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NEW TRANSMISSION COMPONENTS FOR HELICOPTER APPLICATIONS.

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New transmission components for helicopter applications

Although there is no helicopter industry in the Netherlands, the subject of a specific development and the products of specific but existing high technology production methods, turned out to be potential candidates for major transmission components for existing and future helicopters.

With respect to this, components will be presented which have been proposed for the application in the drive systems for new helicopter programs.

First there is a flat crownwheel-pinion drive which is under development at DAF SPECIAL PRODUCTS B.V. An overview will be given of the specific geometry of the crownwheel tooth, the characteristics of this gearset and the results of the static and dynamic tests so far.

Further it will be presented how specific technologies, like flow turning and wet filament winding, that have been developed by UCN-NV for the production of ultra high speed rotors, can be applied for the production of very high quality metal and composite drive shafts in aerospace.

Flow turned and wet filament wound tubes for drive shaft application

In the tail rotor drive line of helicopters, thin walled metal tubes are used for the manufacturing of drive shafts. Until now, it is common practise that these tubes are manufactured by an extrusion process.
Consequently, tubes that are normally used for drive shafts, do have considerable geometrical unaccuracies like wall thickness variations and deviations from straightness. As a result of this, the dynamic specification of these tubes in terms of unbalance is such that they cannot be applied in drive shafts without (costly and time consuming) balancing and straightening procedures.

Because of weight reductions in drive lines longer drive shafts are profitable but the problems concerning balancing and straightening increase substantially. Furthermore drive lines with super critical shafts are much more reliable when high precision shafts are applied, because of low run outs in the critical frequency. Therefore there is an increasing market for high accurate long drive shafts.

The geometrical accuracy that can be reached with the flow turning process is superior to the extrusion process, and as a result of this the initial unbalances and deviations from straightness are much lower.

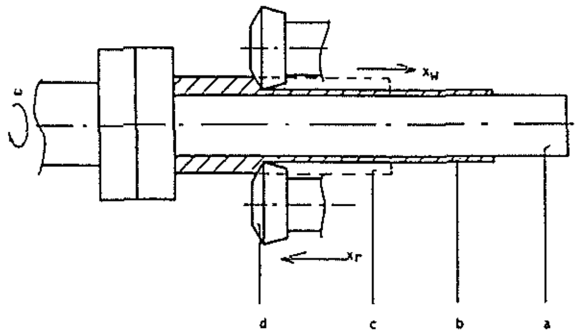
Drive shafts of aluminium, titanium and stainless steel may be applied in drive lines. Aluminium is a very common material for these applications because of high strength and low specific mass.
The titanium drive shaft is used especially when the shaft operates at a high temperature. Instead of a titanium drive shaft, one of stainless steel can be used because of material costs. All these materials can be flow-turned. However only the β-fase of titanium is suitable for this process.

Drive shafts of composites have always been of increasing interest in aerospace because of a high strength to specific mass ratio. The wet filament winding process can be used for the manufacturing of these high quality composite drive shafts. Compared to the unbalance of flow-turned metal drive shafts the properties of the filament wound drive shafts are even better.

As the flow turning and wet filament winding processes that are described below have been developed especially for low unbalance and perfect straightness of ultra centrifuge rotors, all expertise and equipment for production and qualification of long high quality tubes is available.

The flow turning process

To manufacture high quality thin walled long metal tubes for drive shafts a thick walled short tube is flow turned in one or more steps.
The number of reduction steps depends on material properties, geometry and surface conditions.

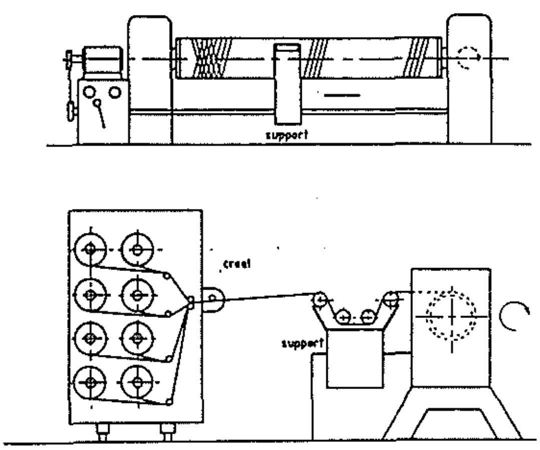


a : mandrel
b : final form of the work piece
c : starting form of the work piece
x_W: flow direction
x_r: direction of motion of the rollers
u : rotation of mandrel and workpiece

The thick walled short tube is put on a close fitting mandrel. Three rollers with a special geometry and in specific position move in axial direction while the mandrel and the workpiece are turning.
Due to the high degree of cold deformation the attainable yield stress is very high. For precipitation hardening materials a hardening procedure in combination with the flow-turning process is possible. This gives the possibility for further increase in strength.

