PAPER NR. : 85

# THE DESIGN OF AN ADVANCED ENGINEERING GEARBOX


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## TENTH EUROPEAN ROTORCRAFT FORUM

AUGUST 28 — 31, 1984 — THE HAGUE, THE NETHERLANDS

## NOTATION

| | | | |
|---|---|---|---|
| A | = | Application Factor | - |
| C | = | Configuration Factor | - |
| GBW | = | Gearbox Weight | lbf |
| GW | = | Gear Weight | lbf |
| K | = | Lloyds K factor | lbf/in/in |
| N | = | Number of Meshes | - |
| P | = | Profile Factor | - |
| R | = | Ratio | - |
| $Sb_1$ | = | Design Stress | $lbf/in^2$ |
| Sb | = | Stress Parameters | $lbf/in^2$ |
| T | = | Torque | lbf.in |
| TGW | = | Total Gear Weight | lbf. |
| WR | = | Weight Ratio | - |
| a | = | Cone Distance | in |
| b | = | Facewidth | in |
| k | = | Combined Load Factors | - |
| $I_,$ | = | Idler Factor | - |
| $\delta'$ | = | Density Parameter | $lb/in^3$ |
| $\sigma$ | = | Basic Allowable Stress | $lb/in^2$ |

Subscripts

| | | |
|---|---|---|
| I | = | Input |
| n | = | Number of Stages |
| o | = | Output |
| PS | = | Parallel Shaft |
| EP | = | Epicyclic |

SUMMARY

Equations are presented which show that gear and gearbox weights are proportional to the torque on the gear and gearbox respectively. An element of these equations called the "configuration factor" can be derived from the gearing layout, and then used as a comparison of gearbox weights. Using this configuration factor, three features essential to any low weight gearbox have been identified. All triple reduction layouts, many using these three factors, have been analysed using the configuration factor technique and a particular low weight configuration identified. The weight of this new configuration has been predicted and verified by traditional means and shows a 38% weight saving when compared to the existing Sea King Gearbox.

Development programmes are now nearing completion to enable this configuration to be incorporated into improved versions of certain current helicopters and new designs.

## THE DESIGN OF AN ADVANCED ENGINEERING GEARBOX

The main rotor gearbox of a helicopter represents a significant proportion of the empty weight, typically about 10%. A major reduction in the gearbox weight can therefore be an important contribution in achieving improvements in helicopter performance.

In the design phase there are numerous gearbox configurations, all providing the requisite torque, speed, life and reliability that could be adopted. It is therefore necessary to have some accurate means of predicting gearbox weights if the lightest is to be selected.

One of the most eminent men of the gearing industry H E Merritt (Ref. 1), said "gear units have one feature which is perhaps unique amongst machines, in that the overall dimensions, weight and cost directly reflect the choice of the working stress, at one point". It follows then, that knowing the working stress it should be possible to calculate the weight of a given gear rim before any design layout work is instigated. If this can be achieved, the weight of each gear rim comprising the configuration can be calculated. The gear rim weights can then be added to give the configuration weight. This would enable all potential configurations to be considered for a particular application and a selection made on weight and cost considerations. Further, by indicating which particular configurations are light and which are heavy, where multi-reductions are necessary, the lightest configuration can be used at the appropriate stage of the transmission. It also indicates where the optimum reduction should be incorporated.

Possibly the first analytical assessment of gearbox weights was by Jupe (Ref. 2). He considered an annular pinion and wheel, and taking gear sizes dependent upon root stress limitations, developed equations which could be used to estimate gear weights. An application factor was then applied to the equation to determine the basic gearbox weight. The application factor was, however, based upon helicopter practice with a particular root stress. The effects of surface durability and rim thickness variation upon the equations were not considered. The basic equations for parallel shaft and epicyclic gearing were shown to be

Parallel Shaft Gearing

$$GBW_{PS} = \left[\frac{1}{N} + \frac{1}{R}\right]\frac{To}{A} \qquad \text{------------------(1)}$$

Epicyclic Gearing

$$GBW_{EP} = 1{\cdot}4\left[\frac{1}{N} + \frac{1}{2} - \frac{1}{R}\right]\frac{To}{A} \qquad \text{---------------(2)}$$

It was shown by Willis (Ref. 3) that the weight of a gearbox assembly could be predicted from a basic knowledge of the horsepower, surface durability and gear configuration. In his paper he assumed that the weight was proportional to swept volume thus assuming, in the first instance, a solid pinion and wheel. An application factor was introduced to convert the weight equation based on configuration to a basic gearbox weight. This application factor was also used to distinguish between different types of gearing, ie aircraft, hydrofoil or commercial, and also to allow for the gears being hollow. The effect of root stress was neglected which, it will be shown later, dominates the weight of annular gears. Taking his equation and rearranging, equations (3) and (4) can be derived

For parallel shaft gearing

$$GBW_{PS} = \left[1 + \frac{1}{R} + R + \frac{1}{R^2}\right]\frac{To}{A\,K} \qquad (3)$$

For epicyclic gearing

$$GBW_{EP} = \left[\frac{1}{N} + \frac{1}{2} - \frac{1}{R} + 1 + \frac{(R-2)^2}{NR^2}\right]\frac{To}{A\,K} \qquad (4)$$

Working from first principles, it has now been shown that using root and surface stress criteria and introducing a rim thickness quotient, equations have been derived which can be used to predict the weight of any individual gear. This work proved that, given the material density, design stress limitations, rim thickness quotient and ratio, the weight of the gear can be determined and is directly proportional to the torque it transmits. The accuracy of the method is shown in Figure 1, the errors being mainly on bevel gears due to variations in shaft angle. Consequently it is possible to evaluate the weight of any gear stage by combining the equations for each individual gear. The corresponding equations for the two types of gearing are:-

