JERKY LOADS ON SURFACE-HARDENED GEARS

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16. Abstract Gear drives are frequently exposed to jerky stresses which are greater than their fatigue limit. These stresses are considered in gear calculations, first, by shock factors when the transmission is to be designed as high-endurance with regard to overloads and, second, in the form of operating ratios when the design is to be time-enduring with regard to overloads [1, 2]. The size of the operating radio depends not only on torque characteristics, drive and processing machine, but also on the material and heat treatment [1]. As a result of these interdependencies and because of the fact that damage occurs again and again in practice in the form of transmissions with surface hardened gears which break after a very long operating time (explained by seldom-occurring jerky loads), the present research project "Jerky Loads on Surface-Hardened Gears" resulted. This study was financed by the Research Society for Propulsion Technology e.V. and the Study Group on Industrial Research Accords (AIF).

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JERKY LOADS ON SURFACE-HARDENED GEARS

H. Rettig* and X. Wirth*

ABSTRACT. Gear drives are frequently exposed to jerky stresses which are greater than their fatigue limit. These stresses are considered in gear calculations, first, by shock factors when the transmission is to be designed as high-endurance with regard to overloads and, second, in the form of operating ratios when the design is to be time-enduring with regard to the overloads [1, 2]. The size of the operating ratio depends not only on the torque characteristics, the drive and processing machine, but also on the material and heat treatment [1]. As a result of these interdependencies and because of the fact that damage occurs again and again in practice in the form of transmissions with surface-hardened gears which break after a very long operating time (this is explained by seldom-occurring jerky loads), the present research project "Jerky Loads on Surface-Hardened Gears" resulted. This study was financed by the Research Society for Propulsion Technology e.V.** and the Study Group on Industrial Research Accords (AIF). This paper is an excerpt from the dissertation work performed by X. Wirth at the Technical Department for Mechanical Engineering.

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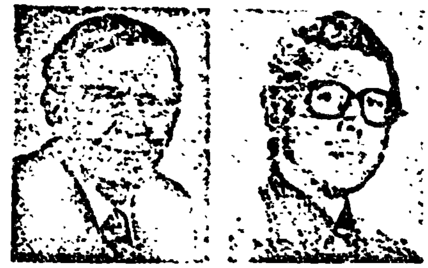
1. Introduction /1*



It was the goal of the studies described here to use dynamic tests and pulsation tests to determine how much jerky overloads (regardless of the stress amplitude) can be endured by use-hardened and gas-nitriding gears without reducing their fatigue limit, and what role is played by load intermixture.

The preliminary research of the literature [3] showed the important result that the load-increase period during mesh is significantly shorter than the torque jerks occurring on a transmission in practice if we ignore very low RPM (e.g., during start-up; see Figure 1). So an external torque jerk only influences the instantaneous load amplitude, but not the load-increase time of the gear stress.

On a very slow-running transmission, an external torque jerk is considered as a simple overload, although we should remember that under some circumstances, this load affects more than one tooth. Because of this fact, it was possible to determine the influence of jerky overloads on the fatigue limit by ascertaining the damage line according to French [4].

By definition, the French damage line in the stress-cycle diagram is the geometric locus of all stresses "σ, N", which designate the beginning of fatigue-limit reduction due to "σ, N". The abbreviations are summarized in Table 1.

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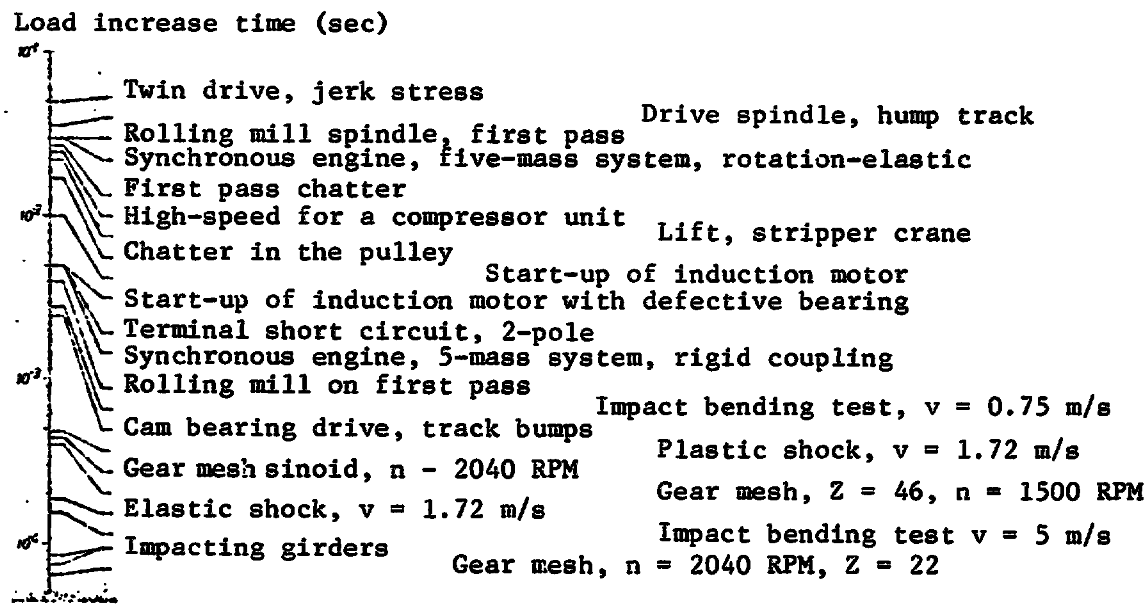


