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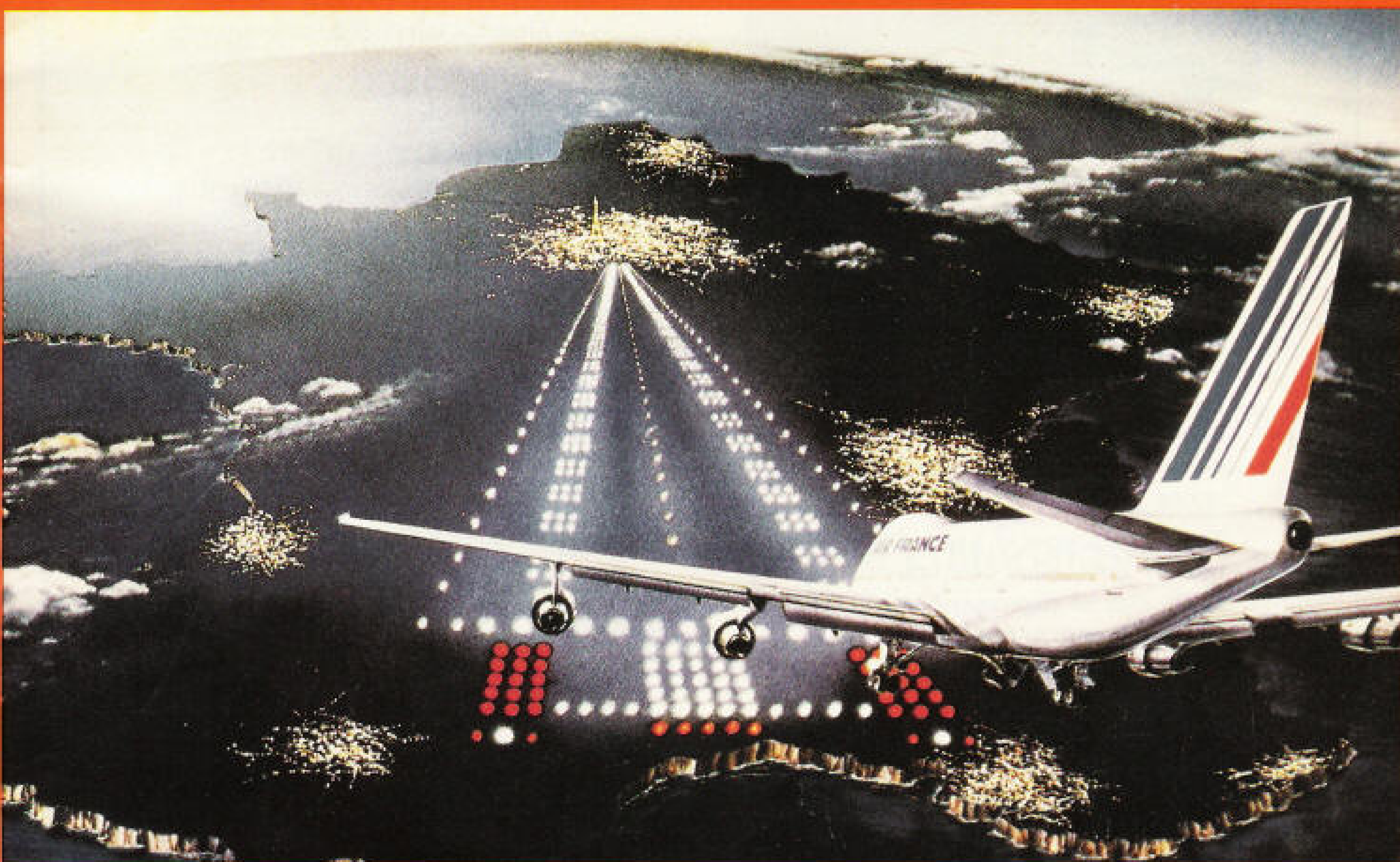

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« fail-safe » concept

and choice of a « limiting condition » indicator

LUBRICATION UNIT FOR THE TURBOFAN JET ENGINE *CFM 56-2*

determination of the mechanical properties of the shear groove in a rotary shaft by the stress gradient method

