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CASE DEPTH ON FLANKS OF GEARS FOR HELICOPTER GEARBOXES.

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One of the most difficult and delicate operations during the manufacturing process of gears for helicopter gearboxes and aircraft gears is the heat treatment. Case hardened alloy steel of high quality are mainly used for aircraft gears.
Not onlv a good structure in the case hardened tooth but also the surface hardness, the core hardness and case depth are very important for the load capacity of these gears.
On the drawing and in the specifications belonging to it, values and tolerances have been provided for the above-mentioned hardnesses. These, however, are not always adequate to guarantee a good manufacture.

1. OBTAINED RESULTS AND POSSIBLE CORRECTIVE ACTIONS.

1.1. OBTAINED RESULTS.

In order to make our exposition more clear, we will illustrate it with a practical example.
Gear 33 teeth - modul 1.81(D.P.14) - 20° pressure angle - 0° helix angle.
Material chemical composition C:0.16 - Si = 0.26 - Mn = 0.56 - Cr = 1.04 - Ni = 4.39
Tolerances on the drawing concerning hardness and case depth :
effective case depth minimum 0.4 mm maximum 0.8 mm (Eht).
Surface hardness minimum 650 HV Core hardness minimum 390 HV .
At first sight these are wide tolerances which can easily be realized. But experience shows us that expecially the case depth is very often the cause of problems. Certainly to keep up the required tolerance limits along the whole toothform, tip, flank and root.
Most specifications forget to determine where the Eht has to be measured. In this case, manufacturing has to stick to the limits of the case depth, both at the tip, flank and root.

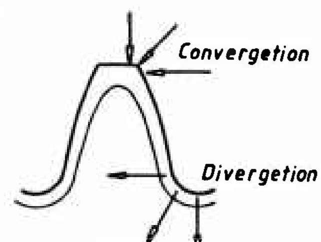


Fig 1.1

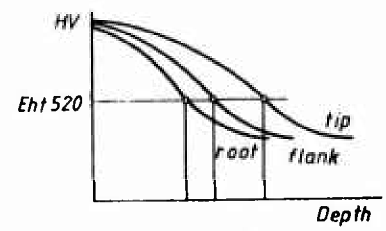
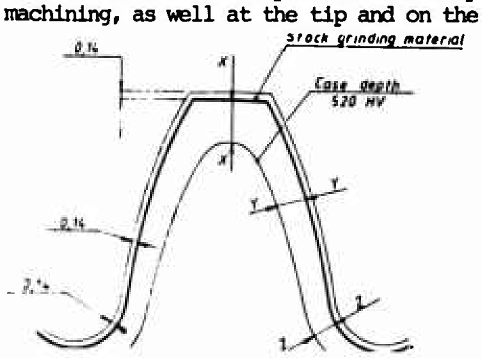
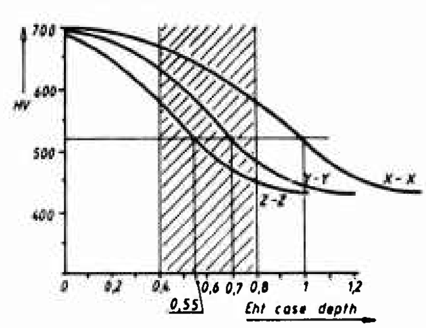


Fig 1.2

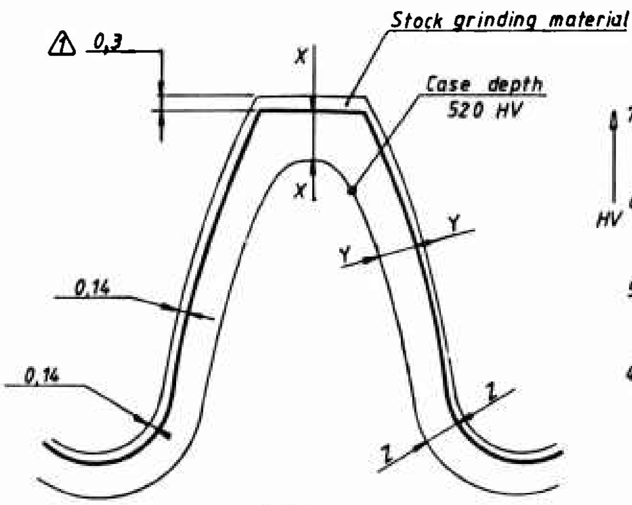
It is a well-known phenomenon that during the carburizing process the teeth of a gear take up more, but also deeper, carbon at the tip than in the root. The reason for this is the converging at the tip and the diverging in the root fillet during the penetration of gas in the carburizing process. As a result of this a carburized area at the edge arises which does not run parallel with the outer edge of the tooth form. The area is deeper at th tip and smaller in the root (see figure 1.1.)
By means of several microhardness measurements, every 0,2 mm from the edge to the core, a hardness gradient can be drawn up.
When we take the value of 520 HV as the limit hardness for the effective case depth (Eht), we obtain three different curves.
The Eht values for the measurements at the tip, flank and root radius are different. (see figure 1.2.).
We return to our practical example. The aircraft gear is ground after the heat treatment because of precision reasons.
Therefor it is necessary to foresee surplus material for grinding during premachining, as well at the tip and on the flank as in the foot.





1.2. Corrective actions.

The following manufacture-lot will now be carried out with more surplus material at the tip. In the premachining we enlarge the outside diameter so that there is a surplus grinding material of 0.3 mm at the tip. On the flank and in the root we keep the surplus grinding material of 0.14.
After the heat treatment, with the same cyclus as above, we obtain on the finished gear an effective case depth of 0.85 mm at the tip, 0.7 mm on the flank and 0.55 mm in the root (see figure 1.5. and 1.6.). We can already see an improvement here of the depth at the tip. But we still remain 0.05 above the maximum limit.
A larger surplus material at the tip during premachining could result in a far too low surface hardness at the tip of the finished part.