Figure 1. Load increase times for various shock processes

TABLE 1. ABBREVIATIONS

a	axis separation
b	tooth width
K_v	dynamic factor
m	modulus
n	RPM
N	number of load cycles
N_D	number of load cycles at fatigue limits
N_R	number of load cycles to break
N_S	number of load cycles to damage according to French
N_{Stoss}	number of load cycles of the jerky overload
N_W	number of load cycles to break of the Wohler-curve
ΔP	threshold rate
P	rate of stress
R_t	peak-to-valley height
s	dedendum chord
t	notch depth
x	profile shift factor
z	number of teeth
α	mesh angle
β	angle of skew
γ	notch angle of sight
Q_F	dedendum rounding-off radius
Q_K	notch radius
σ	stress
σ_D	fatigue limit

2. Test Gears and Equipment /2

2.1. Test gears

Table 2 provides information about materials, manufacture, and heat treatment of the gears used for the dynamic and pulsation tests.

TABLE 2. MANUFACTURE AND HEAT TREATMENT OF TEST GEARS A TO E

Gearing	Material	Manufacture and heat treatment
A	16 Mn Cr 5	Prominence milled, use-hardened, ground sides
B	16 Mn Cr 5	Normal mill rolled, use-hardened, ground sides with ground notch in the dedendum
C	16 Mn Cr 5	Normal mill rolled, use-hardened, ground sides, cut dedendum round-off
D	31 Cr Mo V 9	Normal mill rolled, ground, gas-nitrided, without notch in dedendum
E	31 Cr Mo V 9	Normal mill rolled, gas-nitrided, with notch in dedendum

The modulus m = 2.25 mm was selected for the gas-nitrided gears in order to obtain a definite tooth break as a failure criterion in the dynamic test (use-hardened gears: m = 3 mm).

Grinding the gears proceeded by the Maag-0° process [5]; in version D, it preceded heat treatment; in all other cases, it followed heat treatment.

The hardness depth of gears A, B, and C was 0.5 to 0.7 mm; the nitriding hardness depth of gears D and E was 0.5 to 0.6 mm on the raw component. Hardness values from 660 to 770 HV1 (use hardened) or 680 to 730 (gas nitrided) were measured at the unground edge. The core hardnesses varied between 390 and 460 HV1, or 275 and 300 HV1.

The geometric ratios of the dedendum were maintained by means of the definitions in Figure 2. As notch depth t, we means the maximum ground depth in the dedendum, measured perpendicular to the contour of the unground dedendum round-off.

Table 3 shows the results of the dedendum measurement and the roughness values determined in the break-endangered zone.

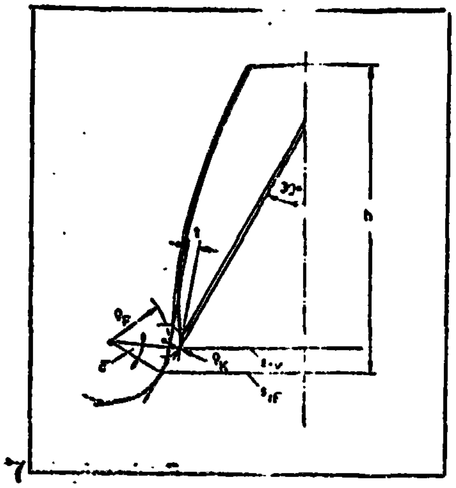


Figure 2. Defined quantities in the dedendum

TABLE 3. RESULTS OF THE DEDENDUM MEASUREMENT (VERSIONS A TO E)

Gearing	Radius (mm) for 30° tangent		Notch depth t (mm)	Notch location γ (°)	Peak-to-valley height R_t (μm)	
	Dedendum	Notch			Dedendum	Notch
A	1.60-1.70	-	-	-	12-25	-
B	-	0.23-0.30	0.16-0.26	20-28	-	2-3
C	0.94-1.47	-	-	-	3-5	-
D	0.80-0.95	-	-	-	4-6	-
E	-	0.20-0.35	0.16-0.26	10-20	-	3-4

2.2. Test apparatus

To determine the Wohler lines and damage lines of the individual gears, the following test apparatus was available. For dynamic tests: two stress test stands with preselection counter, each separated by 91.5 mm; for pulsating tests: an electronically controlled hydropulse unit of Schenck design POZ 0340 (6 Mp).

By selecting a suitable hydraulic cylinder, by building a special stepped test frame, and by appropriate electronic devices, operation was possible with pulse-like loads at relatively small load increase times (see section 3). Figure 3 shows the pulsator unit.