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The very interesting article by M. P. Rochus *is naturally intended primarily for engineers who are conversant with aeronautical problems. In our opinion, it should also be of interest to those working in the field of marine structures, for which the methodologies and modern procedures for project design and analysis, based, in fact, on reliability concepts, have an increasing importance.* M. P. Rochus *describes, in fact, an application of the « fail-safe » concept which consists of choosing a limiting condition which can easily be detected and which, while not resulting in any serious damage, can thus constitute an indicator of the condition of the system. Furthermore, the problem which is examined and resolved by the stress gradient method is entirely structural.*

« Mechanical fuse » for abnormal loadings

The lubrication unit for the turbofan jet engine *CFM 56-2* consists essentially of a set of *five* pumps of the rotary type, mounted on a single shaft : one *supply* pump fed with pure oil by pipeline from the reservoir, and four *scavenge* pumps drawing in an oil-air mixture from the front and rear compartments, the gear box and the accessory mounting unit (fig. 1).

The rotary shaft has a *shear groove* which is *externally visible* on the lubrication unit. This shear groove (« mechanical fuse ») must satisfy several requirements :

— in the event of an *abnormally high reactive torque* being encountered, the shear groove *must fail* before any other component in the lubrication unit, to prevent obstruction of the compartments;

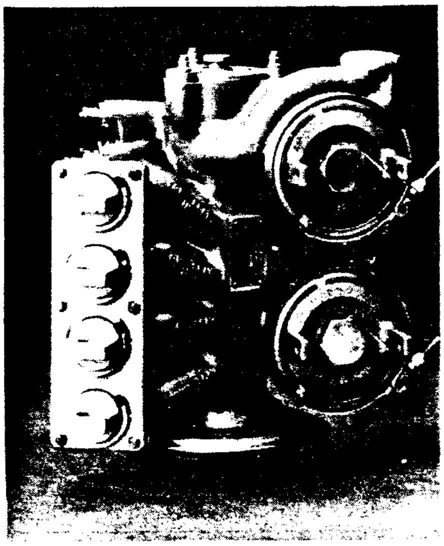
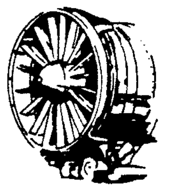


Fig. 1. — Lubrication unit.



- the failure of the shaft is necessary, so as to *avoid damage to the remainder of the transmission;*
- the failure of the shaft, which is *visible from the outside* of the lubrication unit, must prevent the refitment of a damaged unit;
- the presence of the shear groove must not abnormally *reduce* the *fatigue life* (expressed in cycles) of the shaft under normal operating conditions.

The various tests carried out in order to define the extreme torques in « normal » operation are briefly described. The minimum failure torque for the shaft must be *greater* than the *normal* reactive torques. On the other hand, the maximum static *failure* torque for the shaft must be less than the *minimum* failure torques for the *other moving components* in the lubrication unit. In order to determine these static failure torques accurately, taking into account the properties of *plastic adaptation* of the steel at the *bottom of the notch,* the *gradient method* [1] (1) was used, and its results compared with experimental data. The gradient method was then applied to the investigation into the fatigue strength of the shaft, with the object of showing that the life, expressed in cycles, remains within acceptable limits.

Experimental determination of the maximum reactive torques encountered in « normal » operation

For the experimental determination of the *extreme conditions in normal operation,* different tests were carried out. These tests consisted of :

- operating tests in *cold conditions* (— 50 °*C* and — 40 °*C* : simulation of an engine *start* at low temperature, simulation of a *climb to height* followed by cruising flight), and in *hot conditions* (120°*C* for the inlet oil and 160°*C* for the scavenged oil : simulation of flight in *a holding pattern*);
- operating tests with contaminated oil, in turn with Arizona sand, aluminium oxide, steel filings and chips;
- engine *start* tests without oil;
- *cyclic endurance* testing.

