THE DESIGN OF THE DRIVE

By

J L NORTON, A M I Mech E

A Meetıng of the Assocıatıon was held at The Royal Aeronautıcal Socıety, 4 Hamılton Place, London, W 1, on Frıday, 5th March, 1954, at 6 p m

In the Chaır

O L L FITZWILLIAMS, ESQ , B A

INTRODUCTION BY THE CHAIRMAN

The Chaırman, ın ıntroducıng the Lecturer, saıd they were to have the pleasure of lıstenıng to Mr NORTON, of the Engıne Dıvısıon of the Brıstol Aeroplane Company, who would talk about the desıgn of helıcopter transmıssıon

Mr NORTON receıved hıs technıcal traınıng at the Central Technıcal College, Bırmıngham, and hıs practıcal engıneerıng traınıng at Alfred Wıseman, Ltd , and varıous branches of what ıs now Bırmıd Industrıes, Ltd After perıods ın dıfferent departments of Norton Motors, Ltd (founded by hıs father), he joıned the desıgn and experımental staff of that firm In 1931 he set up ın busıness on hıs own account as an electrıcal engıneer In 1936 he joıned the Brıstol Aeroplane Co Ltd (Engıne Dıvısıon), and later became Chıef of the Engıne Desıgn Office In 1941 he was appoınted Specıal Projects Engıneer, beıng responsıble for the desıgn and development of power plants and specıal transmıssıons for the less conventıonal types of ınstallatıon These ıncluded the Twın Centaurus Installatıon ın the Brabazon Aırcraft (for hıs paper descrıbıng whıch, he receıved the Thomas Hawkesley Medal from the Instıtutıon of Mechanıcal Engıneers), the Brıstol 171 Mark 1 Helıcopter Installatıon and the Coupled Proteus Installatıon ın the Saunders Roe Prıncess Boat, etc In 1950 he was appoınted Assıstant Chıef Turbıne Development Engıneer (Mechanıcal) but after two years relınquıshed thıs post to take over the newly created post of Chıef Desıgner, Ramjets, whıch posıtıon he stıll holds

https://doi.org/10.1017/S2753447200002171 Published online by Cambridge University Press

MR J L NORTON

When I was asked to prepare a lecture for presentation to your Association, the subject suggested was the design, manufacturing methods and inspection techniques employed in aero-engine practice This, with particular reference to helicopter rotor drives

At the outset, it was plain that such a subject was extremely wide in scope Thus, it became abundantly clear that, to limit the lecture to a reasonable length, the matter would have to be covered in one of two ways it could either be an extended chronicle of generalities, or, it could follow a hypothetical design case from concept to final assembly Although the latter course was regarded as the better, circumstances and the time factor again have imposed a limitation The paper which I am offering, therefore, will be limited to a dissertation on some of the less obvious features in the design of the drive These features usually obtrude themselves at some stage in the evolution of the type of machinery in which we are interested

In general, my remarks will be confined to rotor drive systems which rely upon a rotating source of power input such as that provided by a piston engine or a shaft-turbine engine

Efficiency

There is invariably a discrepancy between engine R P M and rotor R P M This compels the designer to use some form of speed reduction Usually this is effected by the use of gears and advantage is frequently taken of this necessary evil to get round some obstacle, for instance, to turn a corner or to put the engine where it is least in the way (but probably least happy from the cooling or functional point of view, but don't worry about that—the engine man can take it !) It will be realised that the introduction of gears results in a reduction of efficiency Thus, neglecting bearing losses, each stage in a gear train represents a drop in efficiency of the order of 1% Some workers would say $1\frac{1}{2}$% or even 2%, but I do not regard anything in excess of 1%, at full load, as acceptable since better figures can be achieved This efficiency drop inevitably brings in its wake increased fuel consumption, increased cooling problems and increased weight There are ways and means of keeping these objectionable features within acceptable limits and a mitigating feature is that helicopters are normally regraded as short haul vehicles, thus (and I say this with some diffidence) fuel consumption is of less importance than in certain other classes of aircraft Nevertheless, all will agree that it is important to maintain the highest possible stage efficiency and the lowest practicable number of stages

With modern manufacturing techniques there is no need to be hidebound in the matter of stage ratio Fig 1 shows a diagram of a possible gear drive for a large helicopter rotor Disregarding the 1 1 bevel, it includes three stages of reduction, the greatest ratio being 4 7 1 Fig 2 shows an alternative layout comprising two stages with ratios of 8 2 1 and 6 2 1 With the use of bevel gears a major factor inimical to good operation is distortion of the wheel under load In the case of the wheel for a large ratio, it is virtually impossible to provide adequate stiffness without exceeding acceptable weight limits