Measurement of the relative motion of the two pulsator end pieces during the test run proceeded by means of an inductive precision-motion pick-up Hottinger W1T/2 model; recording the tooth elongations and the actual and theoretical load values occurred by using a multichannel photographical recorder Honeywell 2208 A model.

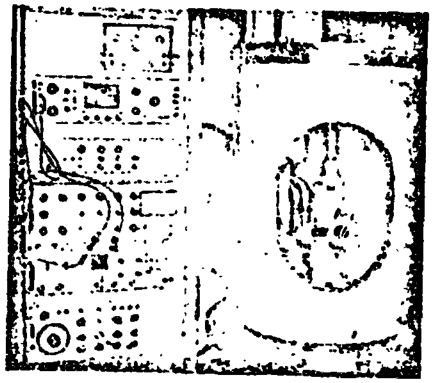
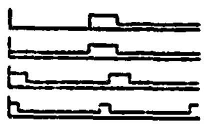


Figure 3. Electronically controlled hydropulse unit

3. Performing the Tests

In Table 4, the test plan is shown in summarized form. First, the Wohler lines of gears A to E were determined in the pulsator and dynamic tests. In a second test, the damage lines from French (load drop I) were determined. Finally, the influence of the time distribution of load jerks was examined over the life span for the location of damage lines (load cases I', II, and III).

TABLE 4. TEST PLAN

Load case	Gear data	
I, I', II, III 	m = 3 mm z_1/z_2 = 23/38 a = 91.5 mm b = 10 mm mx_1 = 0 mm mx_2 = 0 mm	m = 2.25 mm z_1/z_2 = 31/50 a = 91.5 mm b = 10 mm mx_1 = 0 mm mx_2 = 0.38 mm

Type of test	Damage line	Material and mfg. A	B	C	D	E
Dynamic	Load I	WL* SL**	WL SL	WL SL	WL SL	WL SL
Dynamic	Load I'	SL	–	–	–	–
Pulsating	Load I	WL SL	WL SL	WL SL	WL SL	WL SL
Pulsating	Load II	SL	SL	–	–	SL
Pulsating	Load III	–	SL	–	–	–

*Wohler line

**Damage line

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The quantities held constant for all dynamic tests are shown below:

axis separation:	$a = 91.5$ mm
pinion RPM:	$n = 3000$
drive power	$N_N = 11.5$ kW
lubrication:	injection lubrication (1.2 l/min)
lubricant:	FVA oil no. 3 with 4% sulfur-phosphorus added
oil injection temperature:	$t_{oil} = 50°$ C $n = 60$cST (50°C)
type of drive:	wheel driven pinion

In order to be able to make a reliable statement about the actual dedendum load, dynamic gear force measurements were performed after the conclusion of the damage tests. It turned out that a pre-resonance of the transmission system occurred in the region of the test RPM (according to an estimate made at the beginning of the tests by using the ISO-outline [6], the main resonance of the gearing at modulus m = 3 mm was ca. 12,000/min). Figure 4 shows the measured course of the dynamic factor for all five gear versions in the test RPM range.

In order to keep test conditions as similar as possible for the dynamic and pulsating tests, the pulsator load in the fatigue limit range ("jerky overloads") was applied as impulses (frequency: maximum 18 Hz).

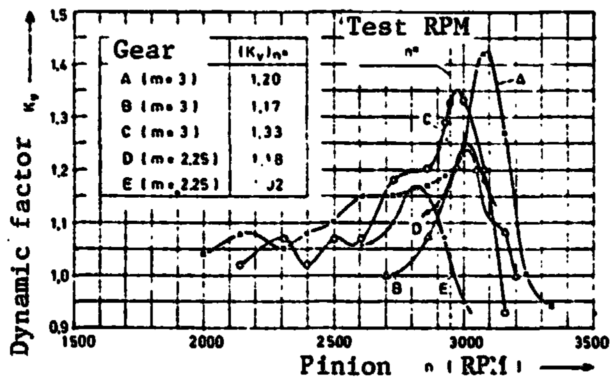

Figure 4. Dynamic factors of gears A to E (for permanent load)

The influence of the load rate on the breaking load was examined on version A. The result is shown in Figure 5. From this, it proceeds that the average break load (50% failure probability) for "jerky loads" ($P = 24 \cdot 10^4$ kp/s) is about 6.5% higher than for "static" loads ($P = 60$ kp/s). /3

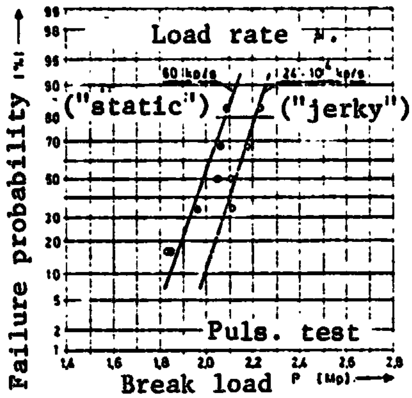

Figure 5. Influence of load rate on break load of gear

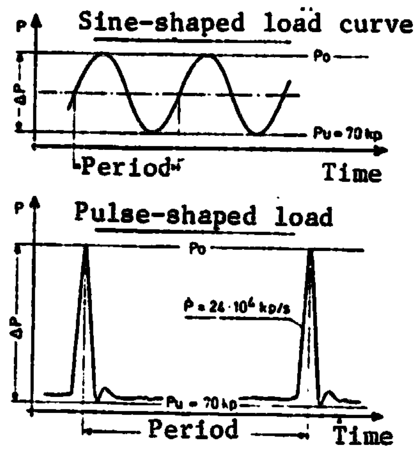

Figure 6. Load chronology in pulsating test in fatigue limits and endurance strength region