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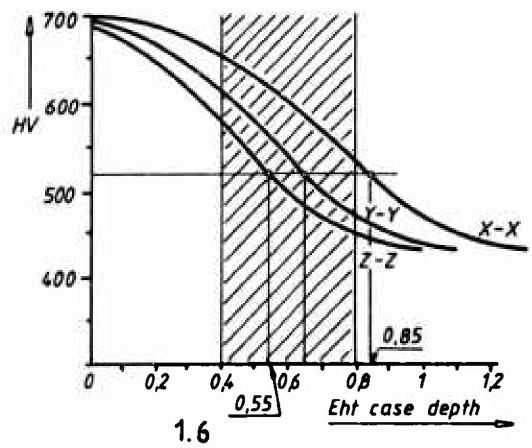
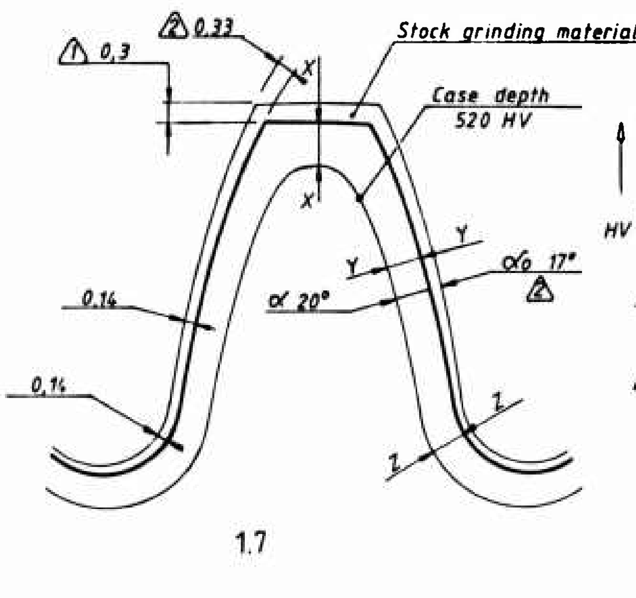


1.6

A second corrective action is applied to the following fabrication lot. Besides enlarging the outside diameter we also modify the tooth thickness at the tip. We realize this by using a hob with a special basic profile.
Namely :
- with a pressure angle of 17° (instead of 20)
- a tooth thickness in the root of the gear unmodified.
- the tooth thickness on the tip enlarged per flank to a grinding surplus value of 0.33 mm (see fig. 1.7.)

After the heat treatment and finish-grinding we obtain new results, namely
Eht tip = 0.8 flank = 0.65 root = 0.55 mm.
Now we are all over within the required toelerances. (see figure 1.7 and 1.8)
But did we manufacture a better gear now ?
Certainly not. In order to prove this we have to study more thoroughly the theory concerning the load capacity of gears.
After this (chapter 4) we will try to stipulate a better or more explicit specification about case depth and hardness.



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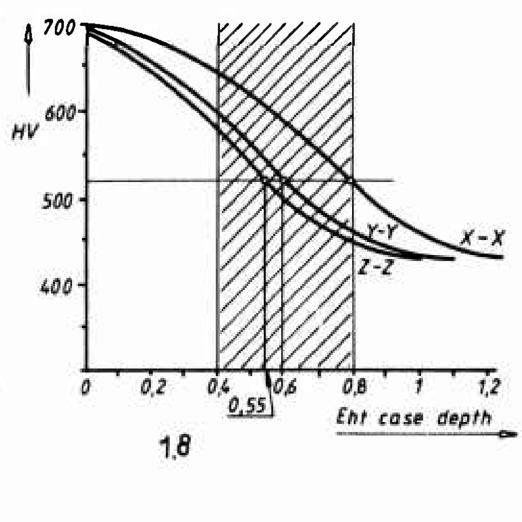


1.8

2. CALCULATION OF SURFACE DURABILITY (PITTING).

2.1. HERTZIAN PRESSURE

Pitting is small material particles breaking out of a tooth flank, leaving pits. This flank damage can be caused by the reversed stress fatigue in the contact area of underload meshing gear-teeth. It arises when the occurring contact stress exceeds the allowable contact stress and depends on the number of load cycles.
Pitting principally appears in the dedendum flank where a negative sliding speed occurs. The small pits, near the root fillet of a case hardened tooth can become the origin of a crack -possibly leading to tooth-breakage. It can also cause unacceptable vibrations and excessive dynamic overloads.
For these reasons Pitting is intolerable for aircraft gears.

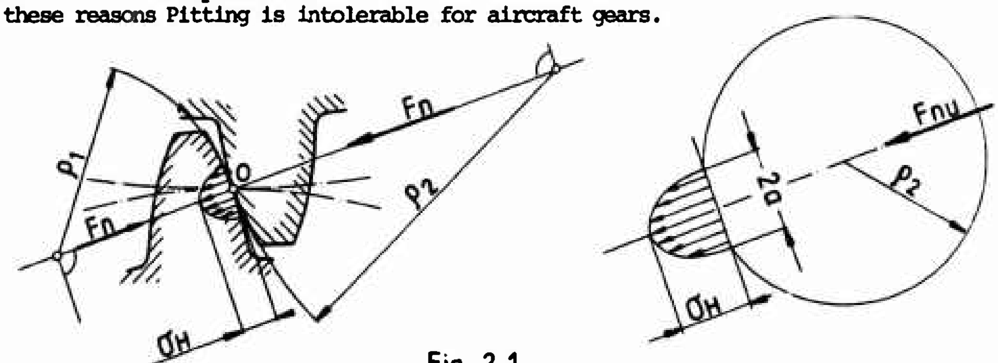


Fig 2.1

The load capacity of tooth flanks is determined according to the principle and formula of HERTZ. Therefore contact stress is sometimes called "HERTZIAN PRESSURE".
By ISO * the basic Hertzian formula is elaborated and completed with all possible factors which can effect the load capacity and surface durability. Therefore, we take this approach as startingpoint in our further explanation.
Further we only examine the endurance limit for contact stress (σ_{Hlim}) - for its involvement with the "case depth".