The wet filament winding process

In the wet-filament winding process one or more tows of fibre which are impregnated with resin are wound on a mandrel. The winding pattern is determined by the desired mechanical properties.



After the winding process the composite is cured on the mandrel and in the furnace following a qualified procedure with respect to temperature and time. The result is a fibre reinforced plastic tube.

Criteria for drive shaft design

The main criteria for designing drive shafts in aerospace are:
- reliability during life period
- low mass

These design criteria may be basically translated into material and mechanical properties of the drive shaft.

In the design process three mechanical criteria have to be taken into account:

1. The strength of the shaft. The shaft has to withstand the maximum torsional moment without any failure.

2. The operating speed range may not coincide with the critical speed of the drive line (sub- or super-critical).

3. The shaft must be stable against torsional buckling (instability requirements).

When pure torsion of a shaft is considered these criteria can be translated into analytical formulae for a drive shaft with:

Length: L.
Radius: R
Wall thickness: t

For isotropic materials the following formulae can be used:

Ad. 1.: $$\tau = \frac{M}{2 \cdot \pi \cdot R^2 \cdot t} \quad \text{and} \quad 2.\tau < \bar{\sigma} \quad \text{(Tresca)} \qquad (1)$$

in which τ: shear stress due to the torsional moment M
$\bar{\sigma}$: allowable stress

Ad. 2: $$n_0 = 60 * \frac{\pi \cdot}{2\sqrt{2}} \frac{R}{\cdot L^2} * \sqrt{\frac{E}{\rho}} \quad \text{(rpm)} \qquad (2)$$

in which n_0: first bending critical speed
E : Youngs modulus
ρ : specific mass

The effects of shear stiffness and gyroscopic moment are neglectible.

Ad.3: $Mcrit = k*4.42* \frac{E \cdot t^{9/4} \cdot R^{5/4}}{L^{1/2}}$

(deduced from Timoshenko [1])
Mcrit: torsional buckling load
k : correction factor between theory and experiment, according to [1] k=0.7

For <u>orthotropic materials</u> such as carbon fibre reinforced plastics (CFRP) the following simple analytical evaluations can be used:

Ad.1: The stresses in the different layers of the composite shaft, as a result of the maximum torsional moment, are calculated using the classical laminate plate theory [2]. In this analyses the material properties for the different layers are calculated from the specified properties of the fibre and the resin and from the experiments on the composite.

Ad.2: The critical speed is approximated using the following simplified formula for a thin-walled tube

$$n_0 = 60* \frac{\pi}{2\sqrt{2}} * \frac{D}{L^2} * \sqrt{\frac{E_{ax}}{\rho}} \quad \text{(rpm)} \qquad (4)$$

The shear stiffness and gyroscopic terms have been neglected.

Ad.3: The critical torsional moment concerning stability can be calculated from the simplified "Simitses relation" [3].

$$Mcnt = k* \frac{\pi^3}{6} * \frac{R^{5/4} \cdot t^{9/4}}{L^{1/2}} * E_{ax}^{3/8} * \left(\frac{E_{tg}}{1 - \nu_{ax-tg} \cdot \nu_{tg-ax}}\right) \qquad (5)$$

Eax: Young modules in axial direction
Etg: Young modules in tangential direction
νax-tg: Poisson ratio in axial-tangential direction
νtg-ax: Poisson ratio in tangential-axial direction

These analysis procedures for orthotropic materials are incorporated in an optimization computerprogramme in which starting with a proposed lay-up pattern the geometry is optimized for mass. The result is an optimal lay-up geometry with respect to fibre orientation and wall thickness.

Mass of metal and composite tubes

The predesign of a helicopter tail-drive shaft is used for a comparison between the materials mentioned in this paper. On basis of the mechanical formulae and the material properties the mass per unit length of aluminium, titanium, stainless steel and composite tubes for a drive shaft as specified are calculated.