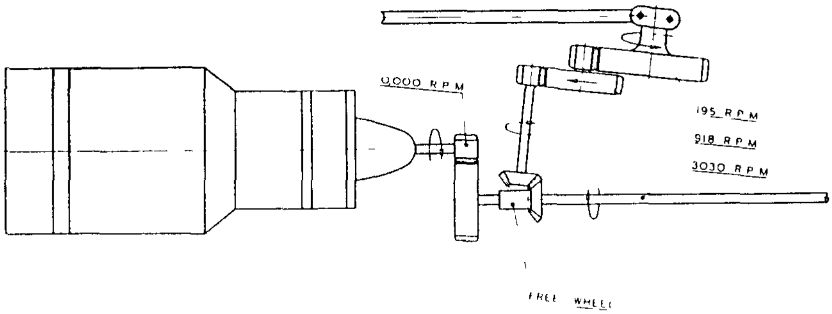

Fig 1 Diagram of 3-stage Reduction Gear train for large Helicopter Rotor

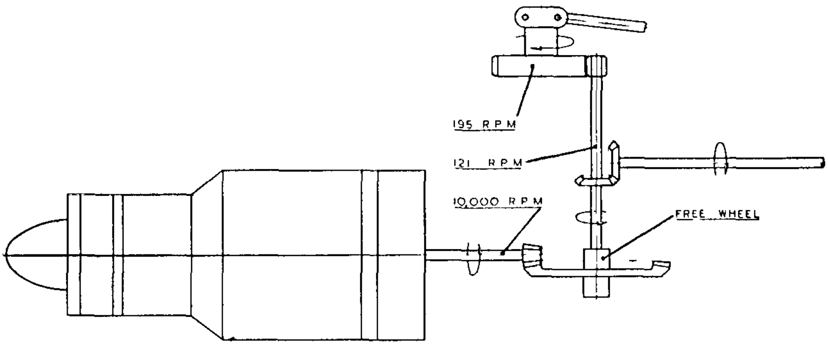

Fig 2 Diagram of 2-stage Reduction Gear train for large Helicopter Rotor

Fig 3 shows how a small thrust pad can be used to take the gear tooth separating load, thus alleviating the duty of the wheel web In addition to relieving the load on the wheel, this stratagem also localises the stresses in the supporting structure thus again improving tooth meshing conditions

While on the subject of gears, it is obvious that the gear teeth themselves must be well designed so that the design stresses are not excessive What is perhaps less obvious is that design stresses can be exceeded by tooth distortion under load, by wheel rim deflection and by manufacturing errors The normal procedure in connection with tooth profile is to apply suitable correction in the manufacturing stages, usually when finish grinding the teeth This takes the form of applying tip and/or root relief (Figs 4, 5, 6 and 7)

As regards wheel rim deflection the answer to the problem appears to be simply a matter of adding material to the rim But there is usually more to it than that ! This is necessary in certain cases, but in others the design of a pair of gears can be made complementary in the deflection sense In other words, an essentially stiff portion of the driving wheel would mate with the less stiff portion of the driven wheel In this way some measure of load spreading can be obtained (Fig 8)

Allied with wheel rim stiffness is the problem of wheel web strength Holes are often drilled in the web in the interest of weight saving That this must be done with extreme caution can be seen from Fig 9

GEAR SUPPORTS

All too frequently one finds that excellently designed gears are supported in an indifferent manner The term "support" includes the shaft, the bearings and the casing or supporting structure There is a component of load trying to separate the gears, and if the supports are insufficiently stiff,

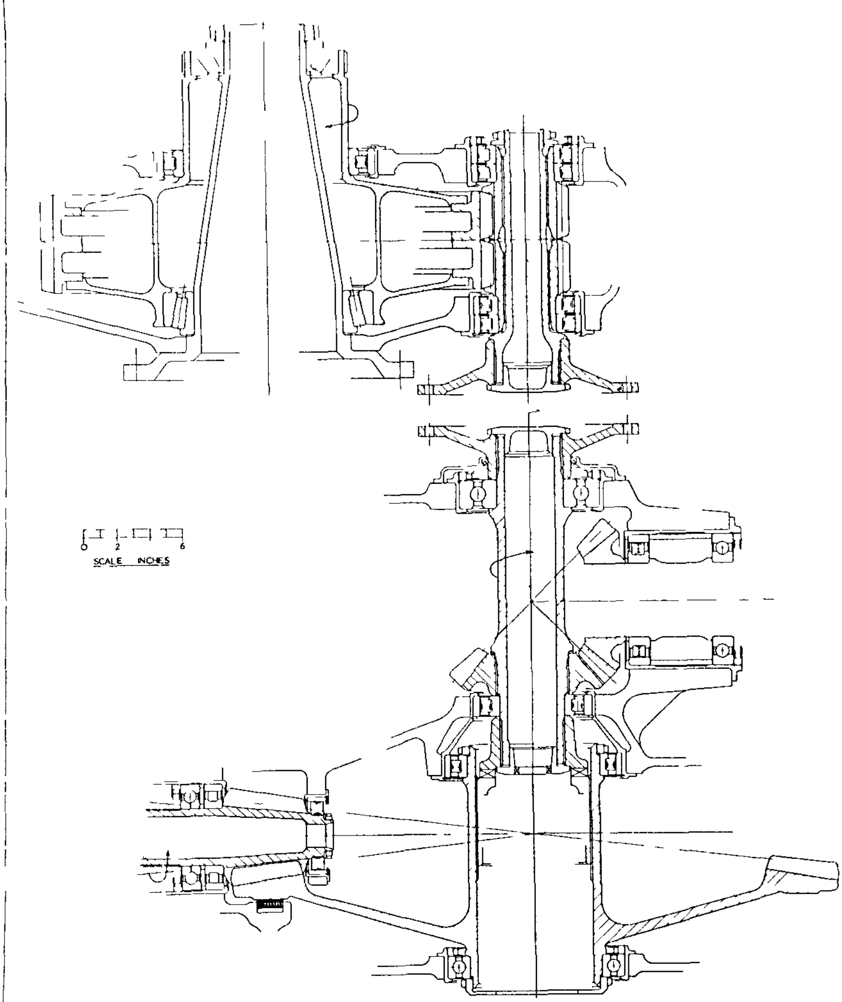

Fig 3 Preliminary layout of 8 2 1 *Bevel Reduction Gear, using Thrust Pad to support Rim of Large Bevel Wheel*

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the teeth will be forced out of mesh to the point where both bending load and surface stress exceed the design figures Lack of symmetry in the casing layout will aggravate this trouble, since, not only will the pitch surfaces be separated, but one will be skewed relative to the other, thus accentuating the tooth load toward one end of the tooth This form of flexibility will also permit rapid displacements at tooth frequency with corresponding diversion of energy That is to say, with adverse effect on transmission efficiency