2.2. ISO APPROACH OF SURFACE DURABILITY (PITTING)

2.2.1. Contact stress (Hertzian pressure) at the operating pitch circle .

$$\sigma_H = \sigma_{HO} \sqrt{K_A \cdot K_V \cdot K_{H\alpha} \cdot K_{H\beta}} \leqq \sigma_{HP} \qquad \text{Wherein :} \qquad (1)$$

- σ_H - Basic value of contact stress
- K_A - Application factor
- $K_{H\beta}$ - Longitudinal load distribution factor for contact stress.
- $K_{H\alpha}$ - Transverse load distribution factor for contact stress.
- σ_{HO} - Allowable contact stress (permissible Hertzian pressure).
- K_V - Dynamic factor.

$$\sigma_{HO} = Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \sqrt{\frac{F_t}{d_1 \cdot b} \cdot \frac{u+1}{u}} \qquad \text{Wherein} \qquad (2)$$

Z_H	- Zone factor	F_t	- Nominal tangential load
Z_β	- Helix angle factor	Z_ε	- Contact ratio factor
d_1	- Reference diameter of pinion	b	- Facewidth
Z_E	- Elasticity factor	u	- Gear ratio Z_2 / Z_1

* The calculation of load capacity of spur and helical gears.
(I.S.O. / DP6336)

2.2.2. Allowable contact stress (Permissible Hertzian pressure)

$$\sigma_{HP} = \frac{\sigma_{Hlim} \cdot Z_N}{S_{Hmin}} \cdot Z_L \cdot Z_R \cdot Z_V \cdot Z_W \cdot Z_X \qquad \text{Wherein :} \qquad (3)$$

σ_{Hlim} - Endurance limit for contact stress
S_{Hmin} - Minimum demanded safety factor for contact stress.
Z_L - Lubricant factor
Z_W - Work hardening factor
Z_X - Size factor for contact stress.
Z_R - Roughness factor
Z_V - Speed factor
Z_N - life factor for contact stress.

2.2.3. Safety factor for contact stress (against pitting)

$$S_H = \frac{\sigma_{Hlim} \cdot Z_N}{\sigma_{HO}} \cdot \frac{Z_L \cdot Z_R \cdot Z_V \cdot Z_W \cdot Z_X}{K_A \cdot K_V \cdot K_{H\alpha} \cdot K_{H\beta}} \qquad (4)$$

The factors were named above

2.3. ENDURANCE LIMIT FOR CONTACT STRESS.

The endurance limit for contact stress can be regarded as the level of Hertzian stress wich a material will endure without damage for at least 50.10^6 loadcycles. Testing discs in disc machines gives an indication of trends of relative values of endurance limit for contact stress. These values can also be established on the basis of data from gears in service.
The fig, alongside can be used as a guidance for surface hardened steels - when the requisite data are not available.
The values correspond to a failure probability of 1%. The endurance limits for Hertzian pressure shown in this diagram are valid for a mean surface roughness Rtm = 3µm (Z_R = 1) a tangential speed V = 10m/s (Z_V = 1) and an oil viscosity ν = 100 mm^2/S (Z_L = 1).
When we look at the graphics zone concerning case hardened alloy steel, principally used for aircraft gears, this zone contains a very wide range for Hertzian pressure endurance values, namely from 1300 to 1650 N/mm^2.
For high quality gears used in aircraft, we must aim at the optimum. This optimum is mainly influenced by :

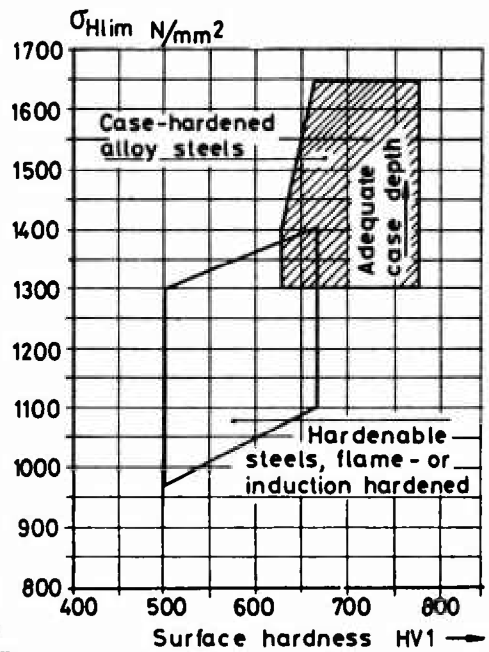


Fig 2.2

1. Material composition
2. Mechanical properties.
3. Hardening process, depth of hardened zone, hardness gradient.
4. Structure (forging-rolled bar-cast).
5. Residual stresses.
6. Material cleanline and defect.
   We will concentrate on point 3.
   According to the I.S.O.-diagram, it is necessary to have a surface hardness of 670-775 H_{V1}, in order to reach a maximum endurance limit for contact stress (1650 N/mm). Nearly the whole gearliterature agrees on these values. But what about the case depth with regard to the limit for Hertzian pressure ? Only an "adequate case depth" is stipulated by I.S.O. We have examined this element more closely.

(* by I.S.O. / DP 6336 III).

2.4. ADEQUATE CASE DEPTH. FOR CASE HARDENED ALLOY STEEL.

2.4.1. Shear stress in the subsurface layer.

Not only the Hertzian pressure, but also an appearing micro-stress in the subsurface layer is the cause of pitting. Hertz already knew this - but up to now there is still no agreement about this shear stress.

Two cylinders represent two meshing gear teeth with respective curvilined radius of ρ_1 and ρ_2. When they are pressed together with a load F_n, then Hertzian pressure and a flat surface occurs in the contact area.
This surface has a value of "2a" (see fig. 2.1.