For parallel shaft gearing;

$$GW_{PS} = f\left\{\left[\frac{1}{N} + \frac{1}{R} - \frac{(N+1)}{N}\,\frac{Th\,K}{Sb\,Y}\right]\frac{To}{Sb}\right\} \qquad (5)$$

For epicyclic gearing

$$GW_{EP} = f\left\{\left[\frac{1}{N} + \frac{1}{2} - \frac{1}{R} - \frac{(R-2)}{R^2}\,\frac{Th\,K}{Sb\,Y}\right]\frac{To}{Sb}\right\} \qquad (6)$$

From which it can be clearly seen that the weight is dominated by the root stress Sb

Equations (5) and (6) can be expressed in the general form

$$GW = \frac{\delta'}{Sb'} \quad C \quad To \text{ --------------------- } (7)$$

The terms inside the square parentheses of equation (5) and (6) are governed by the ratio, number of contacts and the gear layout (configuration); hence the variable 'C' has become known as the configuration factor. From equation (7) it can be seen that the lower the configuration factor C, the lower the gear weight.

A helicopter main gearbox must be capable of accepting the engine output speed, usually about 20000 rpm, and reducing it to the rotor speed in the vicinity of 200 to 300 rpm. In order to achieve this a multistage gearbox must be adopted. The weight of multistage gears can be calculated by summating the gear weights for each individual gear stage:-

$$TWG = \delta' \sum_{n=O}^{n=I} \frac{Cn\,Ton}{Sb'n} \text{ ------------- } (8)$$

Sb'n being retained as a variable since each gear design (bevel, helical or conformal) has its own allowable design stress level.

## VARIATIONS OF Sb' WITH TYPE OF GEARING

During the analysis it became apparent that the variable Sb' would have to be modified to allow for whether the gears are conformal, involute, idler or bevel, and for various dynamic load effects. It was shown by Merritt (Ref. 4) that the strength of bevel gears is less than the strength of parallel gears by an amount (a-b)/a. If this is combined with the idler factor the variable Sb' can be written

$$Sb' = K \, \sigma \, I \left(\frac{a-b}{a}\right) P \text{ -------------------- } (9)$$

where (a-b)/a is only applicable for bevel gears, I for an idler gear and P for the effect of the conformal profile.

We can now accurately predict the weight of individual gears, and the weight of gear stages during the initial design phase. What is now required is some method of converting the gear rim weight to the basic gearbox weight incorporating gears, shafts, bearings and gearcase.

## EVALUATION OF GEARBOX WEIGHT EQUATION

It would seem reasonable to postulate that if a manufacturer designs gearboxes for some specialised field of application, such as helicopter, marine steam turbine, or diesel driven commercial vehicle, then since the allowable gear stresses and bearing lives will be of the same order in each field of application, so will the ratio of total rim weight to gearbox weight.

This ratio of rim weight to gearbox weight will vary widely from application to application and perhaps, but not by such a large amount, from manufacturer to manufacturer. If this is correct the gear rim weight equation can be modified to give the gearbox weight by introducing a factor WR for the weight ratio.

$$GBW = \delta' \sum_{n=O}^{n=I} \frac{Cn}{Sb'n} \, \frac{Ton}{WRn} \quad \text{-----------(10)}$$

Just such a postulation was made by Jupe (Ref. 2) and Willis (Ref 3) and in order to test this hypothesis the weight equation (10) was applied to a wide range of gearbox configurations with different overall ratios. The extent of these is shown in Figure 2.

The weight was calculated for each gearbox and the results compared with known gearbox weights. Graphical presentation of the comparison is shown in Figure 3. It can be seen that a good co-relation was obtained.

## THE USE OF THE CONFIGURATION FACTOR IN DESIGN ANALYSIS

Using this technique we can now obtain the configuration factor C for existing gearboxes. Figure 4 shows the influence lines representing the configuration factor for a Sea King. Bearing in mind that gear and hence gearbox weight is directly proportional to C, it can be seen how varying the number of planets and ratio affects the influence lines and hence changes the gearbox weight.

Figure 5 shows the configuration factor for a basic two-pinion and a three-pinion Lynx gearbox. Note how the weight reduces with ratio and the abrupt change at the output ratio of 7.63/1, caused by the configuration factor being influenced by the output train at ratios below 7.63/1 and the bevel train at ratios above 7.63/1. It can be clearly seen that increasing the output ratio will drastically reduce the gearbox weight.

The study of this factor for various configurations enables the relative merits of each design to be evaluated. Figure 6 shows the configuration factors for all single stage variations of planetary and star epicyclic configurations with the parallel shaft configuration superimposed. This clearly shows why the conflict persists between parallel shaft and planetary epicyclic. Which type is the lightest depends on the ratio and number of contacts selected. The difficulty is compounded due to the inter-relationship that exists between N and R for epicyclic gearing, which restricts the value of C to those within the shaded area on Figure 6.

It can be seen from equation (5) that to reduce the parallel shaft gear Stage weight, both N and R must be increased, whereas to reduce the epicyclic gear stage weight N must be increased and R reduced. Due to the enclosed nature of the epicyclic, any increase in N naturally results in a reduction of R and vica versa. This restricts the epicyclic to the shaded area and reduces its weight saving potential.