In this connection, it is well to remember that deflection is proportional to distance for a given load That is to say, the nearer together the bearings, the stiffer the structure If the design is such that the material of the casing is in compression, so much the better This introduces scope for ingenuity, and internal gears show obvious advantages in this connection (Fig 10)

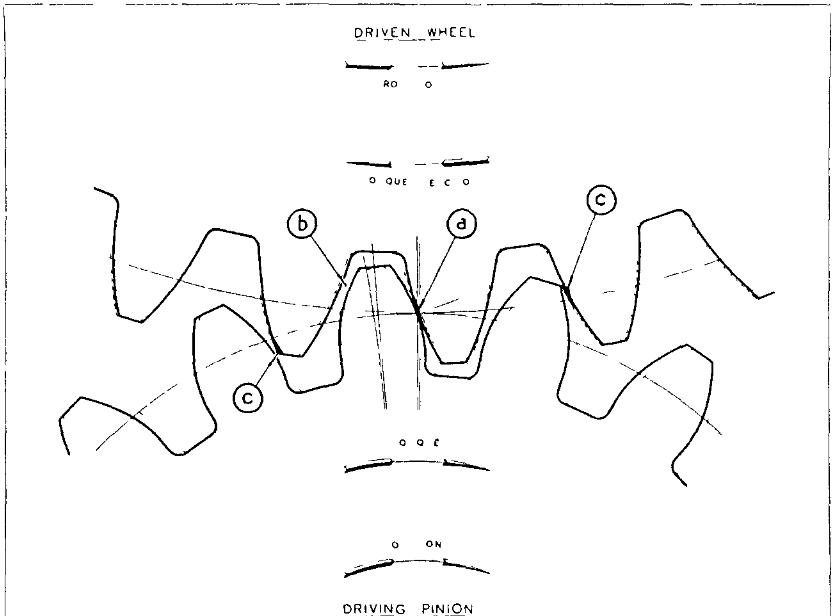

Fig 4 *Diagram demonstrating need for Tip and/or Root relief on Gear Teeth*

OILING

You will note that I didn't say " lubrication "

Oil is circulated round the gear box for the following purposes

(1) To cool the gear teeth
(2) To lubricate the gears
(3) To lubricate the bearings
(4) To cool the bearings

In days gone by, oil was specifically used for lubrication only In these days of compact design and intense loads, I maintain that if cooling is properly done then lubrication of gears looks after itself

It is essential to use oil sparingly but to make good use of it A considerable amount of gear trouble is due to the use of excessive amounts of oil, and a lot of efficiency is lost for the same reason (Figs 11 and 12) As mating

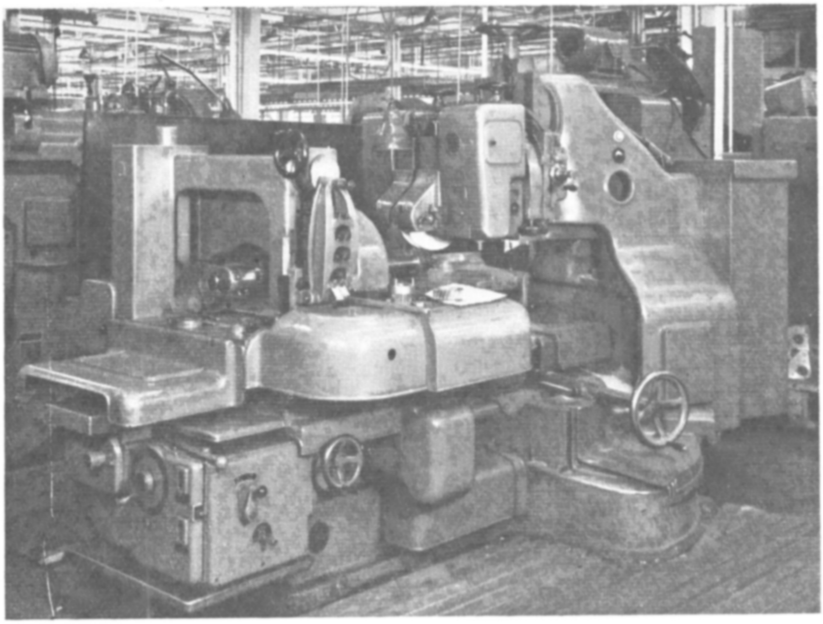

Fig 5 MAAG Helical Gear Tooth Grinding Machine (Type H S S 60)

gears rotate, a pair of teeth come into contact and move on One tooth transmits its energy to the other in the process but some of this energy is not passed on as mechanical work It is converted to heat, the heat of inefficiency Thus each tooth takes away, on and near its surface, a charge of heat Now the best way to dispose of that heat charge is to literally wash it away by means of a jet of oil and to do this as soon as possible after the tooth emerges from the mating tooth space—before it can soak into the body of the gear Nature assists in this operation since the receding tooth leaves a depression or partial vacuum in the space between the mating teeth, and the resulting in-rush of air helps the oil to the areas not directly sprayed by the jets Nevertheless, oil jets of high velocity are usually necessary and these need to be located as close as possible to the disengaging teeth

Certain engineers of my acquaintance regard this principle as somewhat controversial, but it is not so different from that pursued every day in every heat treatment shop Who would dream of quenching a component, or a tool *before* heating ?