Furter, we refer to G. Henriot *1 and the most current theory about the origin of pitting.
The total flattening, the amount of the flattenings of both gear teeth, on the radius is : u = 0,0005 Fnu with : (5)
Fnu = Nominal tangential force (in Newton) or the reference cylinder, in a transversed section on the facewidth of 1mm.
The length of the flat surface 2a is :

$$2a = 0{,}063\sqrt{Fnu \cdot \rho r} \qquad \text{wherein : (6)}$$

ρr = relative profile radius and :

$$\frac{1}{\rho} = \frac{1}{\rho_1} \cdot \frac{1}{\rho_2} \qquad (7)$$

The figure alongside illustrates the different pressures in the subsurface, down to the core for cylinder ρ_2, following stresses can be distinguished :

σ_z = with a direction axis 0-Z

σ_y = with a direction axis 0-Y

σ_x = with a direction axis 0-X

σ_y and σ_z = σ_H on the surface

σc, the shear stress, is the result of two squared stresses. This shear stress is at its maximum under an angle of 45°, with value which is half the difference from the normal stresses.
The shear stress is zero at the surface and comes to its maximum on the "depth of maximum shear stress" ≈ 0,8 a (8)

$$\sigma c \approx 0{,}3\ \sigma_H \qquad (9)$$

$$\approx 59\sqrt{Fnu \cdot \frac{1}{Pr}} \qquad (10)$$

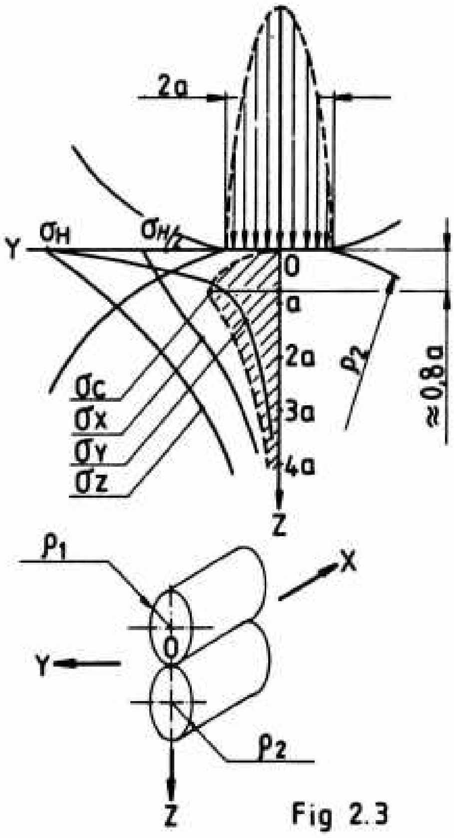


Fig 2.3

This shear stress is very important because it is the most important cause of pitting. For case hardened alloy steel, we must obtain a hardened layer (eff. case depth) of at least twice the depth of maximum shear stress.
Most publication stipulate that σc may not exceed the Yield point of the used steel. But, the determinationsof the Yield point in a hardened layer of case hardened alloy steel is practically impossible !

Klaus Bornecke *2 has made a study about the heat treatment of case hardened cylindrical gears and their load capacity. We have made a summary with the most important particulars.
In this exposition, the shear stress is a standard stress, which can be defined by three hypothesises :

1. Main shear stress Vsub
2. Theorem of the minimum elastic energy VG
3. Reversed shear stress VW

The comparison of these three stresses is represented in fig. 2.4.

*1 : Georges Henriot : Traite theorique et pratique des engrenages 1. Chapitre VII

*2 : Klaus Bornecke : Beanspruchungsgerechte Wärmebehandlung von einsatzgehärten Zylinderrädern.

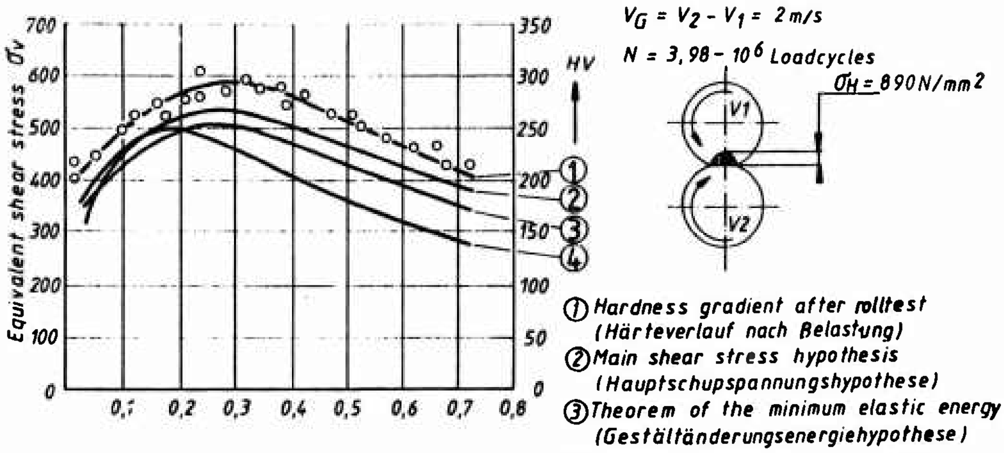


Fig 2.4

Further experiments have shown that theorem of the minimum elastic energy σvg is the most real approach for the shear stress.
As already said before, the yield point defines the limit stress of alloy steel gear. The figure 2.5.1., illustrates the evolution of shear stress and the depth of maximum shear stress when the Hertzian pressure increases. As you can see, they both enlarge respectively from $\sigma vg1$ to $\sigma vg2$ and from T1 to T2. This means that the depth of maximum shear stress lies deeper under the surface. Flank damage or pitting occurs in the lined part "a" of the curve.