As no such restriction occurs with parallel shaft gearing both N and R can be increased. The introduction of the high ratio conformal coupled with the four pinion output stage has therefore tilted the balance in favour of the parallel shaft configuration (location D in Figure 6).

## GEARBOX DESIGN

Work on conformal gearing was commenced in the early sixties by WHL for use in helicopter transmissions. Due to its high load carrying capacity and ability to provide a high ratio stage, final drive conformal gears were incorporated in the Lynx gearbox. This allowed the gear reduction from engine module to rotor to be obtained in only two stages. A third stage reduction is provided in a separate gearbox integral with the engine module, hence the overall ratio could be obtained in only three stages instead of the four previously required.

Subsequent studies outlined above, into gearbox weights showed the advantages of multiple contacts (a philosophy highlighted by epicyclics) for increasing the power capacity of a gear train and when it was required to increase the Lynx capacity an additional conformal pinion was adopted.

These studies also led to the conclusion that four pinions along with an increase in output gear ratio, would provide a considerable increase in torque capacity and/or reduction in gearbox weight. Further, the total reduction in speed from the engine to rotor to could be obtained in only three stages, all housed within the one gearase. This facilitated greater flexibility in the selection and location of the remaining two stages.

### High Ratio Conformals

Examination of a nine tooth involute gear shows a weak tooth due to the heavy undercutting, whereas a nine tooth conformal gear avoids this problem, Figure 7. This enables small strong pinions to be designed and hence the high output ratio is obtained within a volume significantly below that possible with involute gears. Existing Lynx ratios are 7.63/1, but later designs are incorporating ratios of 12/1 and higher ratios could be adopted.

### High Speed Spiral Bevel Gears

It can be seen from equations (5) and (9) that the weight of a spiral bevel gearset is considerably greater than the weight of an equivalent parallel shaft gearset for any given output torque. For this reason significant weight savings are obtainable by locating the necessary bevels in a low torque area. The optimum position is at the input to the gearbox, where they can operate at engine turbine speed and hence low torque.

Interstage Gearing

As the input and output configurations had been defined and it was still required to obtain the overall ratio in only three stages, an interstage configuration which could connect the single input wheel to two output pinions was required; this necessitated the adoption of a divide train.

A gear configuration that incorporates all these features is shown in Figure 8, the configuration being defined as a bevel driven, dual tandem, articulated locked train.

The spiral bevel wheel has been placed between the conformal pinions thereby producing a squat gearbox. This bevel location also enables a completely clean bottom to be obtained which facilitiates the assembly of the control mechanism as there are no fouling points. The high output stage ratio results in a large central hole which also eases the assembly of the control mechanism.

The influence line for this configuration is shown in Figure 9 together with the results of Figure 5, and shows a significant weight reduction. The weight reduction is predominantly a result of the adoption of a high ratio output stage, multiple contact, and high speed bevels.

Gearcase

The adoption of the high ratio conformal output stage causes problems with conventional gearcase designs. As the output wheel becomes larger a conventional case to enclose it will inevitably have a large surface area, be heavy, and reduce the advantage of the gear configuration. If this is to be avoided the advances in gear design must be matched by advances in case design.

A new gearcase called a semi-skeletal case (Figure 10) has been incorporated and whilst this contributes significantly to the weight saving, it does represent a departure from normal case design philosophy. Conventional gearcases are designed to sustain rotor loads which are passed through the case walls from the mainshaft bearings to the fuselage structure via the gearbox mounting locations.

The semi- skeletal gearbox is designed in such a manner that rotor loads applied to the wheel bearings are transferred through a stiff central stub axle directly to the fuselage. This design enables the loads to by-pass the case walls and so enables the walls to be used purely to support gear loads, retain the oil, and prevent the ingress of debris. This enables an extremely light outer case to be used.

This Semi-Skeletal Case has been subjected to a finite element analysis (Figure 11) which showed the maximum deflection to be reduced by 45% under ultimate load conditions compared with the Lynx case under the same loads. This reduction in deflection has been obtained with a simultaneous reduction in case weight of 56%.

These four features combine to improve the following gearbox characteristics.

| | |
|---|---|
| Specific Weight (Basic Weight per unit output torque) | High Speed Spiral Bevels<br>High Ratio Conformal<br>Multiple Pinion<br>Semi-Skeletal Case |
| Reliability | Three Stage Gearbox |
| Reduce Costs | Reduced number of gear components<br>Three Stage gearing<br>Simple gearcase |
| Environment | Reduced number of exitation frequencies |

## ADVANCED ENGINEERING GEARBOX

Figure 12 shows the aforementioned configuration mounted inside the semi-skeletal case to produce what is known as the Advanced Engineering Gearbox.

The two major technological developments which have contributed to this advance are high ratio conformals and the semi-skeletal case. Figure 13 shows a view of the high ratio conformal test gears which are now undergoing fatigue substantiation on a purpose built test rig (Figure 14).

The semi-skeletal case has been designed and manufactured in titanium and is shown in Figure 15. Tests are currently being conducted to measure deflections under representative loading (Figure 16) and these will be compared with results from a finite-element model. Static testing on a complete gearbox will commence later this year.

## CONCLUSION

We now have a technique to predict the weight of any gearbox and so enable all potential configurations to be analysed for a particular application, and the optimum design selected.

A detailed design study has been conducted on a large helicopter gearbox and from detail drawings the weight was calculated to be within 7% of that predicted. The weight of the new gearbox, based on existing stress limitations and reliability, is only some 62% of the more conventionally designed gearbox.
In conclusion, it has been shown that both gear rim and basic gearbox weight are directly proportional to the output torque. This leads to the identification of three configuration features which must be included if a low weight gearbox is to be designed.