All fatigue strength tests were performed under a sine-shaped load curve (110 Hz) for reasons of time saving, since no influence of load shape was expected on the fatigue strength [3]. Figure 6 shows the chronological load curves for the pulsating test. Stressing the test gears occurred between two parallel pulsator end pieces (Figure 4) over 4 (m = 3 mm) or 5 (m = 2.25 mm) teeth.

The number of cycles to break was determined for the endurance tests in the pulsator in three different manners:

- The break in the hardened layer showed up by means of an acoustic signal for all pulse loads clearly above the fatigue limit.
- A small permanent elongation discernable on the elongation-time chart appeared simultaneously with the acoustic noise.
- By means of the color-penetration process, the crack was made visible right from the beginning (the color penetration process was not successful at first for unstressed teeth; it was successful only after crack expansion under stress).

4. Damage Lines after Dynamic Tests

Figure 7 - 10 show the Wohler and damage lines according to French, as determined in the dynamic test. All lines are valid for a failure probability of 10%, and do not consider the measured dynamic factors. The loads thus correspond to the static values set at the test stand. This is of no importance to the validity of the result, since only relative values are compared. Although the endurance lines consist of 5 - 10 test points each, the fatigue limits of the individual versions were determined statistically with ca. 10 gear pairs by means of the stair procedure [7].

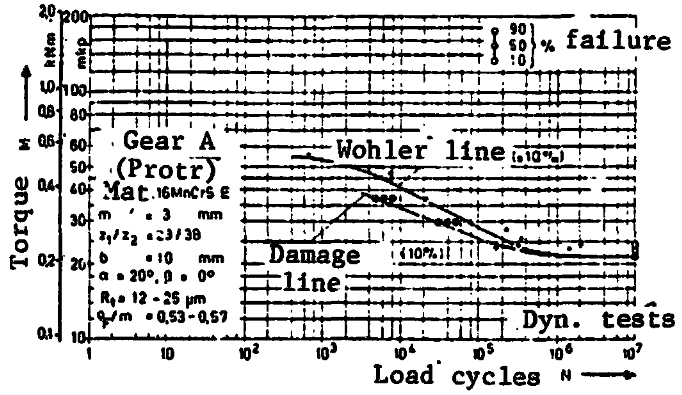

Figure 7. Wohler and damage lines of gearing A after dynamic tests

Protr — protrusion; mat — material

The corresponding damage lines were determined statistically with the same procedure for two load horizontals. The procedure was such that one fatigue limit test was connected to one run with overload; this served to determine whether the fatigue limit found in the Wohler test had been retained. The load in this second stage corresponded in all cases to the original failure limit with a 10% failure probability.

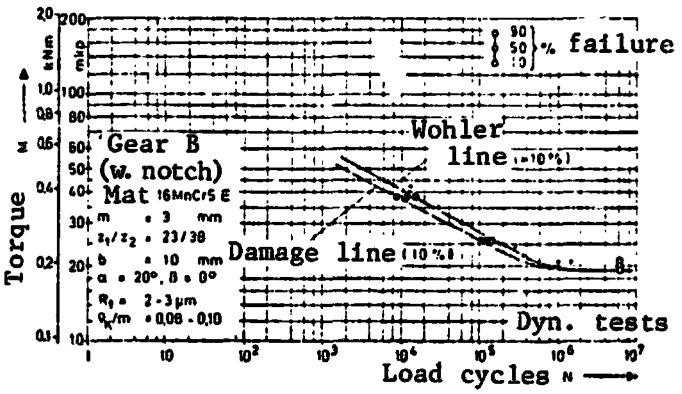

Figure 8. Wohler and damage lines of gearing B after dynamic tests

Mat — material

Because of these test-caused conditions, only Wholer and damage lines with 10% failure probability correlate. Since only at 10% for all two-stage tests did the load amplitude of the second stage correspond to the actual fatigue limit. Ninety percent of all two-stage tests were consequently run with too little stress in the second stage, which in turn resulted in a somewhat too high load cycle for the damage according to French.