Trough hardened steel

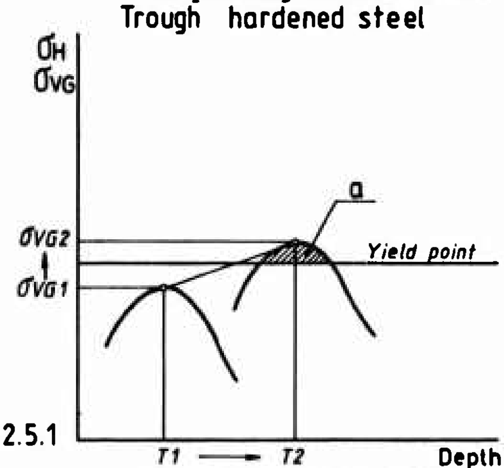


Fig 2.5.1

Case hardened alloy steel

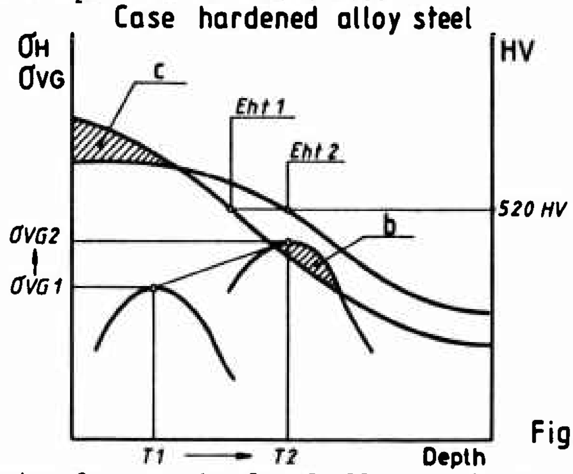


Fig 2.5.2

When we make the same exercise for case hardened alloy steel, we have to take the $\sigma 0,2$ limit as equal parameter. This because, as said before, the determination of the yield point in a hardened layer of case hardened alloy steel is practically impossible. $\sigma 0,2$ is the limit where an irreversible deformation of 0,2 % appears in the bending test. (see figure 2.7.)
It has already been proved that the curve of $\sigma 0,2$ limit and the hardness gradient have a parallel trend. For each quality of case hardened alloy steel, there is a correlation between the $\sigma 0,2$ limit and the hardness gradient. (see figure 2.6.1. and 2.6.2. an example for 16 MnCr5 - K. Bornicke).

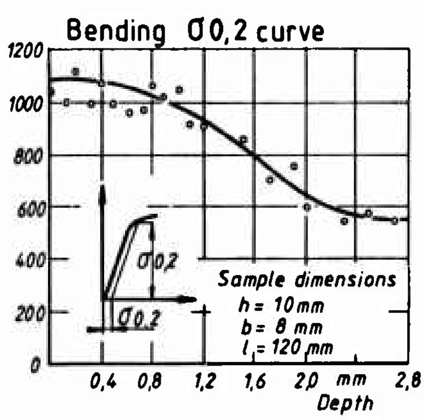


Fig 2.6.1

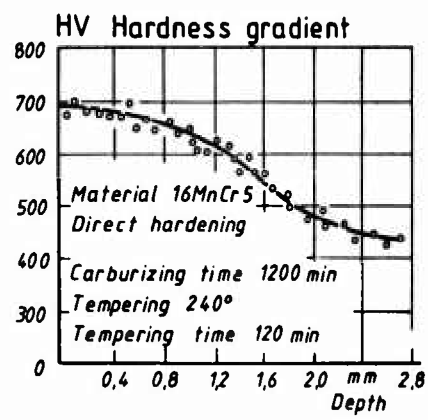


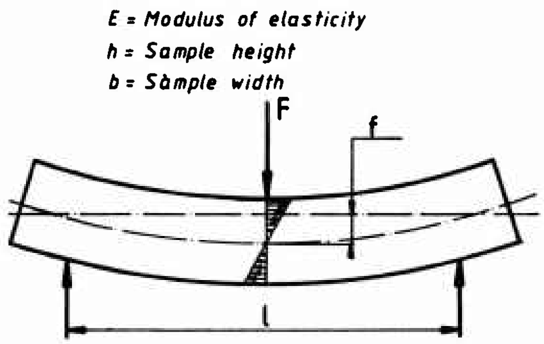
Fig 2.6.2

So, the hardness gradient can be seen as the limit of allowable shear stress. When we study this line, we see that "b" is the zone where pitting appears for case hardened alloy steel. (see figure 2.5.b.). There are three conclusions to be made when we study figure 2.5.2. :

1e : The surface hardness is less important with respect to pitting. This is the reason why we can take a bigger tolerance, namely (zone "c") 670-775 HV1.

2e : The effective case depth must be larger than the "depth of maximum shear stress". A bigger hardened layer on the flanks can never be harmful or decrease the strength against Hertzian pressure.

3e : A higher core-hardness can also improve the strength against Hertzian pressure. This because the hardness gradient becomes flatter. As you can see in figure 2.5.b., the curve "2" is flatter than curve "1" and has a higher limitvalue against shear stress.



$$\sigma = \frac{f.6.E.h}{l^2}$$

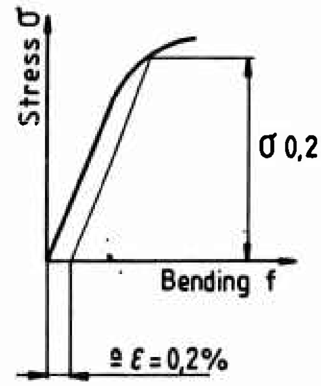


Fig 2.7

3. CALCULATION OF TOOTH STREGTH

3.1. TOOTH BREAKAGE and TENSILE STRESS at the TOOTH ROOT.

In the first place, gear-teeth must be resistant against tooth breakage. This means that there must be enough resistance, in the root radius against the occurring tensile stress.
The calculation of this resistance is based on those used for a steel beam, loaded on its free end with a force F.
In the clamped area, a tensile stress σ_F arises (figure 3.1.1.).
A gear tooth is different from a square beam in form and function.
The tooth load in case of single engagement, this means that only one flank pair is in contact, is shown in figure 3.1.2.
The tooth form factor Y_F is the most important parameter in the calculation of tooth strength. This factor depends on the involute form, the value of the root radius, the pressure angle and the pressure angle of the highest point for single contact.
As for Hertzian pressure - the tooth breakage has been studied by I.S.O. (*) We have accepted method-B and a copy of the formula is given as information. Further, we will concentrate on the case depth in the tooth root radius and its relation to the bending-endurance limit σ_{Flim}.