Length: L = 1500 mm
Radius: D = 50 mm
Speed: n = 5000 rpm
Maximum power: P_{max} = 500 kW
(Maximum torsional moment = 1000Nm)

It turned out that for all the materials the drive shaft design is critical for torsional buckling.
When a safety factor against buckling of 0.54 and a k-factor of 0.7 (eq. (3)). is taken into account the following results for metal tubes are obtained:

	ρ [kg/m3]	E [GPa]	σ 0,2 [MPa]	no [rpm]	tmin [mm]	σmax [MPa]	Mass per unit length kg/m
Aluminium (6061)	2850	72	170*	7440	1.50	84	1.48
β-Titanium	4850	103	1300*	6820	1.26	134	1.92
Stainless Steel	8030	200	1100*	7390	.95	102	2.40

* yield stress after flow turning.
tmin is minimum wall thickness

The aluminium drive shaft turns out to be preferable concerning mass. The flow turned titanium and stainless steel drive shafts are overdimensioned (very safe) concerning strength.
The drive shaft as specified, operates at a sufficient distance below its critical speed $n<0.8*n_0$.

The characteristics of an optimized CFRP-wound shaft are:

Material: T300 HT-fibre Epoxy resin Winding pattern: 0.93 mm $\pm$ 80° 0.47 mm $\pm$ 25° Total wall thickness= 1.40 mm Eax = 40 Gpa Etg = 97 Gpa Gax-tg = 15 Gpa νtg-ax = 0.27 νax-tg = 0.11 ρ = 1600 kg/m 3 τmax = 0.217 $n=0.8*n_0$ mass per unit length= 0.78 kg/m k=0.7 according to [1]

Gax-tg: shear modulus in the axial-tangential direction
τmax : maximum shear stress in one layer
τ : allowable shear stress

The weight of the carbon composite shaft is 47% lower in comparison with the aluminium shaft.

Experimental data

Aluminium and stainless steel drive shafts.

The flow turning process has been used to manufacture aluminium (6061) and stainless steel (AISI-304 L) -tubes with an internal diameter of 86 mm and a length of more than 1400 mm. Data on wall thickness and geometry-accuracy are given:

	Aluminium 6061	Stainless Steel AISI 304 L
Wall thickness [mm]	1.65	1.05
Wall thickness variation		
tangential	< $\pm$ 0.01	< $\pm$ 0.01
axial	< $\pm$ 0.03	< $\pm$ 0.03
Straightness [mm/m]	x < 0.15	x < 0.2
Roundness [mm]	< 0.1 mm	< 0.1 mm

The flanges have been EB-welded (Electron-Beam) to the aluminium and stainless steel tubes with a resulting shaft length of 1400 mm. The total mass of an aluminium shaft was 2.0 kg and 3.5 kg for the stainless steel shaft.
The measured unbalance for the resulting shafts at the position of the flanges was less than 50 grmm.
The torsional (static) strength of the shafts has been tested. In all cases the shafts failed due to buckling. The results are summarized and compared to the theoretical value:

	Buckling load	Theoretical
Aluminium	= 1800 Nm	2020 Nm (k=0.7)
Stainless Steel	= 2700 Nm	2890 Nm (k=1.0)

In a transmission test rig aluminium tubes have been tested for hundreds of hours at a torsional moment of 500 Nm and a speed of 5000 rpm.

Composite drive shafts:

Because of the availability of a mandrel within UCN the first CFRP-wound drive shafts are produced at a diameter of 150 mm and a length of 1400 mm. The manufactured lay up is a multi layer pattern of T300/Epoxy composite with a total wall thickness of 0.94 mm and a fibre volume fraction of 60%.

For this geometry the torsional stiffness and the critical torsional moment is calculated and determined from experiments wich resulted in a 20% higher value for the experimental results. Further study will be performed to get the theory in agreement with the experiments.

For experiments end flanges out of aluminium are connected to the composite tube. The connection between tube and endflanges is realised by bolts as well as by adhesive. The connection between CFRP-tubes and end flanges is studied in a separate research programme within UCN.
The composite drive shaft can be manufactured with less than 15 grmm unbalance at the end flanges and 0.2 mm/m straightness.

Literature

[1] S.P. Timoshenko and J.M. Gere
Theory of elastic stability.

[2] R.M. Jones
Mechanics of composite materials.

[3] W.Fuch, P.Lutz
CFRP-precision wound tubes in Aerospace Engineering.

The DAF Crownwheel

1.Introduction

First I would like to mention some of the characteristics of the DAF Crownwheel. After that I would like to give insight into the stress calculations we perform on this new type of gear.
Finally I will tell something more about the testing we did to prove the capability of this new transmission.