An incidental advantage of this principle of cooling is that at these conditions the maximum temperature differential obtains between the cooling oil and surface to be cooled , therefore, optimum conditions exist for effective heat transfer

I referred to this as an incidental advantage, it is an important one nevertheless and enables us to reduce the quantity or rate of oil circulation to a minimum, for we are transferring a given quantity of heat at a higher temperature differential The higher oil temperature also promotes more thorough scrubbing of the walls of the casing, again helping the transfer of heat a further step towards the ubiquitous heat-sink, the atmosphere

Your gear tooth is now cool and it bears a film of oil on its surface If, then, the tooth speed is not too high and, assuming the use of the right oil, that film will provide adequate lubrication when next the tooth meets a mate

To exploit the possibilities of this system to the full, thorough scavenging of the oil is necessary from the casing With good general design, quantities of un-controlled oil splashing around inside the gear casing do nothin g to assist in lubrication, and result in a temperature rise due to " churning" or even to troubles of a more serious nature

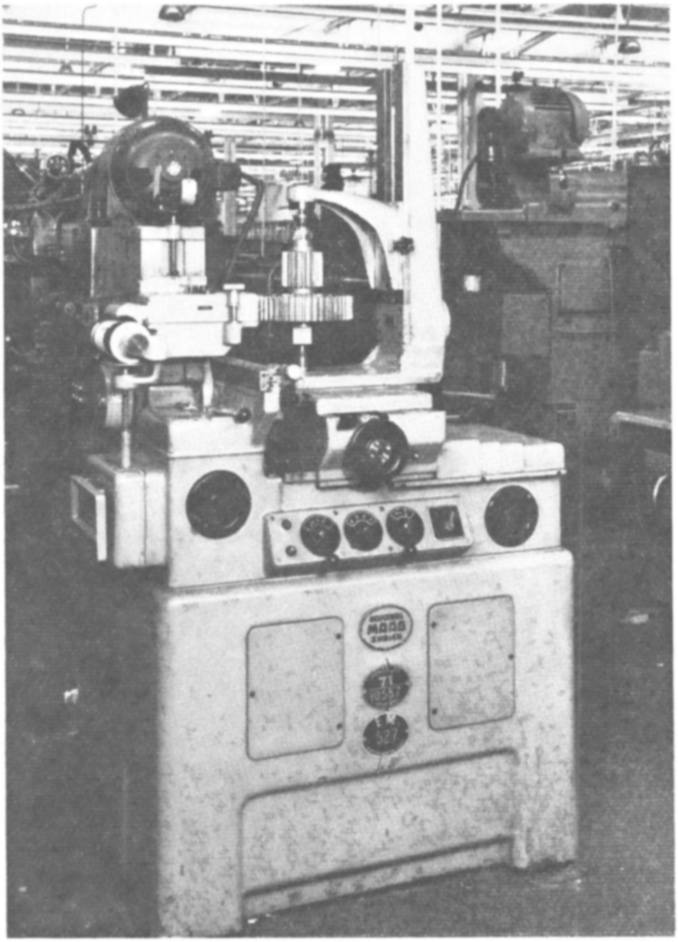

Fig 6 MAAG Gear Tooth Profile Checking Machine (Type P H 60)

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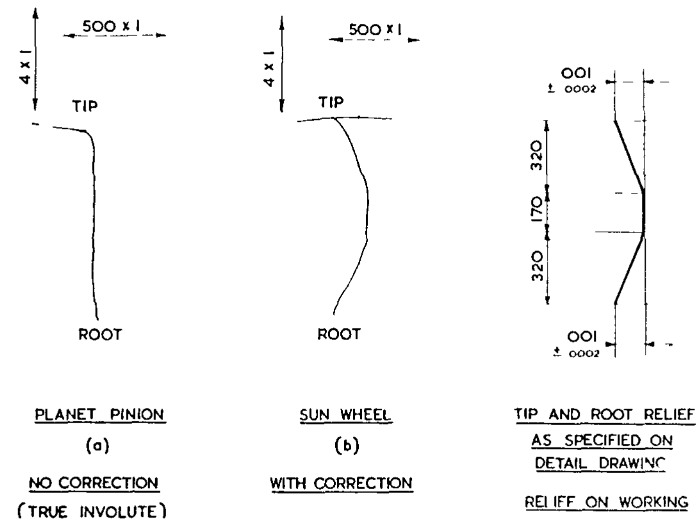

Fig 7 Specimen diagrams produced by MAAG Gear Tooth Profile

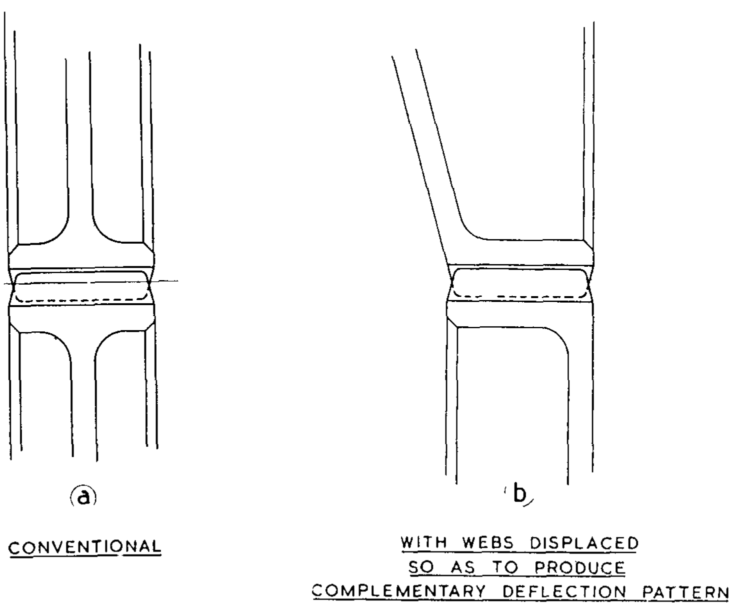

Fig 8 Gear Wheel Design

Fig 13 shows the cluster of jets used for cooling one of the main pairs of gears in the Brabazon reduction gear It will be noted that a scoop is so situated as to collect oil thrown back from the gear teeth This then passes, via suitable drillings, to the pinion shaft bearings