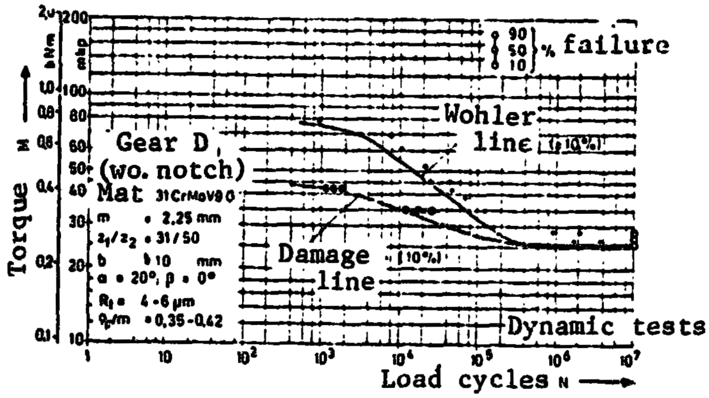

Figure 9. Wohler and damage lines of gear D after dynamic tests

Mat — material

With regard to the problems posed in the introduction, the dynamic tests yield the following results:

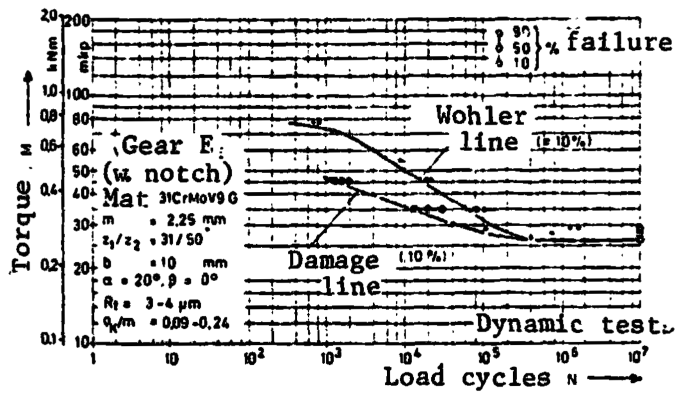

Figure 10. Wohler and damage lines of gear E after dynamic tests

Mat — material

a) Damage according to French, i.e., a reduction in the fatigue limit due to jerky overload, resulted — with regard to full load service life — earlier, the greater the overload amplitudes were. For example, for gear A (protrusion milled, use hardened) with a ratio of σ_{stoss}/σ_D = 1.62, damage according to French occurred right at 28% of the number of load cycles to break, whereas at σ_{stoss}/σ_D = 1.16, this type of damage was observed only at 52% of the number of load cycles to break. /4

b) Gas-nitrided gears were significantly more sensitive to overload than use-hardened ones. At 1.6 times overload, the fatigue limit drop of gear D began at ca. 5% (compared to 30% for gear A) of the number of load cycles to break.

c) Ground notches in the dedendum had a favorable effect on the relative overload endurance in this case (related to the fatigue limit); naturally, the strength itself was reduced. For example, damage according to French occurred for version B (use hardened with ground notch) only at ca. 60% of the number of load cycles to break.

d) The test gears C (use hardened) with a ground dedendum showed a similar damage behavior in the range examined as the unnotched gear A. Therefore, illustration of the corresponding diagram was omitted. But the results are included in the summarizing overloadability diagram (Figure 11). This diagram contains the results of the two-stage tests of French: The load cycle ratios N_S/N_W (damage according to French/number of cycles to break) for all gear versions studied were plotted with the overload ratio σ/σ_D.

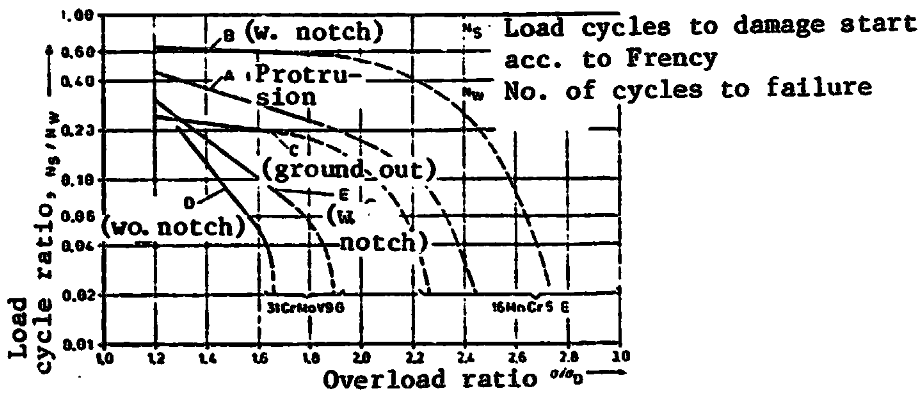

Figure 11. Loadability of gears A to E after dynamic tests

5. Crack and Damage Lines after Pulsation Tests

The Wohler, crack, and damage lines of gears A, B, D and E determined in the pulsation test are illustrated in Figure 12 - 15 in the load cycle range of $1 \leq N \leq 2 \cdot 10^6$. In contrast to the dynamic tests, the endurance lines were also statistically covered. Figure 16 shows the illustration of the crack and number of load cycles to failure of gear A in the probability net with logarithmic normal distribution. The determination of endurance limits and of the damage line initiation proceeds analogous to the dynamic tests (section 4) by means of the stair procedure (20 - 25 test points per endurance limit).