In the majority of these tests, the rotary torque was less than 10 *Nm,* the only exceptions being the following :

- the passage of an *adversely-situated large particle,* which becomes trapped between the pump and its end plate, can cause an excess torque, which remains below 10 *Nm;*
- during an engine *start* at — 40 °*C,* the drive torque rose to a maximum of 53.5 *Nm* after 1 *second* of rotation; it fell to 12 *Nm* after 16 *seconds,* and after one *minute* the torque was only 9.5 *Nm.*

In conclusion, the *static failure torque* must be *greater* than 53.5 *Nm.* A sinusoidal cycle, composed of an *alternating torque* of 10 *Nm* superimposed on a *constant torque* of 10 *Nm,* will serve as the reference cycle (more severe than the true cycle) for the *fatigue* investigation in normal operation. A sinusoidal cycle, composed of an *alternating torque* of 25 *Nm* superimposed on a *constant torque* of 30 *Nm,* will also allow the effect of *engine starts* in very cold conditions (— 40 °*C*) to be simulated.

Theoretical and experimental determination of the static torque for the *CFM 56-2* shaft

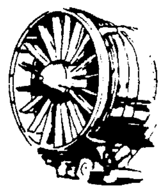
stress concentration and plastic adaptation

The shear groove has the following dimensions (fig. 2) :

φ_P = 10.6 to 11.1 *mm;* φ_M = 8.45 to 8.55 *mm.*

The groove is machined with a surface finish of 1.6 *μm,* and the shaft material is *12 NC 12,* with a *Brinell* hardness of 277 to 352.

(1) The numbers in square brackets denote the references listed at the end of the article.



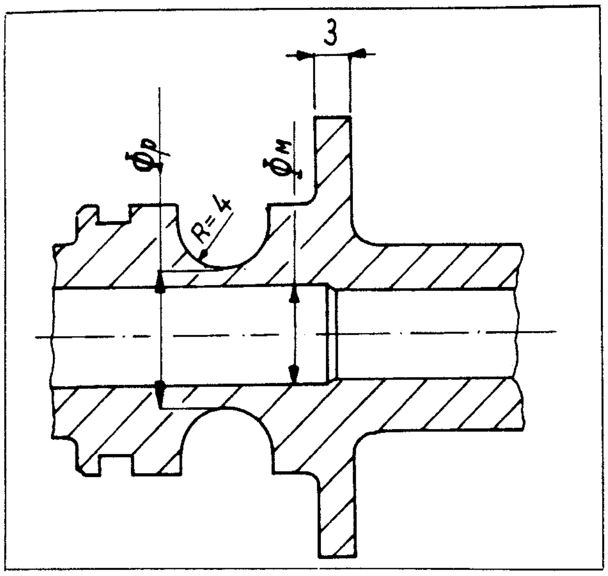


Fig. 2. — Shear groove.

For the determination of the failure torque with stress concentrations present, the assumptions of a *linear elastic* mechanism are *no longer valid.* Calculations based on the theory of a stress concentration factor [2] obtained from elasticity theory [3,4] are rarely in close agreement with experimental results. The results of such calculations are *too conservative,* due to the effects of *plastic deformation,* which are not taken into account and which cause a redistribution of the local stresses. In the case we are considering, the shaft must fail within a *very precise range* of values (with no safety factor). It is therefore necessary to take into account the concepts of the *elasto-plastic failure mechanism* [5,6,7,8,9,10]. This study of the *static* failure and of the *dynamic* failure is based on the *gradient method* [1] initially advanced in reference [11] and developed by *CETIM.* The method used is based on an impressive number of *tests* carried out on different steels. The equations used are phenomenological, are not invariable with regard to a change in the units of measurement, and are valid *for a certain range of stress gradients only.* These equations must therefore be adapted to a *supporting microscopic theory,* if one exists. On the other hand, the method is *very simple* to use and shows *excellent agreement with experimental results.* In this connection, the appendix gives the *results* of torsional failure tests on steel bars, both of constant section and with a groove, all with the same minimum cross-section.