(I.S.O. / DP 6336)

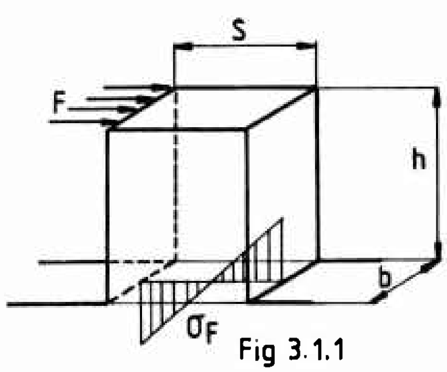


Fig 3.1.1

$$Y_F = \frac{6(h_F/m_n)\cos\alpha_{en}}{(S_{Fn}/m_n)^2\cos\alpha_n}$$

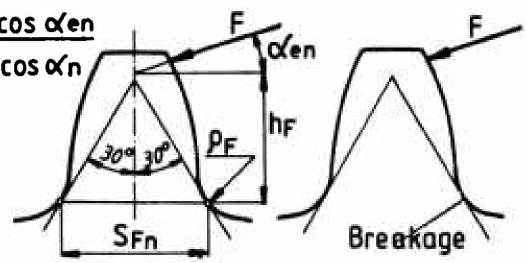


Fig 3.1.2

3.2. METHOD-B of the I.S.O. APPROACH of TOOTH BREAKAGE.

Hereby it is assumed that the highest tooth root stress arises by applying a force at the outer point of single tooth pair contact.

3.2.1. Tooth root stress.

$$- \sigma_F = \sigma_{FO} \cdot K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \leqq \sigma_{FP} \quad \text{Wherein :} \quad (11)$$

σ_{FO} : the local tooth root stress defined as the maximum stress at the tooth-root when loading a flawless gear by the static nominal moment.

K_A : application factor

K_V : dynamic factor

$K_{F\beta}$: longitudinal load distribution factor for tooth-root-stress

$K_{F\alpha}$: transverse load distribution factor for tooth-root-stress.

$$- \sigma_{FO-B} = \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \quad \text{Wherein :} \quad (12)$$

F_t : Nominal tangential load.

b : Facewidth.

m_n : Module, normal section

Y_F : Tooth form. factor.

Y_S : Stress correction factor.

Y_β :Helix angle factor.

3.2.2. Permissible tooth root stress.

$$- \sigma_{FP} = \frac{\sigma_{F\,lim.} \cdot Y_{ST} \cdot Y_{NT}}{S_{F\,min.}} \cdot Y_{\sigma rel\tau} \cdot Y_{R\,rel\,\tau} \cdot Y_X \quad \text{Wherein :} \quad (13)$$

$\sigma_{F\,lim.}$: Nominal bending endurance limit.

Y_{ST} : Stress correction factor, for testgear dimensions.

Y_{NT} : Life factor for tooth-root-stress related to test gear dimension.

$S_{F\,lim.}$: Minimum safety factor.

$Y_{\sigma rel\,T}$: Relative sensitivity factor, related to the test gear dimension (takes into account the notch sensitivity).

$Y_{R\,rel\,T}$: Relative surface condition factor.

Y_X : Size factor for tooth root strength.

3.2.3. Arithmetic safety factor for tooth root stress.

On the basis of the strength determined at a test gear. The factors were named above.

$$- S_F = \frac{\sigma_{F\,lim.} \cdot Y_{ST} \cdot Y_{NT}}{\sigma_{FO-B}} \cdot \frac{Y_{\sigma rel\,T} \cdot Y_{R\,rel\,T} \cdot Y_X}{K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha}} \quad (14)$$

3.3. ENDURANCE LIMITS FOR BENDING STRESS.

The nominal bending endurance limit takes into account the influence of the material on the tooth root stress witch can be permanently endured. A running time of 3.10^6 cycles is regarded as the beginning of endurance limits.
The limit can be found by pulsating tests or gear running tests for any material and any state of that material. Limiting values, obtained by field of experience, can also be used.
If such data are not available, guide values can be determined with the help of the fields in figure 3.2.. The here indicated values for nominal bending endurrance limits apply to the following gear dimensions at service conditions :

- Module m = 3 up to 5 mm Helix angle = 0°
- Roughness in the tooth root R_{tm} 10µm
- Basic rack according to I.S.O. 53-1974
- Stress correction factor Y_{ST}= 2.1
- Linear speed v = 10 m/s
- Facewidth b = 10 up to 50 mm.

The values included in the diagrams correspond to a failure probability of 1 %.