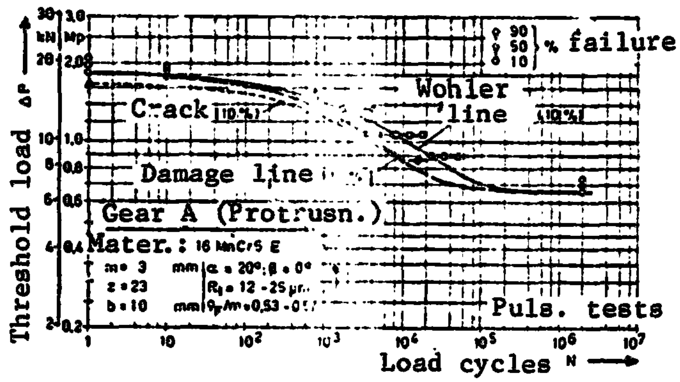

Figure 12. Wohler and damage line of gear A after pulsation tests

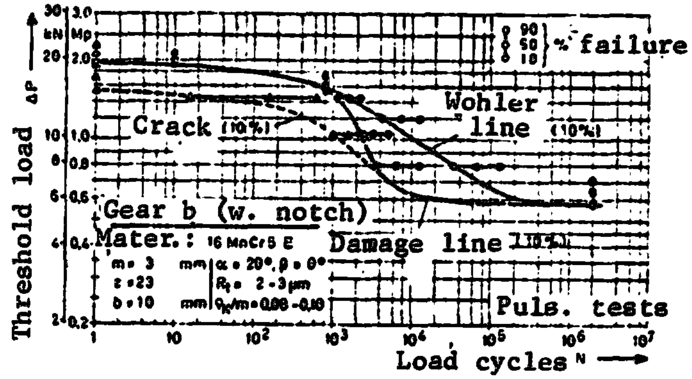

Figure 13. Wohler and damage line of gear B after pulsation tests

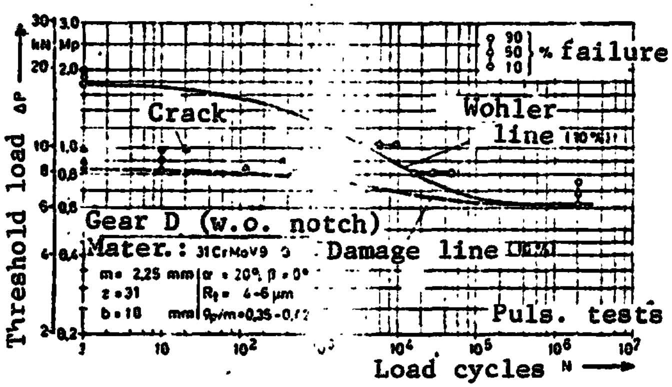

Figure 14. Wohler and damage line of gear D after pulsation tests

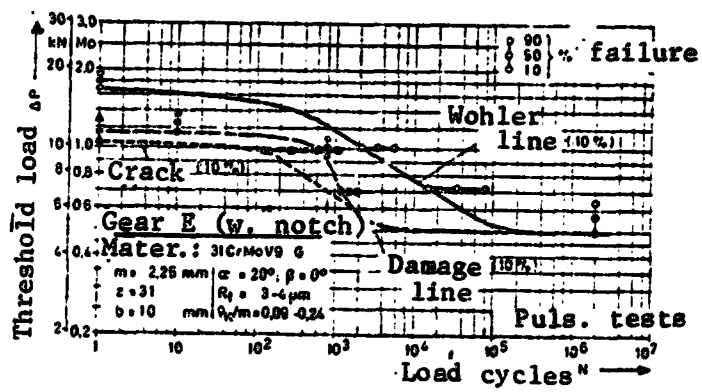

Figure 15. Wohler and damage line of gear E after pulsation tests

The pulsation tests should show primarily whether the damage lines ascertained in the dynamic test stand and in the pulsator are in a similar systematic interrelation as the Wohler lines of the two test processes [8] and what relation exists between crack and damage lines.

The test results can be presented as follows:

a) The damage lines of unnotched gears A and D exhibit a similar curve as the corresponding ones from the dynamic test. However, the damage line of gear A opens into a Wohler line (Figure 12) at very high overloads. The crack results, for this gear, in the region of higher load amplitudes just before the damage (according to French) occurs; for smaller overload, crack and damage lines run together. Here, there develops the beginning of the fatigue crack from the crack itself. The low overloadability of the gas-nitrided version D (without notch) determined in the dynamic test was also confirmed in the pulsator. Both crack and damage seem to have a common beginning here (Figure 14).

b) Large differences between the damage lines of dynamic and pulsation tests resulted for versions B and E with ground notches in the dedendum (Figures 13 and 15). The damage line of the use-hardened version B practically coincides with the Wohler line for very small overload cycles ($N < 10^3$), whereas in the region between 10^3 and $4 \cdot 10^3$ it has a steep drop. So, in the pulsator, a 20% overload led to a drop in endurance after about 7% of the load cycles to failure, whereas in the dynamic test, the damage according to French occurred in the total overload region only at 60% of the full load service life (see section 6).