As with the gradient method, the tests result in a *higher failure torque for the bar with a groove, contrary to what is predicted by the classical theory.* The gradient method allows for the true *optimisation* of the parts, taking into account the properties of *plastic adaptation at the bottom of the notch* in metals, when high static and dynamic stress concentrations are present, and when different loadings are superimposed.

stress gradient method

The *von Mises* criterion is altered by the stress gradient theory :

$$\frac{C_{r,s}}{I_p/r} = \frac{R_m}{\sqrt{3}}$$

where :

$C_{r,s}$ is the static failure torque,
I_p is the polar moment of inertia,
r is the radius,
R_m is the ultimate stress,

replacing R_m by $\hat{R}_m$, given by :

$$\hat{R}_m = R_m\,(0.25 \log_{10} \chi + 1.4)$$

The parameter χ must be expressed in mm^{-1} and represents the *stress gradient* :

$$\chi\,(\text{mm}^{-1}) = \left(\frac{1}{\sigma}\frac{d\sigma}{dx}\right) \text{max.}$$

$$= \left(\frac{d}{dx} \ln \sigma\right) \text{max.}$$

σ represents the local stress at the notch.

In the case of torsion, we have :

$$\chi = \frac{1}{r} + \frac{2}{\varphi_P}$$

type of test specimen	hardness *HB*	R_m *hbar*	φ_M *mm*	φ_P *mm*	χ mm^{-1}	$C_{r,s}$ calculated *Nm*	$C_{r,s}$ experimental *Nm*
1	352	120	8.45	11.1	0.43	158	147
2	352	120	8.55	10.6	0.44	120	107
3	277	94	8.45	11.1	0.43	124	139
4	277	94	8.55	10.6	0.44	94	98

Table I. — Static failure torque, calculated and measured.

And the static failure torque is then :

$$C_{r,s} = \frac{I_p R_m}{r\sqrt{3}} (1.4 + 0.25 \log_{10} \chi)$$

This relationship takes the stress concentration factor into account. Table I compares the results obtained with it to those from tests carried out on test specimens representing the limiting cases (specimen No. 1 : maximum material, maximum hardness; No. 2 : minimum material, maximum hardness; No. 3 : maximum material, minimum hardness; No. 4 · minimum material, minimum hardness).

Figure 3 illustrates a shaft after failure on test.

Fig. 3. — Shaft failed on test.
(Fabrique Nationale, Herstal).

Fatigue strength at the shear groove

The *scatter* in fatigue results is today accepted, both as an *experimental fact* and as a *physical fact* [9,5]. The physical causes are essentially :

— causes *inside the material* (inclusions, non-homogeneity of the structure, etc.). Under the effect of the stresses, these defects can be displaced, or can multiply or disappear, and such defect activity will be subject to *fluctuations*;

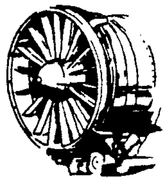
— causes stemming from the *manufacture of the parts* (milling, grinding, turning, heat treatment, etc.);

— factors *unconnected with the part itself*, such as imponderables affecting installation and the setting-up of parameters.

If the scatter in the experimental results is fairly wide, a greater number of tests must be carried out, in order to determine the *mean response* to fatigue and the *scatter*. It is therefore very useful to have a *reliable method of calculation* which allows the *number of costly tests to be reduced.* Despite numerous theoretical studies [5,6,7,8,9,10] devoted to the fatigue failure mechanism, the phenomenon is still *not very well understood,* and microscopic theories, which define mathematically the stages of a fatigue failure, i.e. crack initiation, and the propagation and rapid and unstable enlargement of cracks, are either difficult to apply, or remote from reality. The equations which link distorsions to stresses in the *elasto-plastic range,* and the relationship between distortions and the number of cycles, introduce parameters for which data *are still lacking* in data banks, even for classical materials.

Once more, the gradient method [1] is used for the fatigue calculation. It provides an *endurance limit* $C_{e,a}$ for a pure alternating torque, given by the formula :

$$C_{e,a} = \frac{(a \log_{10} \chi + b)(I_p/r)\, k_s}{k_t \sqrt{3}}$$



where :

χ is the stress gradient,
k_t is the stress concentration factor [2],
k_s is a surface factor which depends on the surface roughness and on the ultimate strength of the material.