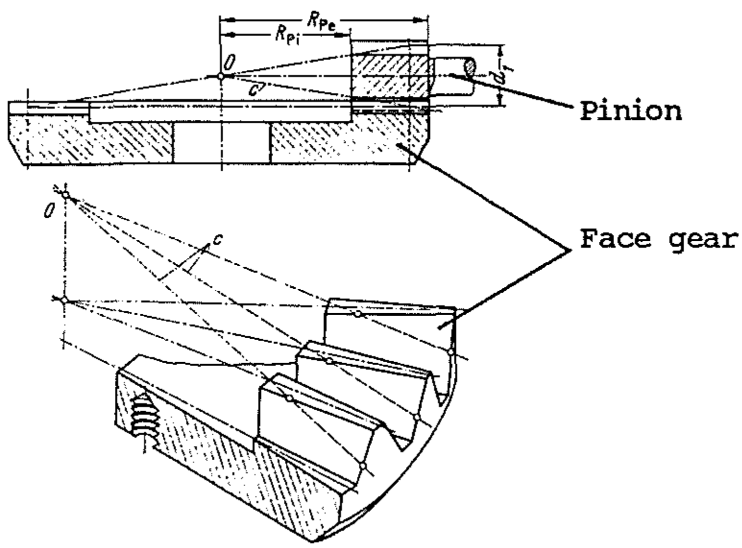


fig.1 Crownwheel gearing

The DAF Crownwheel gearing consists of a face gear and a spur gear which have intersecting shafts and which have some differences/advantages compared to conventional spiral bevel gears. The face gear, as it is often called, is similar to bevel gears but mates with spur or helical pinions. By this combination a bevel-like gear is created which meshes through line contact between pinion and crownwheel creating large contact ratio's, which has some interesting characteristics.

Advantages:

- Forces
  Reaction forces resulting from the gear contact (mesh) can be resolved in a tangential and a radial force on the pinion shaft. There is no axial force on the pinion shaft as it is an involute straight spur gear.
  For the same reason mentioned above there is no radial reaction force on the crownwheel.

- Assembly
  Assembly of the Pinion is easier as the axial positioning of the Pinion shaft is not critical.

- Efficiency
  The gear teeth slide relative to another only in one direction. A good gear efficiency can be achieved. Efficiency ratings are at least in the same range as those feasible for ordinary spiral bevel gears

- Contact ratios
  Large contact ratios (larger than 2.5 or even over three) can be obtained, which result in lower bending stresses, lower contact stresses and potentially lower noise level.

- Line contact
  Meshing is performed through line contact between Pinion and Crownwheel. Results of computer simulation on page 10 show these contact lines in various meshes.

- Symmetry
  The Crownwheel is circular symmetrical which implies no difference in specifications when rotating in either forward or reverse direction.

Limitations:

- Minimal reduction ratio
  As tooth face width is related to the reduction ratio, it is not possible to accomodate reduction ratios of less than 3.5:1 and at the same time being weight efficient.

2.Geometrical description

When calculating stress of the DAF Crownwheel, the geometry of the crownwheel-tooth has to be described first. This mathematical description has been made, by rotating theinvolute pinion and calculating the contraform which is the crownwheel-tooth geometry.

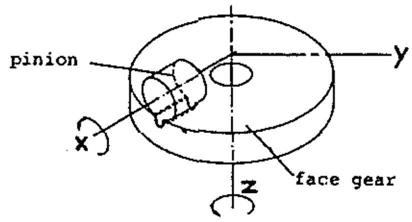


fig.2

Defining the coordinate system and gearparameters according to fig.2; the geometry of the crownwheel-tooth can be described as follows:

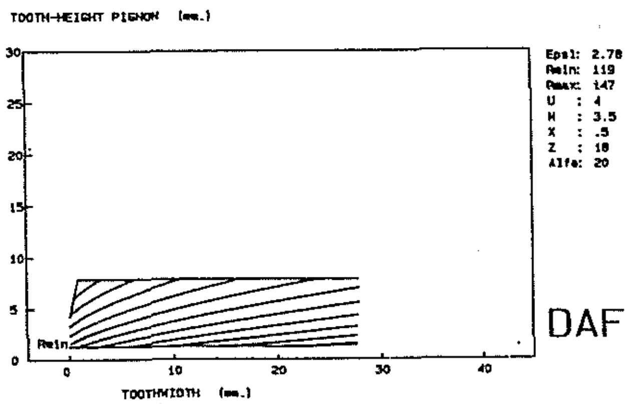
U : ratio
k : 1/U
α_1: press. angle pinion
α : press.angle crownw.
r_{b1}: radius base-circle