1) High Ratio Output Stage

If the output stage adopted is of the parallel shaft configuration, the output ratio should be as high as possible; this enables the torque on the output pinion and all proceeding gearing to be minimised. The adoption of a conformal profile, with its ability to provide a higher ratio than an involute tooth form, greatly enhances the reduction in torque.

2) Multiple Output Pinions

These reduce the torque per mesh on the output gear train and so utilise the wheel teeth more efficiently. If the output stage is an epicyclic, selection of the maximum number of planets forces the epicyclic ratio towards 2/1, and the conflict between maximum ratio and the number of planets arises. The selection of an epicyclic output stage therefore, prevents the achievement of the ultimate weight saving.

3) High Speed Bevels

Due to the higher weight per unit torque of bevel gears against parallel shaft gears, any bevel gears should be located in the area of minimum torque ie the input to the helicopter main gearbox.

ACKNOWLEDGEMENT

The author would like to thank the MOD(HRP/RW) for their support of this work and permission to publish the paper.

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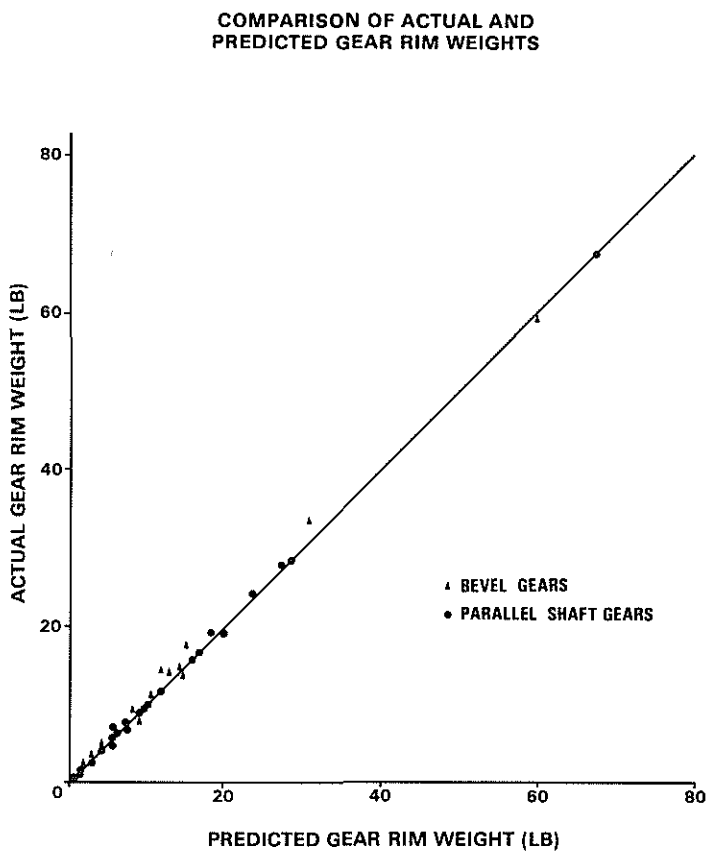