The gradient method also provides the *limiting amplitude $C_a(N)$* of the pure alternating reactive torque for a *given number of cycles N* :

$$C_a(N) = \frac{(a_2 \log_{10} \chi + b_2 \log_{10} N + c_2)\, k_s}{k_t \sqrt{3}}$$

The coefficients a, b, a_2, b_2, c_2 and k_s are given in reference [1] for different values of material properties and surface roughness.

The different fatigue failure torques were calculated by choosing a probability of non-failure of 90 % (the gradient method also defines the scatter in the results). The calculations were repeated for the four types of grooves defined.

The limiting torque at N cycles may be written (linearised portion of the *Wöhler* curve) :

$$C_a(N) = A - B \log_{10} N$$

The different results of the calculation are summarised in table II.

type of test specimen	$C_{e.a}$ *Nm*	A *Nm*	B *Nm*	torques C_a in *Nm* for N =		
				10^4	10^5	10^6
1	46.5	162.9	18.7	88.1	69.4	50.7
2	35.2	123.4	14.2	66.6	52.4	38.2
3	37.3	160.2	19.9	80.6	60.7	40.8
4	28.3	121.2	15.1	60.8	45.7	30.6

Table II. — Endurance limit torques and limiting torques for a given number of cycles calculated by the gradient method.

For each of the four types of groove, we made two test specimens, and subjected them to fatigue tests, with static torques C_s of 20 *Nm*, for the first specimen, and 30 *Nm*, for the second specimen, and with alternating torques C_a of $\pm$ 10 *Nm* and $\pm$ 20*Nm*, until failure took place. The test results are summarised in table III. In order to compare the experimental results with the theoretical predictions, the following assumptions must be made :

— assumption that an *equivalence* exists between an *alternating* torque C_a superimposed on a *constant* torque C_S and an *equivalent alternating* torque $C_{a,eq}$. *Goodman*'s assumption was adopted :

$$C_{a.eq.} = \frac{C_a}{\left[1 - \left(\frac{C_s}{C_{r.s}}\right)\right]}$$

— assumption of *damage*. The total damage S suffered by the shaft is determined by adding the different values of damage caused by each series i of n_i cycles with an equivalent alternating torque $C_{a.eq.}(i)$. Each damage value is n_i/N_i where N_i represents the number of cycles leading to failure for an *equivalent alternating torque $C_{a.eq.}(i)$:*

$$S = \sum_i \frac{n_i}{N_i}$$

$S = 1$ if failure takes place.

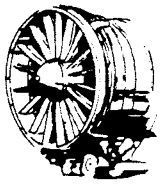
It is normal for S to be greater than 1 in the majority of cases, since the calculated *Wöhler* curve corresponds to a probability of non-failure of 90 %, to err on the side of safety. It should, however, be noted that the values obtained for S will depend on the *equivalence assumption* chosen. *Gerber*'s or *Smith*'s assumption would have resulted in *lower* values of the damage S.

A limiting condition suitably chosen by means of a practical method

Table IV gives the acceptable alternating torques C_a for values of the superimposed static torque C_s, and of the number of cycles N, under the least favourable conditions (probability very low; shear groove type 4).