Oil injected between the teeth on the engaging side can cause considerable trouble (Fig 12) The more thoroughly it is injected the more trouble it will cause Pumping action between the teeth expels whatever is there, presumably air, and this air stream opposes the entry of oil If or when one becomes aware of this, one's natural reaction is to move the jets in closer and to increase the oil pressure The effect of these changes is that each engagement is forced to swallow a " gulp " of cool oil and it promptly tries to compress it Quite a big percentage of the " gulp " is expelled from the ends of the tooth space, but, at the speeds normally employed in aero-engine design, incredibly high oil pressures can nevertheless be developed These produce three evils

(1) Efficiency is depressed and therefore temperature rises
(2) Tooth pressures and bearing loads and, therefore, deflections are increased
(3) Torsional and linear vibrations at tooth frequencies are introduced or considerably increased in amplitude These can prove to be very destructive but at least they produce noise

A feature arising out of these comments, but which is not peculiar to gears oiled in this way, is that tooth length, by this is meant the ratio of tooth length to depth, should be kept small Naturally there are many factors affecting this but proportions of 3 to 4 times tooth depth are preferred for most jobs, and can generally be obtained By this means danger through oil trapping troubles is kept small , moreover, there is a good chance of every increment of tooth length contributing its quota of work

Vibration

The calculation of basic modes of torsional vibration is normally a straightforward mathematical process It is, however, virtually impossible to predict the degree of damping in the system and its effect upon amplitudes Also whilst the designer can usually skirt round known major resonance problems it is not unknown to find forced vibrations at comparatively high frequency in a gear driven system The source of these vibrations is often traceable to imperfect gear tooth meshing In piston engine installations, crankshaft speeds are low, therefore the frequency of tooth impact is consequently not very high Now that turbine engines are coming into the picture for helicopter propulsion, frequencies well up in the audio region can be produced and considerable quantities of energy must often be dissipated as a result Helical teeth offer promise of a cure for this evil by virtue of tooth overlap The use of either form of helical gear will result in reduced vibration, and it is popularly supposed that the double helical is better than the single because of the internal cancellation of end thrust For applications other than for high-speed, high-duty, I think I agree, but for difficult aero-engine engineering jobs the single helical can show to advantage apart from ease of manufacture The result of the helical line of engagement is that, in addition to the tangential force, an axial component is present and this compels the use of a thrust bearing This in itself is not so bad since the end load need not be great In any case, it is always a good thing to know in which direction thrust operates and to design accordingly The important feature is that, in addition to a steady axial component due to the mean torque, there is a superimposed fluctuating load due to resolution of the torsional vibration forces Inherent elasticity, however small, permits the casings and supports to respond to these forces Since the two mating gears react in opposite directions some rubbing occurs between the teeth with consequent damping of these high frequency vibrations (Fig 14)

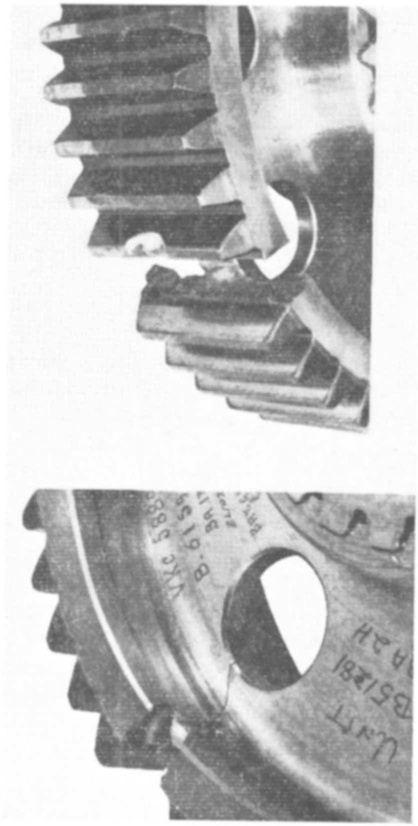

Fig 9 Fatigue Failure of Gear through Drilled Web and insufficiently Stiff Rim

Transmission Shafts

One primary consideration in the design of transmission shafts is to maintain adequate stiffness to prevent whirl The stiffnesses to watch include those of the shaft supports Usually, if this requirement is met, the shaft is adequately strong to transmit the drive, but careful optimization will produce a shaft in which the torque stresses are reasonably high and the weight, therefore, low A factor which can be dangerous, however, is to permit induced torsional vibration in that part of the shaft between two torsionally flexible couplings Conditions can arise in which a torsional excitation, applied through either the driving or the driven coupling, can produce high amplitudes of vibration in the intermediate shaft This can cause trouble with the couplings and even with gears, etc I have had experience of coupling failures which were positively traced to this cause I have also had experience of other failures for which no alternative explanation could be found The cure in such cases consisted of replacing one of the flexible couplings by a Hooke's type joint If the vibration characteristics of the system are understood, however, this form of trouble can be avoided in the design stage

The judicious use of rubber in a shaft coupling is, or can be beneficial in some cases The reasons include the following

(1) The elimination of at least one lubrication point

(2) Introduction of torsional flexibility This provides some measure

of frequency control, while the rubber effects a limited degree of damping

(3) A degree of axial flexibility is provided

This axial flexibility makes the use of a sliding spline unnecessary In fact its use could be dangerous At the best of times a sliding spline will only slide when the axial load exceeds the product of torque times co-efficient of friction over nominal spline diameter In practice this usually means that the spline slides only when little or no power is being transmitted As soon as the torque builds up, the spline resists any tendency to slide (due to frame deflection, or the like), and this can result in bearings, etc, being overloaded If the sliding joint has any clearance, as it must if it is to work at all, even when static, the individual male and female splines can in extreme cases, " walk " up each other, so as to speak, the resulting displacement producing phenomenal end loads These loads can be enough to disrupt a rubber type flexible joint possibly with catastrophic results