The crack line runs approximately parallel to the Wohler line over the entire load cycle range and opens — at small overload amplitudes — into the damage line. This means that a crack which has resulted from a small number of high overloads does not necessarily cause any loss in endurance strength, whereas at a small overload stress above the endurance limit, the damage line according to French is already reached.

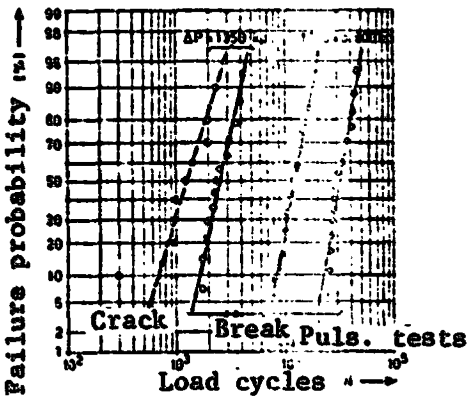

Figure 16. Endurance of gear A

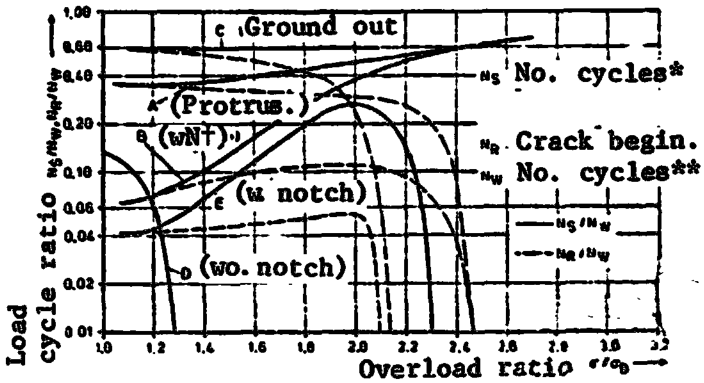

Figure 17. Overloadability of gears A to E according to pulsator tests

*Number of load cycles for damage to begin according to French.
**Number of load cycles to failure
†(with notch)

A qualitatively equal curve of crack and damage lines shows up for the gas-nitrided version E with cut notch (Figure 15). However, the damage line of this gear also has a clear separation from the Wohler line at small load cycles.

d) Version C (use hardened, dedendum ground out) showed a similar crack and damage line curve in the pulsator as gear A. The result is derived from the overloadability diagram (Figure 17), which is similar to Figure 11.

6. Significance of the Different Results from Dynamic and Pulsation Tests /5

Since the damage lines from dynamic and pulsator tests on the gears without cut notches in the dedendum do differ, the following discussion is limited to the notched versions B and E.

The characteristic steep drop of the damage line observed here might possibly be caused by an initial increase in fatigue limit of the core lattice at high overload with increasing load cycle; this, in turn, leads to a strength increase of the stressed tooth. From a certain load cycle to failure of the overload, the fatigue limit begins to drop off again until it reaches the original value (beginning of damage according to French) and continues to drop down to zero (sample failure).

That the state of affairs occurring in the pulsator test cannot also be observed in the dynamic test is certainly attributable to the change in load distribution of the teeth after cracking of the dedendum. Although the load amplitude in the pulsator is constant after sample cracking, the tooth undergoes a release after cracking in the dynamic test. This release is due to the fact that the gear spring constant decreases with increasing crack length, and because the gear undergoes a plastic deformation after crack initiation and is bending more (increase in degree of overlap and angle of mesh). So, for example, the permanent elongation of a gear in the pulsator in the direction of the applied load on notched gear B after the first shock (ΔP = 2000 kp) was ca. 15 μm; after the tenth shock — about 25 μm. For unnotched gear A at ΔP = 1800 kp after the first shock, elongation was 8.5 μm and, after the tenth shock, 15 μm constant elongation was measured.

The release of a cracked tooth in dynamic testing lasts probably until the neighboring teeth exhibit cracks. Only then is the first crack-damaged tooth subject to increased load again, whereby quick failure of the transmission results.

These thoughts were confirmed by observations of the damage diagram of the running gears that failed due to broken teeth. On these gears, 3 - 4 teeth were broken in sequence; the break surface of the first (counted in the direction of rotation) tooth had the largest percentage of fatigue cracking, whereas the following teeth had increasing degrees of mechanical brake surfaces.

7. Modified French Two-Stage Tests /6

Strictly speaking, the damage line according to French is only valid for a two-stage load cycle. How much the position of this line changes in the Wohler diagram when jerky overloads occur in uniform sequence over the service life shall be demonstrated by so-called modified two-stage tests. For this, the overloads are applied in the form of individual jerk bundles, whereby the initial endurance limit was checked after each overload interval by an endurance test. Figure 18 shows the test results of gear B in a probability diagram for an overload in intervals of 500 shocks (load amplitude = 1450 kp). The hollow circles represent the jerk load-cycles of the individual tests for which no decrease in endurance limit as found, and the solid circles represent the load cycles where the gear samples were no longer found to be durable.