test specimen characteristics	mean torsion load *Nm*	$C_{a,eq}$ *Nm*	alternating torsion load C_a *Nm*	number of cycles n_i	inspection	N_i	$\sum_i \frac{n_i}{N_i}$
N° 1 φ_M = 8.45 *mm* φ_P = 11.10 *mm* static torsion strength : $C_{r,s}$ = 158 *Nm*	20	11.45	± 10	$2.16 \cdot 10^6$	N/A	$1.3 \cdot 10^8$	
	20	22.9	± 20	$2.175 \cdot 10^6$	N/A	$3 \cdot 10^7$	
	20	34.3	± 30	$2.386 \cdot 10^6$	N/A	$7.5 \cdot 10^6$	
	20	45.8	± 40	$2.16 \cdot 10^6$	perm. distort. 2 drive faces twisted 0° 35′ 45″	$1.8 \cdot 10^6$	1.5
N° 1 φ_M = 8.45 *mm* φ_P = 11.10 *mm* static torsion strength : $C_{r,s}$ = 158 *Nm*	30	12.3	± 10	$2.16 \cdot 10^6$	N/A	$1.1 \cdot 10_8$	
	30	24.7	± 20	$2.16 \cdot 10^6$	N/A	$2.5 \cdot 10^7$	
	30	37	± 30	$2.16 \cdot 10^6$	N/A	$5.4 \cdot 10^6$	
	30	37	± 30	$2.16 \cdot 10^6$	N/A	$5.4 \cdot 10^6$	
	30	49.4	± 40	176 000	failure	$1.2 \cdot 10^6$	1
N° 2 φ_M = 8.55 *mm* φ_P 10.60 *mm* static torsion strength : $C_{r,s}$ = 120 *Nm*	20	12	± 10	$2.1756 \cdot 10^6$	N/A	$7 \cdot 10^7$	
	20	24	± 20	$2.16 \cdot 10^6$	N/A	10^7	
	20	36	± 30	$2.16 \cdot 10^6$	N/A	$1.4 \cdot 10^6$	
	20	48	± 40	30 400	failure	$2 \cdot 10^5$	1.9
N° 2 φ_M = 8.55 *mm* φ_P = 10.60 *mm* static torsion strength : $C_{r,s}$ = 120 *Nm*	30	13.3	± 10	$2.16 \cdot 10^6$	N/A	$5.7 \cdot 10^7$	
	30	26.6	± 20	$2.2428 \cdot 10^6$	N/A	$6.6 \cdot 10^6$	
	30	40	± 30	$2.16 \cdot 10^6$	N/A	$7.5 \cdot 10^5$	
	30	53.3	± 40	81 500	failure	$8.6 \cdot 10^4$	4.2
N° 3 φ_M = 8.45 *mm* φ_P = 11.10 *mm* static torsion strength : $C_{r,s}$ = 124 *Nm*	20	11.9	± 10	$2.16 \cdot 10^6$	N/A	$2.8 \cdot 10^7$	
	20	23.8	± 20	$2.16 \cdot 10^6$	N/A	$7.1 \cdot 10^6$	
	20	35.8	± 30	$2.16 \cdot 10^6$	N/A	$1.8 \cdot 10^6$	
	20	47.7	± 40	207 000	failure	$4.5 \cdot 10^5$	2
N° 3 φ_M = 8.45 *mm* φ_P = 11.10 *mm* static torsion strength : $C_{r,s}$ = 124 *Nm*	30	13.2	± 10	$2.16 \cdot 10^6$	N/A	$2.4 \cdot 10^7$	
	30	26.4	± 20	$2.21 \cdot 10^6$	N/A	$5.3 \cdot 10^6$	
	30	39.6	± 30	$2.16 \cdot 10^6$	N/A	$1.1 \cdot 10^6$	
	30	52.8	± 40	1 000	failure	$2.5 \cdot 10^5$	2.4
N° 4 φ_M = 8.55 *mm* φ_P = 10.60 *mm* static torsion streength : $C_{r,s}$ = 94 *Nm*	20	12.7	± 10	$2.245 \cdot 10^6$	N/A	$1.5 \cdot 10^7$	
	20	25.4	± 20	$2.16 \cdot 10^6$	N/A	$2.2 \cdot 10^6$	
	20	38.1	± 30	3 000	failure	$3.2 \cdot 10^5$	1.1
N° 4 φ_M = 8.55 *mm* φ_P = 10.60 *mm* static torsion strength : $C_{r,s}$ = 94 *Nm*	30	14.7	± 10	$2.16 \cdot 10^6$	N/A	$1.1 \cdot 10^7$	
	30	29.4	± 20	$2.185 \cdot 10^6$	N/A	$1.2 \cdot 10^6$	
	30	44	± 30	1 000	failure	$1.3 \cdot 10^5$	2

Table III. — Results of fatigue tests on four types of groove.