Fig. 1

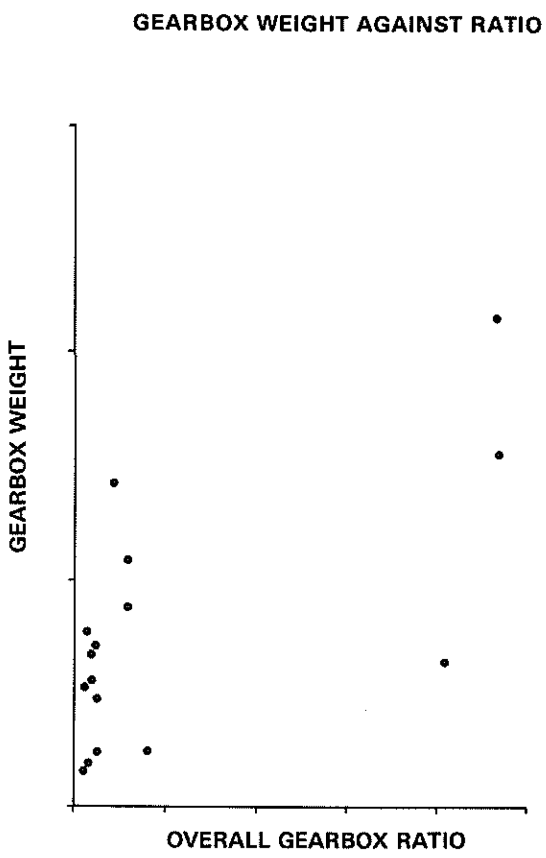



Fig. 2

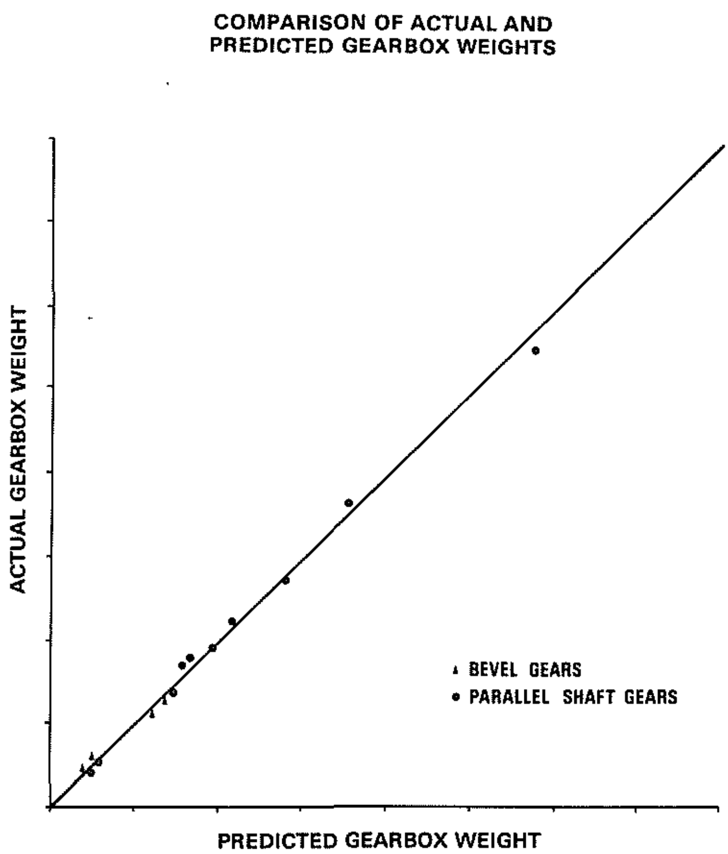



Fig. 3

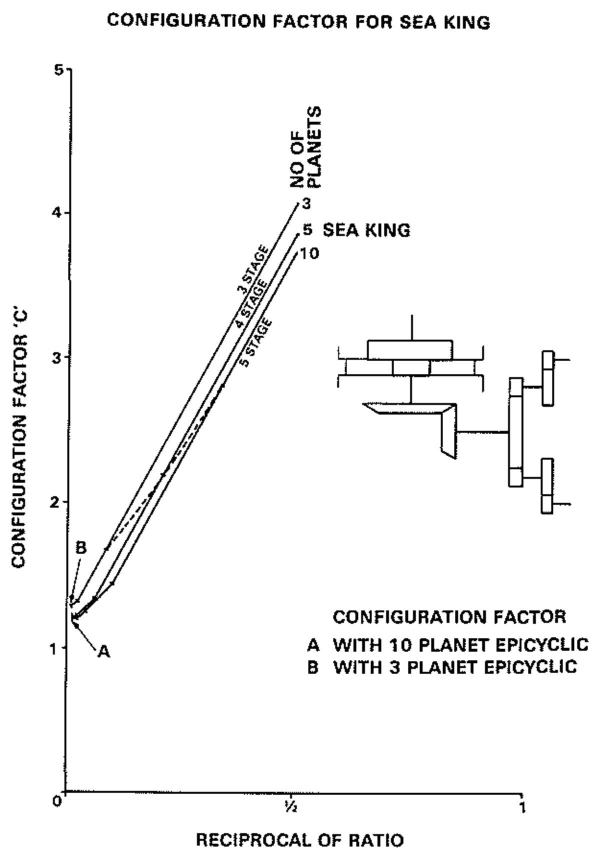



Fig. 4

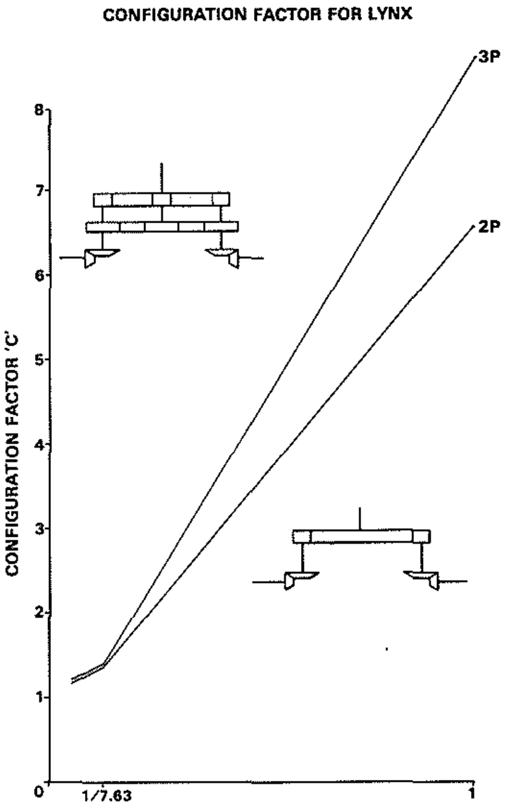