$$
\begin{cases}
x = r_{b1}\left[\dfrac{U\cos(k\varphi(\alpha,\alpha_1))}{\cos(\alpha)} + \dfrac{\sin(\alpha_1-\alpha)\sin(k\varphi(\alpha,\alpha_1))}{\cos(\alpha_1)}\right], \\[2ex]
y = -r_{b1}\left[\dfrac{U\sin(k\varphi(\alpha,\alpha_1))}{\cos(\alpha)} - \dfrac{\sin(\alpha_1-\alpha)\cos(k\varphi(\alpha,\alpha_1))}{\cos(\alpha_1)}\right], \\[2ex]
z = -\dfrac{r_{b1}\cos(\alpha_1-\alpha)}{\cos(\alpha_1)},
\end{cases}
$$

3. <u>Meshing characteristics</u>

Lines of contact and the total contact length can be determined by using the geometrical equations mentioned above. The results of these calculations are depicted in the following figures.

Epsl: contact ratio
Rmin: inside radius crownwheel
Rmax: outside radius crownwheel
U : ratio
M : module
X : profile corr.
Z : pinion teeth
Alfa : nom. press. angle pinion



DAF Sp

fig.3 Lines of contact in several meshes

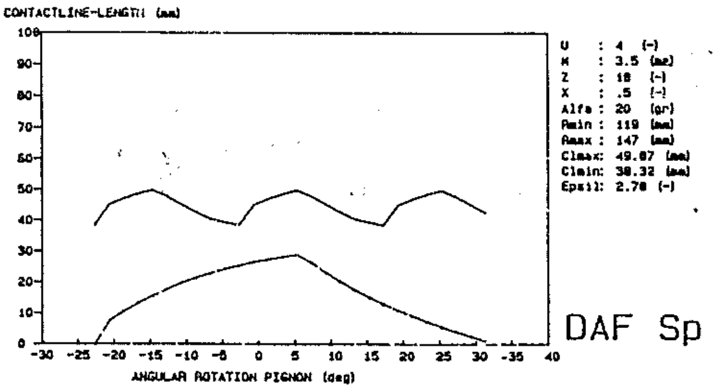


fig.4 Total contact-length
Clmax= maximum contact-length
Clmin =minimum contact-length

Corresponding stiffness of teeth and flange yields the load distribution over the teeth and tooth-width.

4.Strength calculations

The pinion appears to be the weakest part of the transmission. As the crownwheel has an average pressure angle which is higher than the 20 degree pressure angle of the pinion, the bending stresses are higher in the root of the pinion. The larger rotation speed of the pinion results in a lower endurance for this component as well.

Because of the high contact ratio of this new transmission there is no occurance of single pair contact. So the calculation of stresses is based on the highest point of double pair contact, the result of which is shown in the following figures.

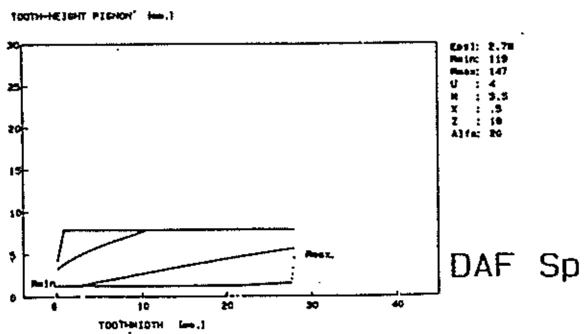


fig.5 Highest point of double pair contact

F_{bt}: load
h_F : bending moment arm
α_F: current press. angle
s_{Fn}: foot thickness
b : width
ρ_F: radius

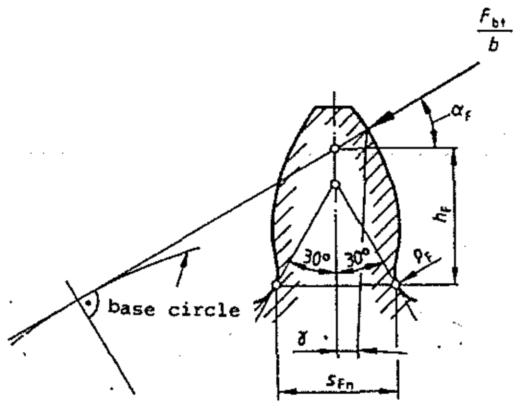


fig.6

The stress concentration factor Y_S and geometric factor Y_f can be derived when the height of the bending moment arm h_f is found on the outside of the gear contact line. When this factor is determined, formulae of DIN 3991 are applicable for the strength calculations of the pinion. This results in a lower value for the bending moment because of the double pair contact (a lower value for the bending-moment arm h_f).
So a lower value for the bending stress is found in the root.