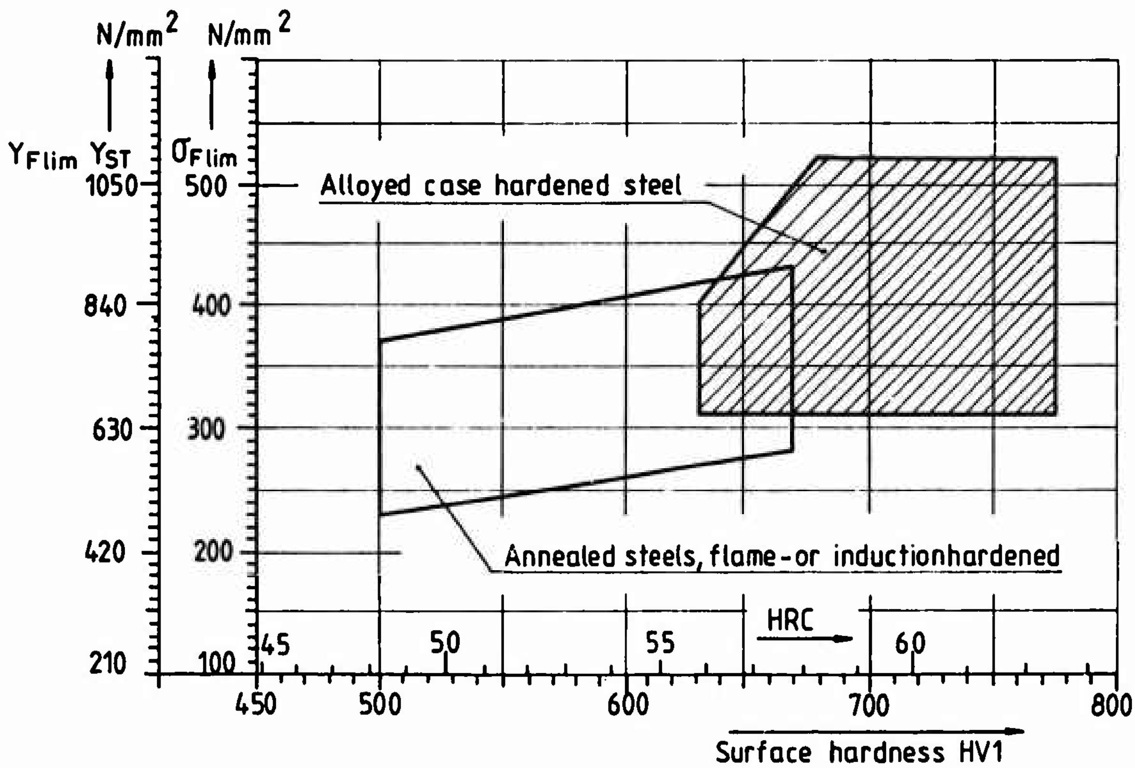


Fig 3.2

The bending endurance limit for case hardened alloy steel, shown in figure 3.2. has a very wide range. Values for $\sigma_{F\ lim.} \cdot Y_{ST}$ go from 650 N/mm2 to 1.300 N/mm2. You can find more specific values in some gear literature, such as in the German book "Machinen-elemente G. Niemann H. Winter", from which we take the following values for "bending endurance for case hardened steel (according to D.I.N. 1 7210).

DIN 1 7210 Quality	Core hardness HV_{10}	Surface hardness HV_1	Bending endurance limit Flim. . Y_{ST}	static
16 MnCr5	270	720	860 N/mm^2	2150 N/mm^2
15 CrNi6	310	730	920	2300
17 CrNiMo6	400	740	1000	2800

Gears as used for aircraft are of high quality steel, for example 17CrNiMo6 and even the higher classed 14CrNi18.
For the bending stress limit we have the same main influencing factors to aim at the optimum as for the contact stress limit (see text 2.3.).

I.S.O. only mentions that the endurance limit for case hardened alloy steel is applicable to an effective case depth of at least 0,15 module on a finished part. This means, for aircraft quality, after grinding the flanks and the root-radius.
According to K. Bornicke - the resistance against breakage in the tooth-root with case hardened steel depends on the effective case depth in the root-radius - at 30° tangent (figure 3.1.2.).

3.4. ADEQUATE CASE DEPTH FOR TOOTH BREAKAGE.

The figure below 3.3.1. shows very clearly the endurance limit value in function to the case depth. The optimum limit for bending endurance occurs when we have a case hardened layer 0,1 modul. A smaller layer decreases the limit quickly. A bigger layer decreases the limit slowly till 0,3 modul and from then on, the limit decreases more quickly. The core hardness at the rooth-cylinder, measured at the tooth-center, is also very important for the bending stress limit value.
Bornicke and also G. Niemann-H. Winter both agree on the case depth and the core hardness to obtain a maximum resistance. Values of these are printed in figure 3.3.2.

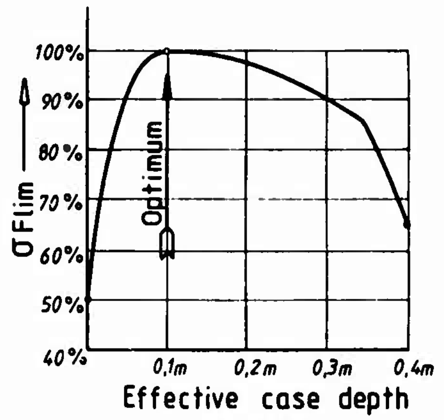


Fig 3.3.1

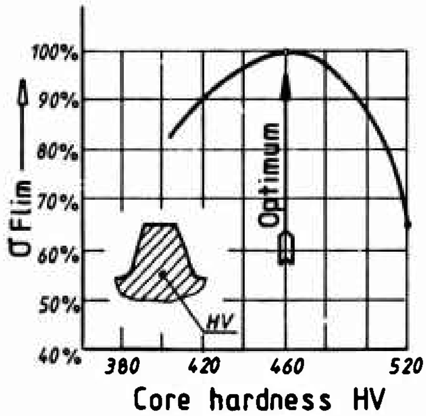


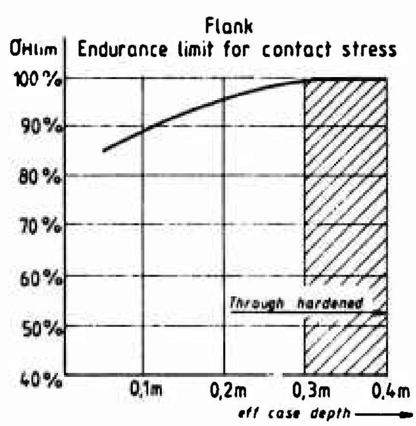
Fig 3.3.2

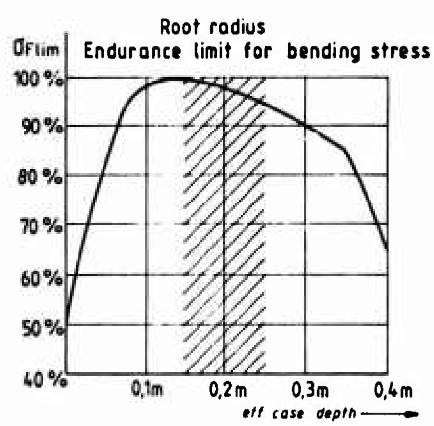
Again, three important conclusions are to be made :

1. A surface hardness of 670-755H_{V1} guarantees a maximum resistance against tooth breakage.
2. The core hardness is an important parameter for the strength resistance of gear-teeth. For high quality alloy steel a core hardness of 400-460 H_{V10} must be obtained.
3. The effective case depth - measured in the root radius at 30° tangent - is optimum 0,1 modul - but may not exceed 0,3 m. A tolerance of 0,15 - 0,25 modul is usually applicable.