Fig. 5

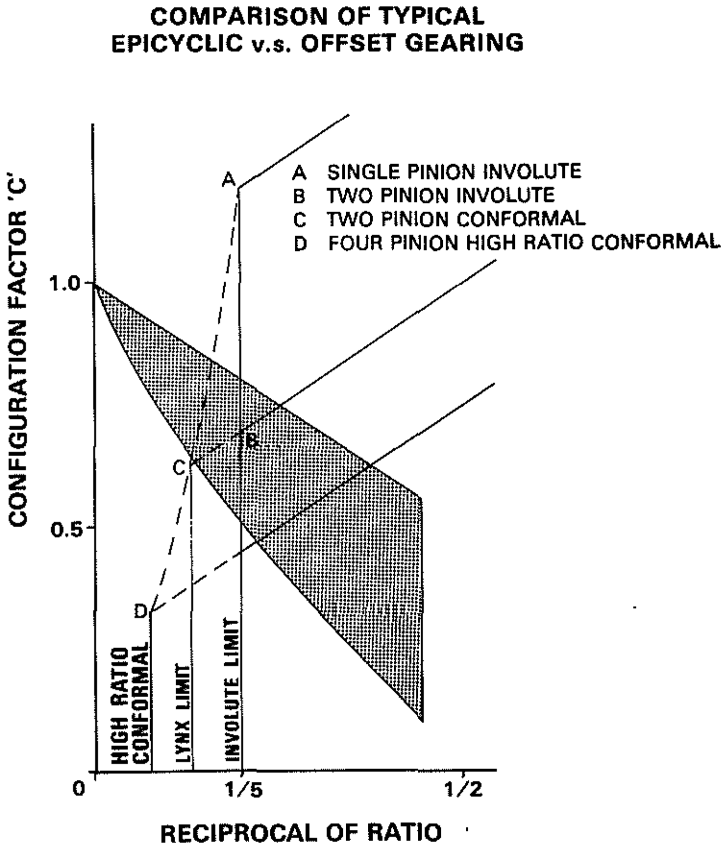



Fig. 6

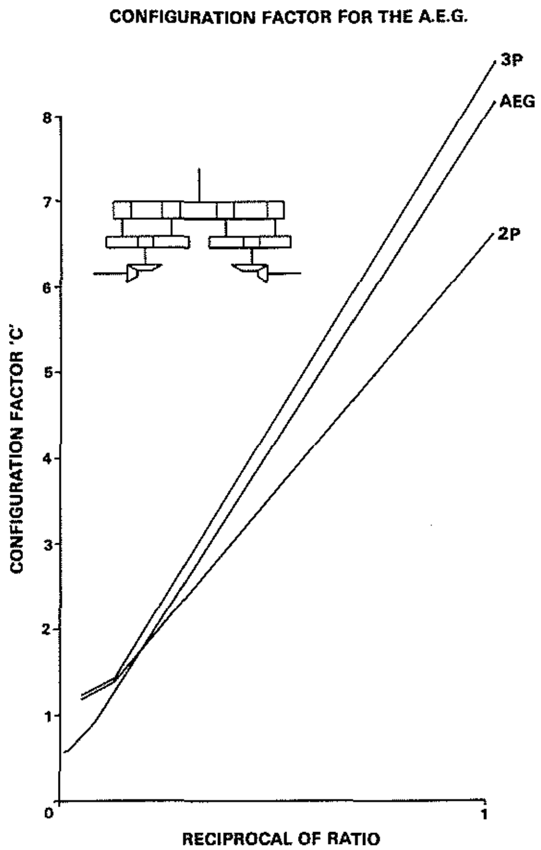
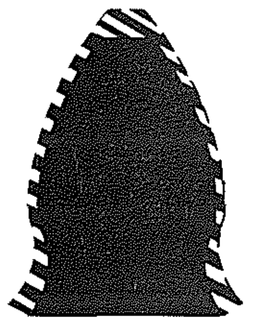
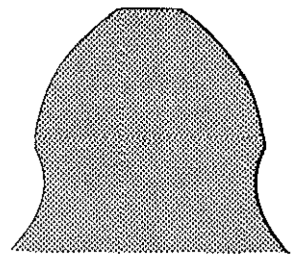