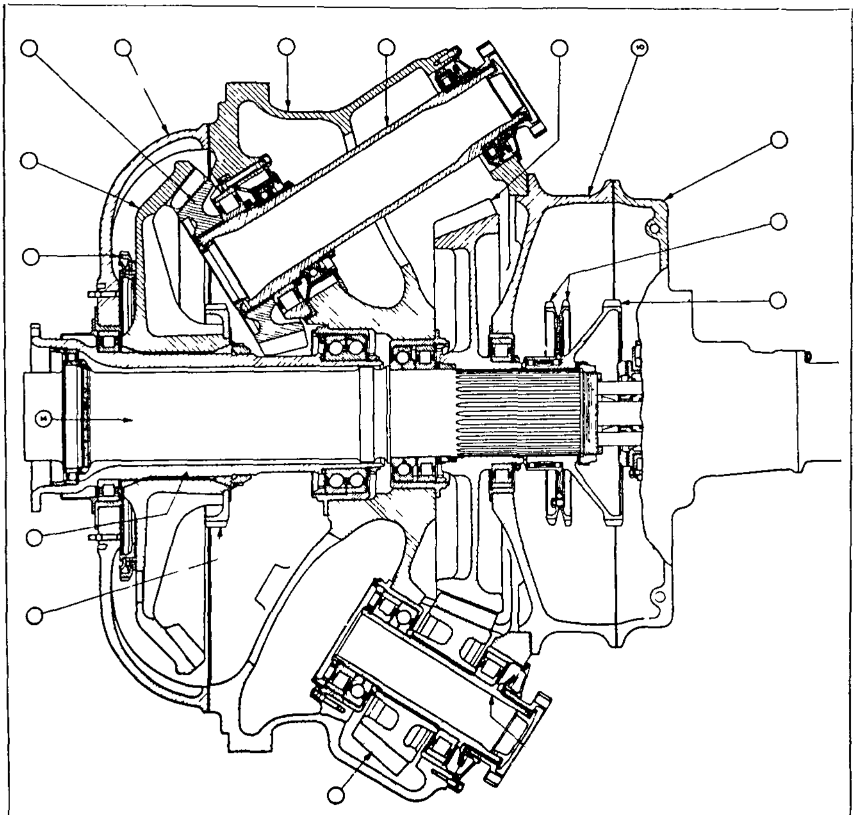

Fig 10 Horizontal section of Dual Reduction Gear

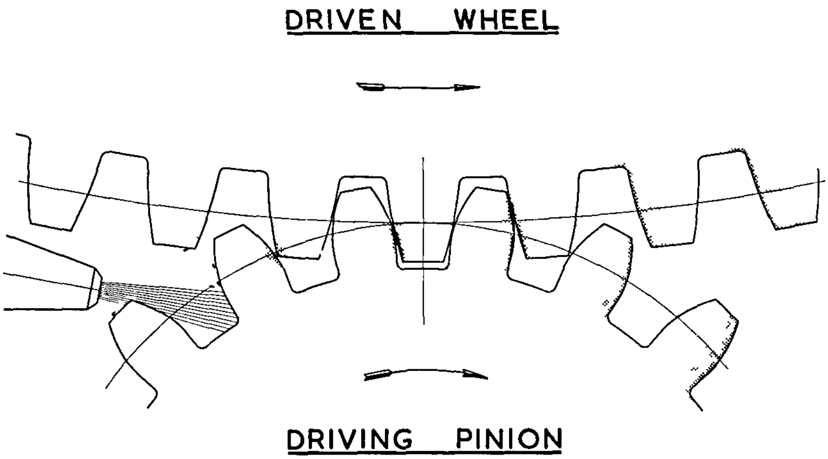

Fig 11 *Diagram showing areas of heating of gear teeth*

Friction

The question is sometimes asked " Is friction a friend or an enemy ? " I don't propose to attempt to answer that question, but I do propose to bring friction into the limelight a little In a simple bolted-up joint we rely on friction to prevent the nuts running back It is true that we lock the nuts but I submit that if the joint be correctly designed, the nut will remain tight even when no locking is provided A locking device will probably be used, however, firstly to satisfy some regulation or other, and secondly to give some peace of mind !