5.Testing and experiments

Performance testing

For testing a pretensioned testrig was developed incorporating dynamic and static test facilities.
The following figure shows the lay-out of the rig.

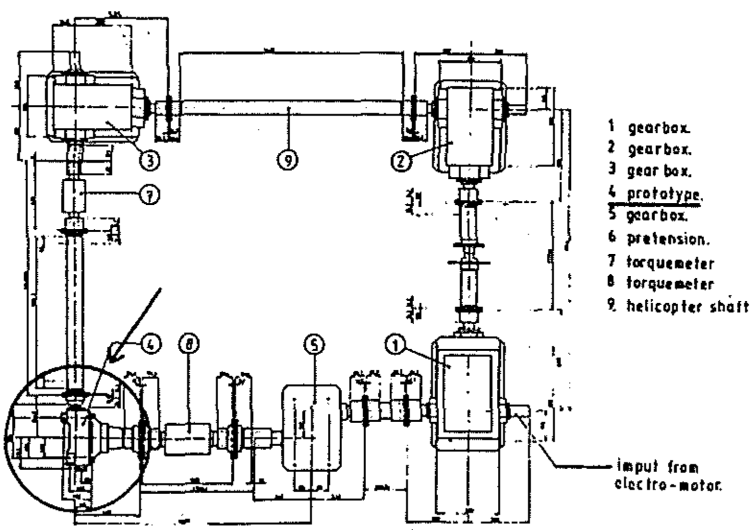


fig.7

Speed, pretension, oil temperature, oil-flow variation and efficiency measurement is possible, to test transmissions under all circumstances.

A crownwheel transmission suitable for a helicopter tail-rotor gearbox has been tested for several hundreds of hours.
The table below lists the data of this particular transmission.

Data	: Tail-Rotor Gearbox
Ratio	: 4 ;pressure angle pinion = 20 deg.
Module	: 3,5 width = 28 mm
Pinion	: 18 teeth: spur 63 mm: pitch diameter material: 17 CrNi 6
Crownwheel	: 72 teeth 266 mm, pitch diameter material: 17 CrNiMo 6
Endurance test	: 250 hours at 250 Kw
Measured efficiency	: 98,5% for the complete gearbox, including bearings
High load	: 10 hours at 400 Kw
(Semi)static	: optimizing tooth clearance and tooth contact

<u>Stress-verification testing:</u>

Strain gauge tests were carried out to measure bending stresses in pinion and crownwheel root.
These tests showed the stresses measured to be in good accordance with the calculated stresses.
The following figure shows the mounted strain gauges.

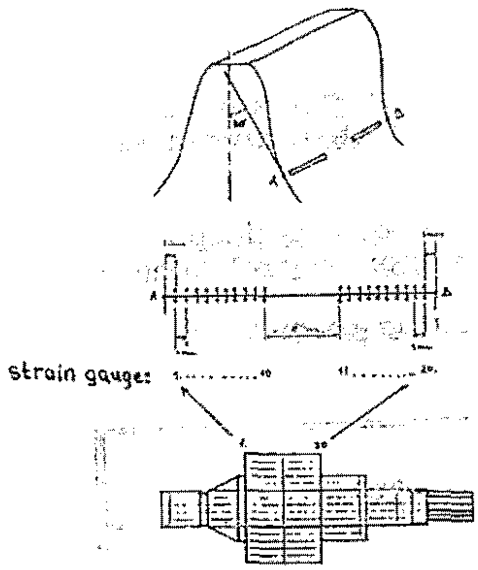


fig.8

Actual situation crownwheel development: Over the last few years DAF SP has performed both theoretical and experimental work on the Crownwheel. Although the development of the basic know-how has not been completed yet, the results so far are such that a dedicated production method could be defined. Moreover they have given so much confidence in this gearset that a separate company has been established to develop this production method and to produce crownwheels for many applications.