Fig. 9

Fig. 7



9 Tooth Involute Pinion Tooth Form



9 Tooth Conformal Pinion Tooth Form

Fig. 8

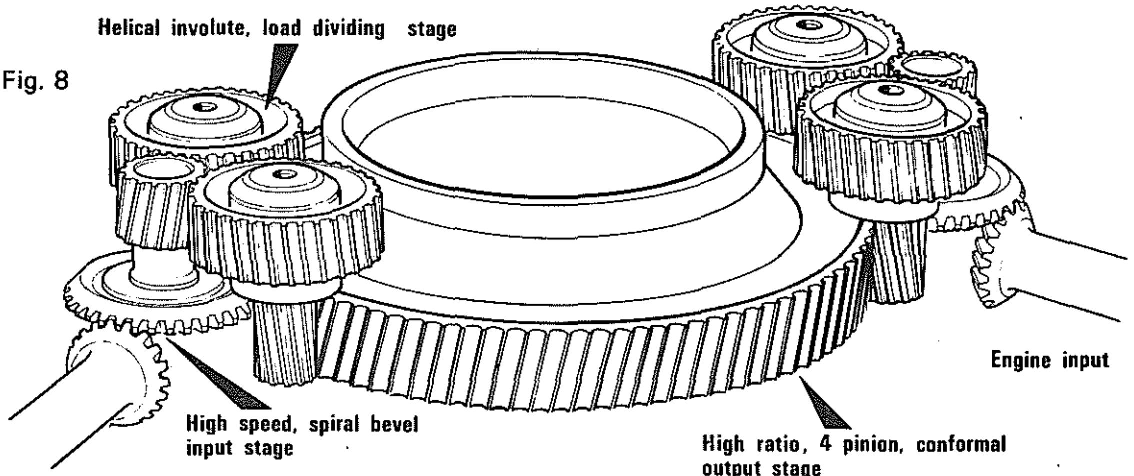
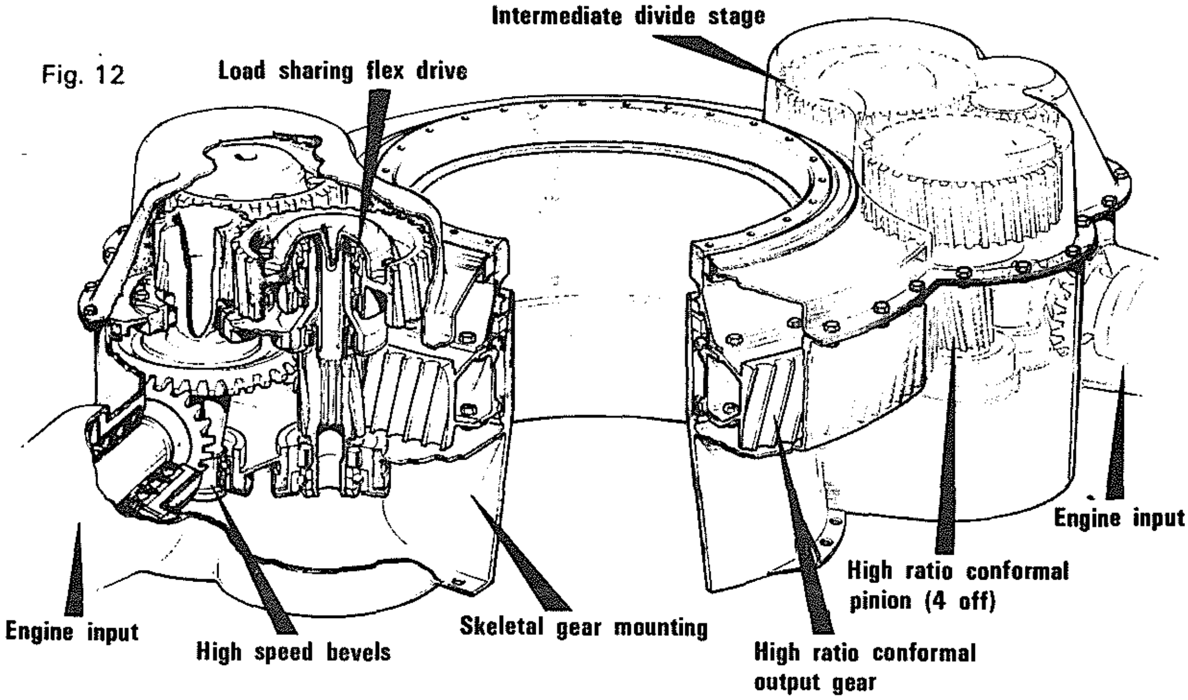