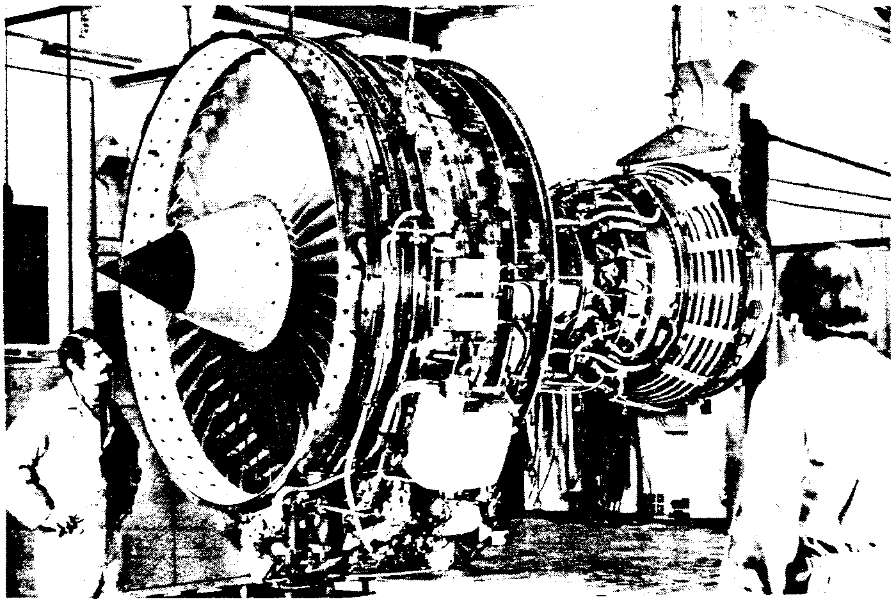
C_s Nm	C_a in *Nm* for N = 10⁴		
	10^4	10^5	10^6
10	54	41	27
20	48	36	24
30	41	31	21

Table IV. — Acceptable alternating torques for different values of the static torque and of the number of cycles.

During the normal tests at positive temperatures, the torque remained less than 10 *Nm*. During engine starts in very cold conditions (— 40 °C), the torque can rise to 55 *Nm* for one *second*, then fall fairly rapidly to normal; the least favourable cycle is represented by $C_s = 30$ *Nm*, $C_a = 25$ *Nm*, but the frequency of occurrence of this cycle is *very low*, since engine starting very rarely takes place at — 40 °*C*. In conclusion, the shear groove *fulfils its function well* : it *fails* when an *abnormal reactive torque* is present (seizure), while the *fatigue life meets the technical requirements*. The gradient method allows the static and dynamic strengths of steel parts with notches to be predicted by a fairly simple calculation. The results show *good agreement with experimental* data.

Acknowledgements

The author wishes to acknowledge the very helpful advice given by *M. Brand (CETIM)* and *M. Joris (FN)*.



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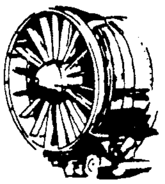

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APPENDIX

Torsion tests on five FN 87,2 steel bars

Characteristics of the bar test specimens used

Figure 4 shows the bars used, after failure :

a - two constant-section bars, 6 *mm* in diameter;

b - three bars 8 *mm* in diameter, with a groove at mid-length, 3 *mm* in radius. The diameter of the minimum cross-section is 6 *mm*.

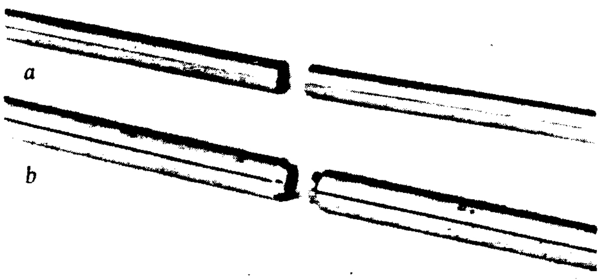

Fig. 4. — Bars with and without grooves used for the torsion tests *(after failure).*
(Fabrique Nationale, Herstal).

Results

For the bars *a*, the mean failure torque was 260 *kpm*.

For the bars *b*, the mean failure torque was 280 *kpm*.

Remarks

The gradient method predicted the experimental results with high accuracy : the bars with notches are stronger than the constant-section bars, contrary to what is predicted by the classical theory.

In the elastic range, the bars *b* would have been less strong and, in the plastic range, the two bars would have the same failure torque.