Many engineers would firmly deprecate the use of a friction drive, but

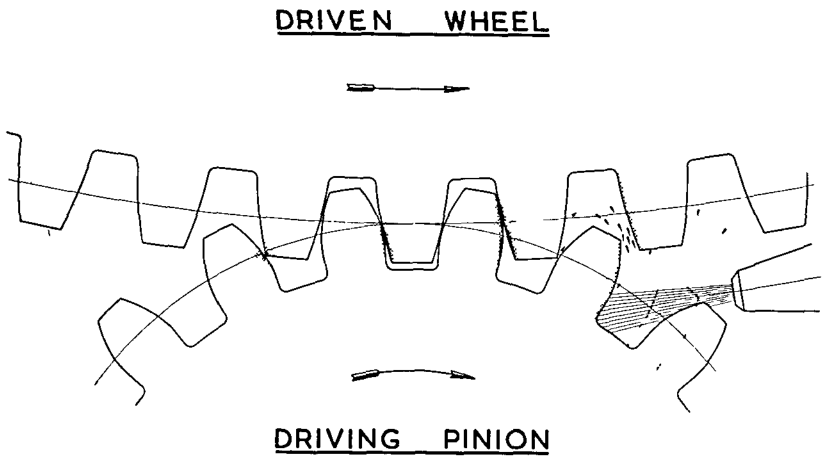

Fig 12 *Diagram of Tooth Cooling by means of Oil Jets*

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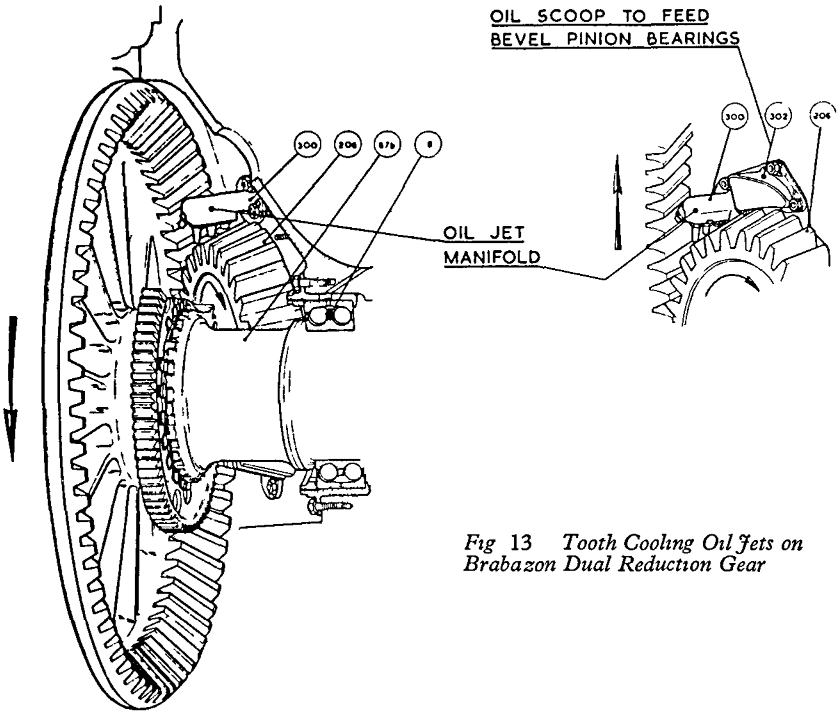


Fig 13 Tooth Cooling Oil Jets on Brabazon Dual Reduction Gear

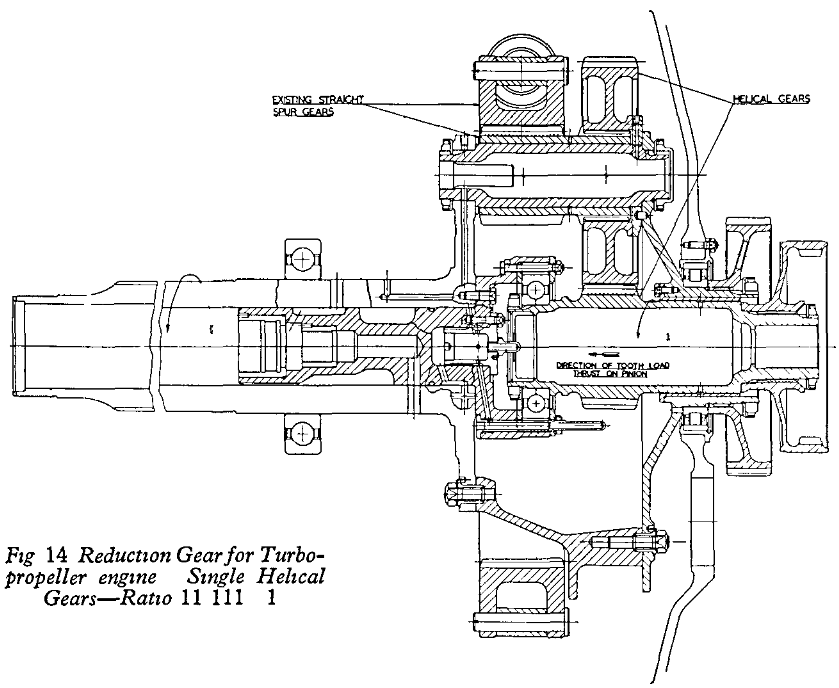
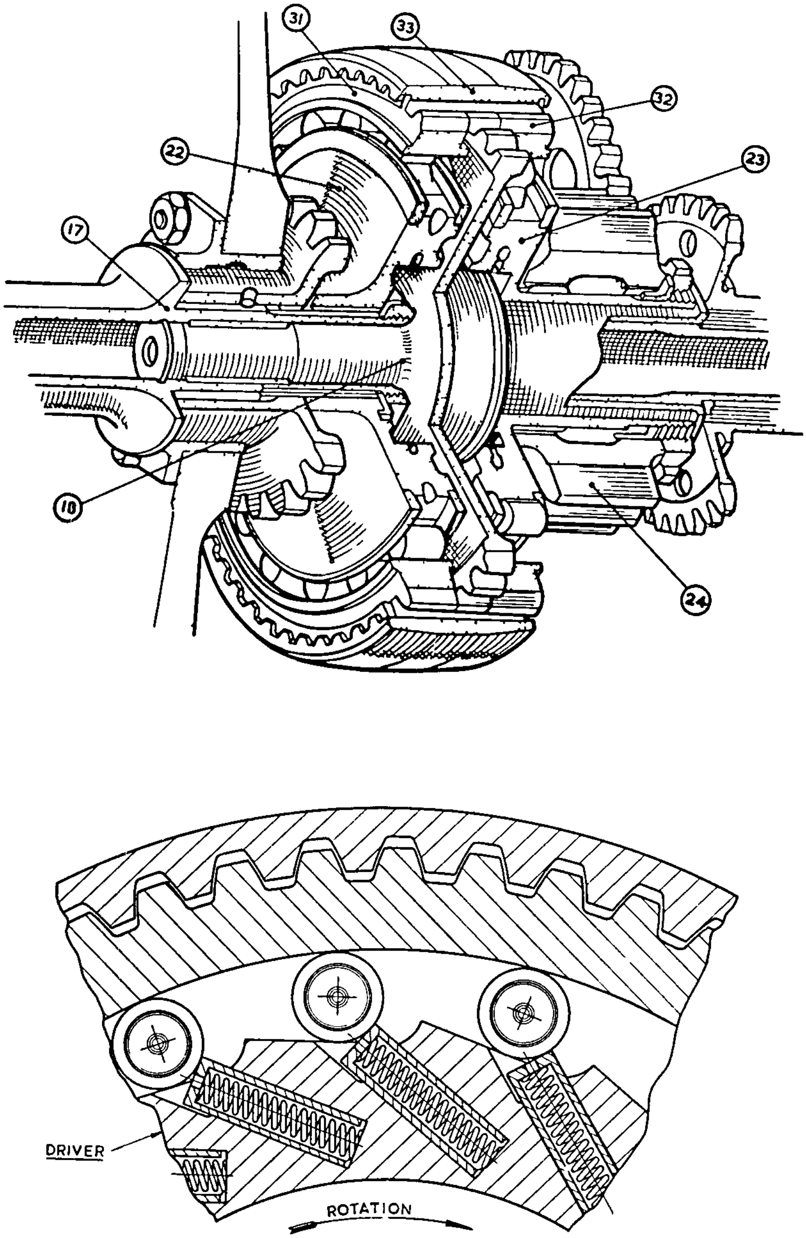