Fig. 12



**BASIC AND DEFLECTED VIEW OF FINITE ELEMENT MODEL OF SEMI-SKELETAL CASE**

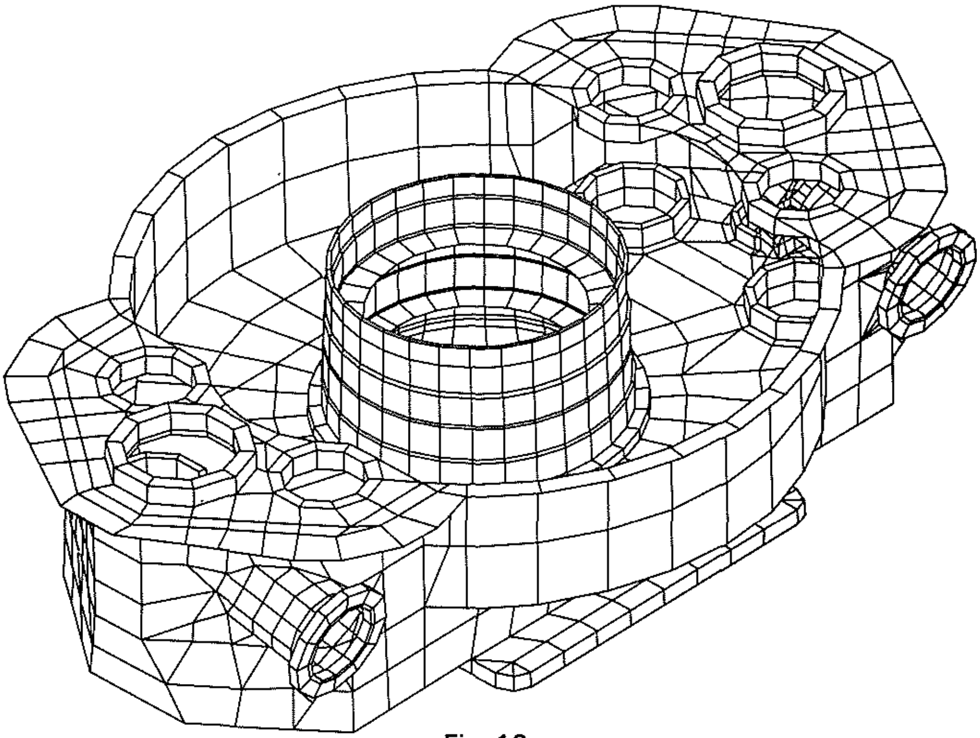


Fig. 10

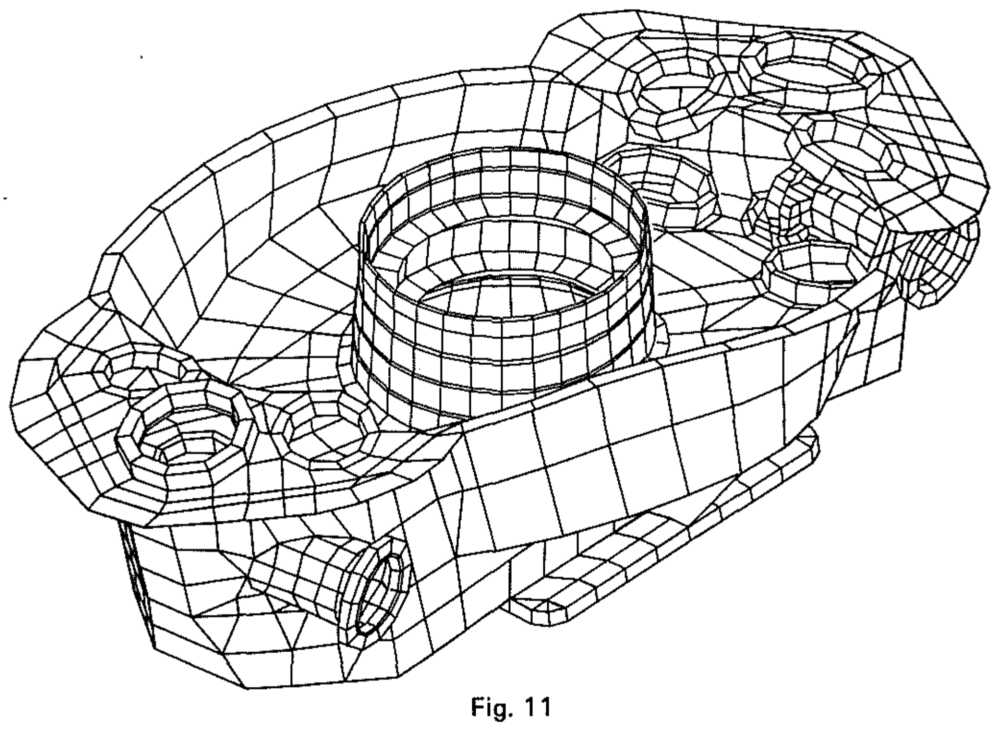
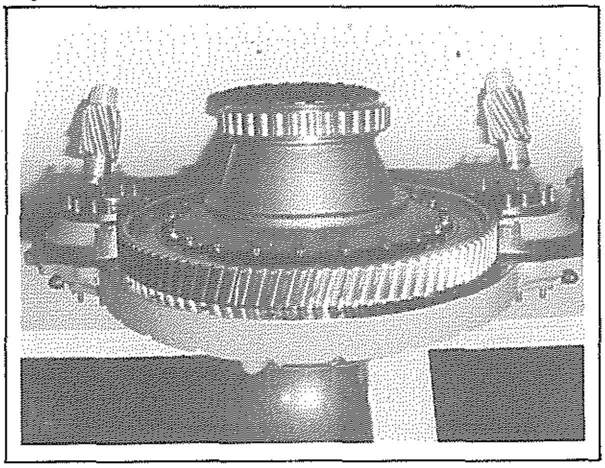


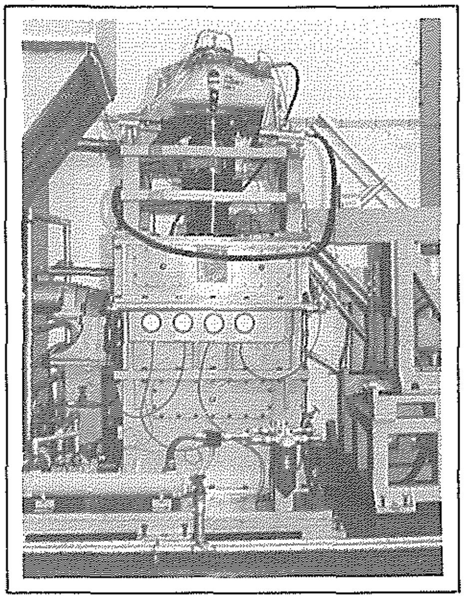
Fig. 11

Fig. 13



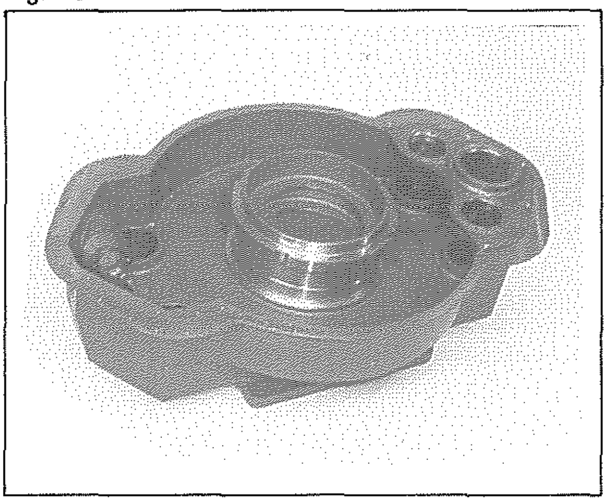
HIGH RATIO CONFORMAL TEST GEARS

Fig. 14



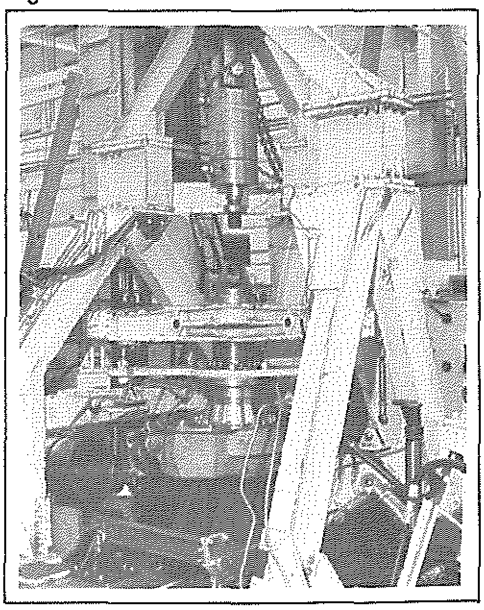
HIGH RATIO CONFORMAL TEST RIG

Fig. 15



SEMI-SKELETAL CASE

Fig. 16



SEMI-SKELETAL CASE AWAITING TEST