4. PROPOSAL for SPECIFICATION of CASE DEPTH - SURFACE and CORE HARDNESS on CASE HARDENED STEEL GEARS.

When we put the conlclusions about the endurance limit for Hertzian stress and tooth-root-stress together - we see that a different adequate case depth is asked.





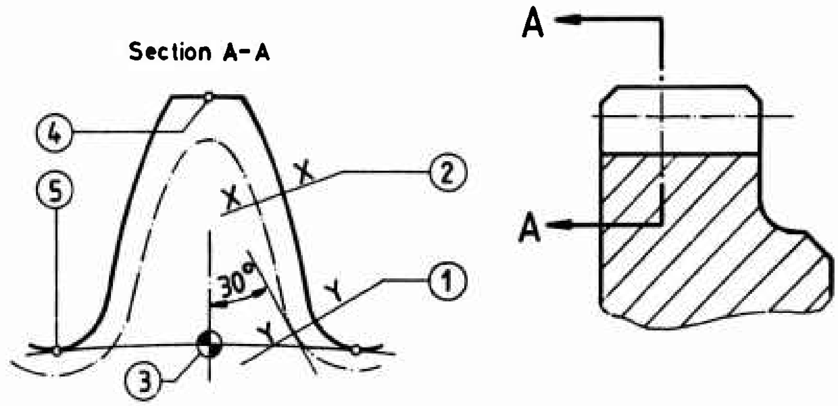
Namely : 1. On the flanks a minimum effective case depth larger than the "depth of maximum shear stress"is asked but no maximum because this maximum does not decrease the strength against Hertzian pressure.
As you can see in figure 4.1.1. we reach an optimum at 0.3 modul.

2. On the 30 tangent in the root-radius - a minimum effective case depth of 0.1 modul and a maximum of 0.3 modul is asked because both a bigger and a smaller layer decrease the strength against bending stress (figure 4.1.2.).

So, for example - taken in 1.1. - we can recommend the following specification.

INSPECTION METHOD CONTROL

HARDNESS DETERMINATION



1	EFFECTIVE CASE DEPTH X-X	MIN.0,55 mm	MAX —
2	EFFECTIVE CASE DEPTH Y-Y	MIN. 0,27 mm	MAX. 0,45 mm
3	TOOTH CORE HARDNESS	MIN. 420 HV	MAX. 480 HV
4	SURFACE HARDNESS	MIN. 670 HV	MAX. 775 HV
5	SURFACE HARDNESS	MIN 670 HV	MAX. 775 HV

NOTES:

1. EFFECTIVE CASE DEPTH IS THE LENGHT OF LINE X-X OR LINE Y-Y MEASURED FROM THE SURFACE TO THE LAST POINT TOWARD THE CORE HAVING THE REQUIRED MINIMUM HARDNESS VICKERS 520 HV
2. LINE X-X IS LOCATED ON THE WORKING PITCH CIRCLE, SQUARE TO THE TOOTH FLANK
3. LINE Y-Y IS LOCATED AT SQUARE ON THE 30° TANGENT TO TOOTH ROOT RADIUS.
4. THE TOOTH CORE HARDNESS IS TO MAKE IN THE CENTER OF THE TOOTH TO THE ROOT CIRCLE. (±1mm)

REFERENCES : 1. Beanspruchungsgerechte Wärmebehandlung von einsatzgehärteten Zylinderrädern
K. Börnicke 1976

2. Carburizing and carbonitriding.
A.S.M. 1977

3. Maschinen-elemente band II
G. Niemann H. Winter 2e Edition

4. Tandwielen
Prof. R. Snoeys Ir. R. Gobin 1979

5. Traité theorique et pratique des engrenages 1
G. Henriot 6e edition

6. I.S.O. DP 6336
The calculation of load capacity of spur and helical gears.

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3. Mrs. Hilde Watteeuw
M.C. Watteeuw - N.V. - Brugge - Quality Control Laboratory

DISCUSSION

A.Borrien, Fr

The definition of case depth that you have given corresponds to the depth at which the hardness is 520HV. Does this value come from the normal specifications or from particular standards?

Author's Reply

The hardness of 520HV comes from the specifications of the client, the builder. Personally, I prefer for high quality case hardened materials (1% chrome, 4.4% nickel) a limit of 550HV for the effective case depth.

B.A.Shotter, UK

The analysis of this problem is even more complex than the author has suggested. The root stress fluctuations which are experienced by planet or idler gears are significantly different to those of unidirectionally loaded teeth. In the case of contact stresses one has to be careful as to the surface fatigue initiation mode: many examples of surface breakdown start as micropitting. In this case, the origin of the failure is much smaller than the Hertzian contact width. The propagation of this damage is highly dependent upon the stress state of the surface layers. Thus, whilst the authors' approach is considered to be an excellent starting point, even more factors have to be considered to make full appraisal of the required case definition.

Author's Reply

I agree with the point of view of Mr Shotter.

But, nevertheless, it cannot be contested that the effective case depth on the flanks and in the root radius have a different optimum value for endurance limit for contact stress and bending stress.

The actual information on the drawings and in the specification is often inadequate.