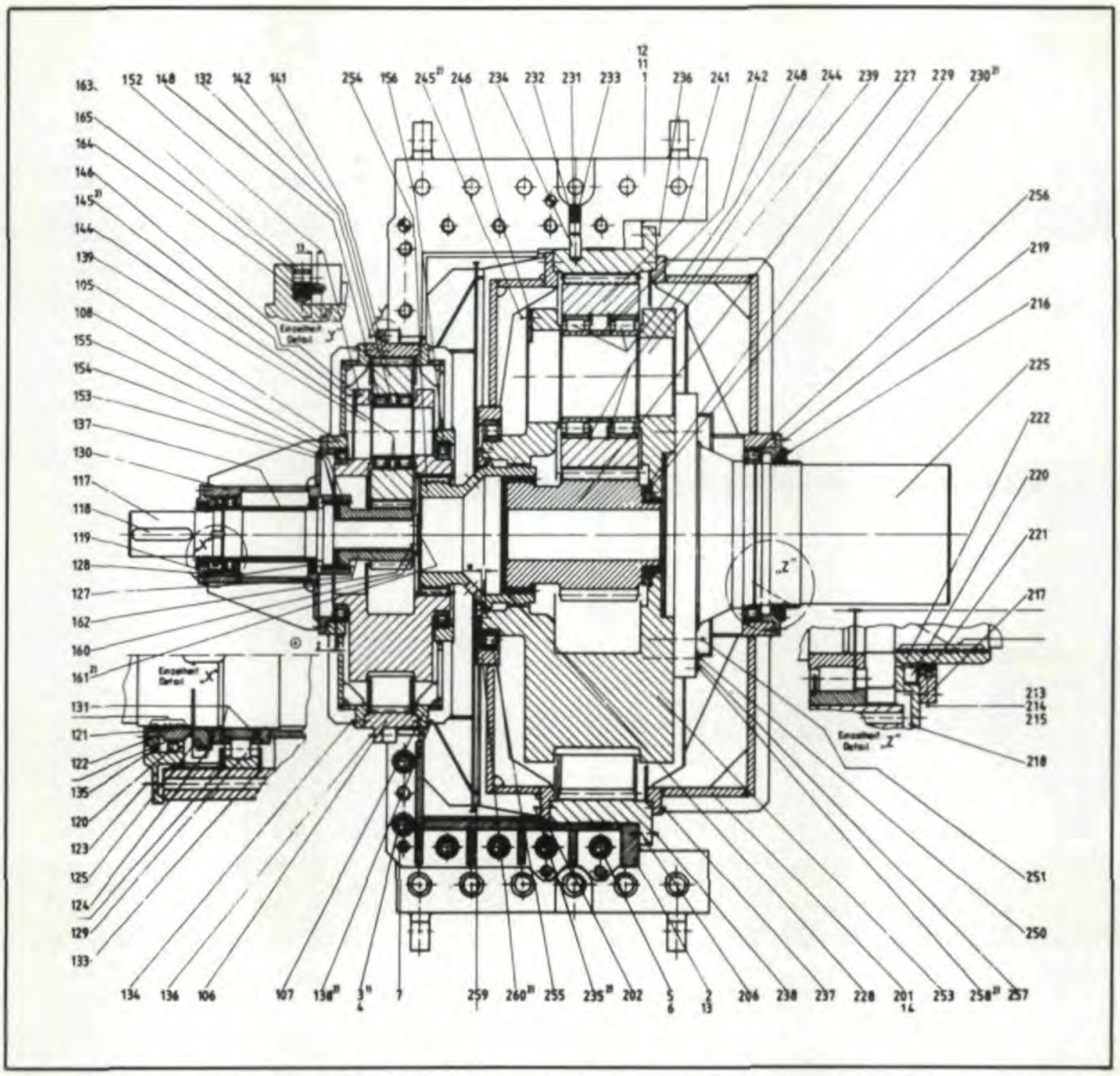

Fig. 10—Two stage planetary gear unit for ball mill drive with three way power branching per stage. Transmission power P = 4050 kW; shaft speeds n_1 = 485 min^{-1}; n_2 = 13.8 min^{-1}; gear unit weight G = 72 t

High Power Transmissions
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A comparison of size and weight relative to output torque shows that among the power branching gear units planetary gear units are favourable and better especially for smaller transmission ratios of single planet stages. The relative torque decreases with increasing transmission ratio. For transmission ratios above i = 8 the power branching gear units having external gearing only are preferred. Their relative torque decreases at a lower rate with increasing transmission ratio. However, two or more planetary gears, coupled one after another, may still be more advantageous for higher transmission ratios above i = 15.

The efficiency of planetary gear units is comparatively better than that of other power branching gear units. However, for higher transmission ratios, one must couple more planetary gear units one after another and this advantage may be lost.

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