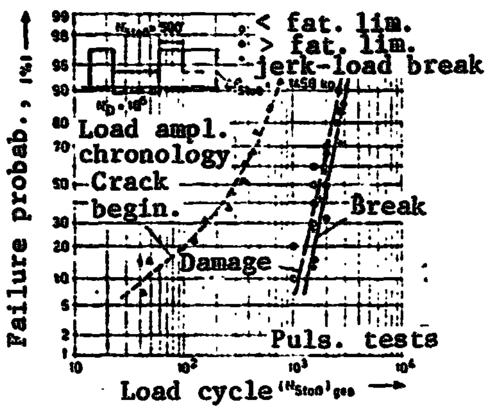

Figure 18. Crack beginning and damage to gear B for overloads in intervals of 500 shocks

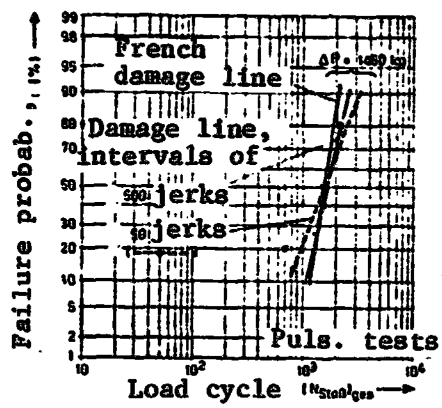

Figure 19. Influence of load-bundling on the position of the damage line of gear B

Under otherwise equal conditions, additional tests with bundling of 50 jerks were run. The test results are summarized in Figure 19. We see that the position of the damage line (for 50% failure) was not changed in the examined load range by the jerk bundling.

[The load test I' run on the same toothing in a dynamic test — the two-stage test according to French was preceded by an endurance run, see Table 4 — resulted in minor shifting of the damage line of ca. 30% (overload torque: M = 38.6 kpm) in the direction of greater load cycles.]

A similar test series with a use-hardened gear A (without notch) led to the same result. Another result was obtained for the gas-nitrided gear version E, for which the decrease in endurance

endurance limit in the modified two-stage test occurred much earlier than in the French two-stage test.

At the amplitude level used in the tests, the number of load cycles endured without any drop in fatigue limit from bundles of 50 jerks was reduced to ca. 25% (Figure 20). Thus the damage line attains an even flatter curve, and it drops near the crack line. For this reason, it is recommended in practice that for nitrided gears the crack line be considered as a determinative damage line.

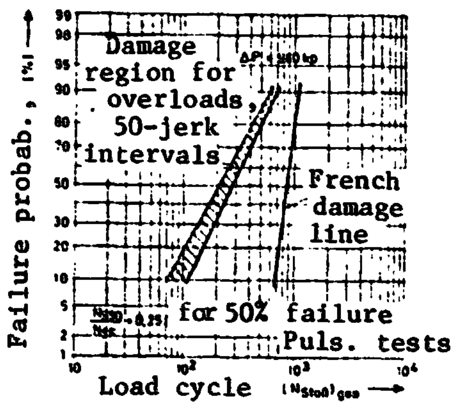

Figure 20. The influence of jerk bundling on the position of the damage line of gear E

8. Summary

The influence of jerky loads on the dedendum fatigue limit of use-hardened (material: 16 MnCr 5) and gas-nitrided (material: 31 CrMoV 9) gears with and without ground notches was studied in dynamic and pulsator tests.

The influences were determined in the form of French damage lines, since — with the exception of very low RPM (start-up processes) — the load increase time for gear mesh is significantly shorter than the load increase time of an external jerky torque.

For very slow-running transmissions, an external torque jerk can be considered as a simple overload, although one must consider that these overloads affect more than one tooth under some circumstances.

The influence of the chronology of jerks over the service life on the position of the damage line was illustrated in simplified form by jerk bundling.

The studies showed that the sensitivity of a surface-hardened gearing to short-term overloads is expressed by the position of the damage line.

The damage lines of gas-nitrided gears runs generally flatter than those of use-hardened gears. Nitrided gears can thus be exposed to smaller overloads.

Notched dedendums in the range of the 30°-tangent led to an increase in relative overloadability (related to full-load service life) in the dynamic test; however, the endurance strength itself was reduced by the ground notch.

Although the damage lines from dynamic and pulsator tests of un-notched gears agree relatively well, the damage lines of gears with notches determined in the pulsator exhibit a characteristic steep drop which was not observed in the dynamic test. The reason for this is probably found in the dynamic test run, since a tooth that has initially cracked undergoes a stress release due to elastic and plastic deformation until neighboring teeth are also cracked.

For low overloads, the crack represents the beginning of a fatigue break. Large overloads first lead to a crack and then, much later, to a reduction in endurance limit. After the beginning of cracking in the large overload range, the tooth undergoes a continuing plastic deformation which was noted especially on notched teeth (see section 6).

The time distribution of the jerky overloads over the service life had no notable influence on the position of the damage line for use-hardened gears.

For gas-nitrided gears, however, the damage line of the modified two-stage test (shock bundling) ran flatter than according to the French test. It lay in the region of the crack line for the load level examined. Therefore, in practice, it is recommended that for nitrided gears, the crack line be considered as a determinative damage line.

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