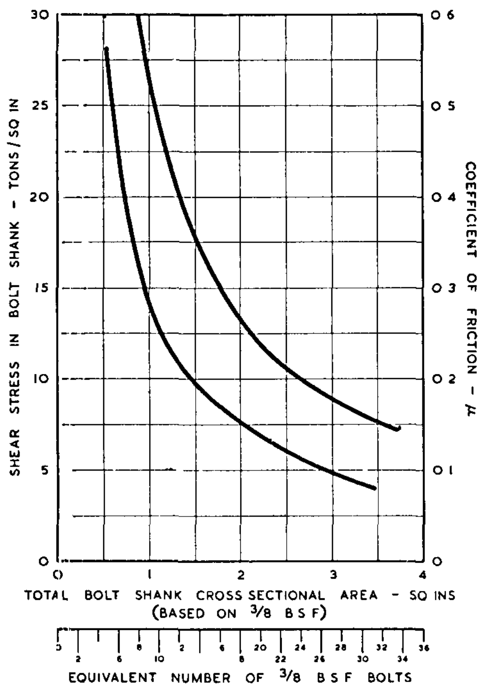
Fig 14 Reduction Gear for Turbo-propeller engine Single Helical Gears—Ratio 11 111 1



DOUBLE FREE WHEEL ASSEMBLY

FIG 15

upon analysis, one finds that there are many points in a drive where the medium of torque transmission is friction and nothing else An obvious case is the jamming-roller type of free-wheel, where the drive is transmitted through friction, between the rollers and the driven track (Fig 15) A less obvious case is where torque reaction is transmitted from one section of, say, a reduction gear casing to another Some form of spigot or dowel location is usually provided to ensure alignment of bearings, but this is rarely sufficiently robust to accommodate the torque loading, friction at the joint does this



Now, consider the case of a simple flanged joint in a power transmission shaft (Fig 16) Fitted bolts or dowels are provided, nominally to transmit maximum torque through the medium of shear stress Now, in order to develop shear stress, compression must be produced in and near the surface of the material of the contacting components and since Hooke's law has to be satisfied, this means that there must be strain Strain implies movement, so, to develop shear stress in the dowels, or the bolts, one flange must rotate relative to the other Were this relative rotation to be permitted, even if only of a microscopic order, the coupling would suffer from

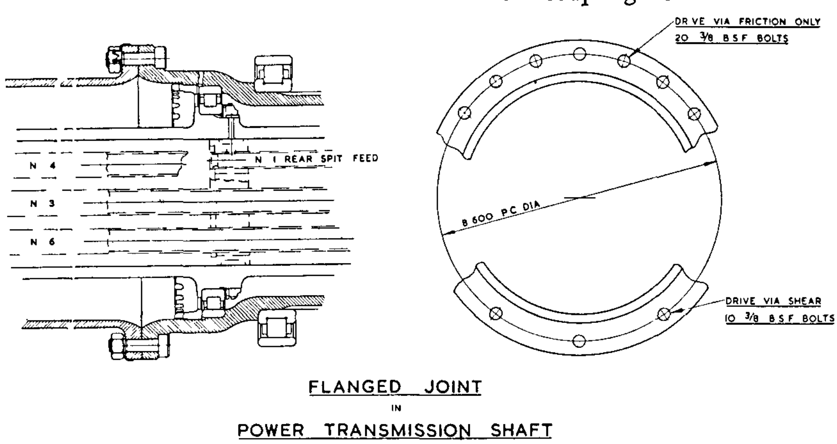


Fig 16 Flanged joint in power transmission Shaft, Comparison of "Shear" and "Friction" drives

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frettingtroubles and would probably fail eventually, as a result of fatigue Normal practice, of course, is to tighten the coupling bolts sufficiently to prevent movement I submit then that the drive is being transmitted entirely by friction So why not design in the first place on the basis of a friction drive ? Salve your conscience, certainly, by the use of dowels—they will certainly ensure alignment on assembly !

While on the subject of friction why despise the simple belt drive for low power transmission ? In multiple form it is reliable and light, it will not transmit high frequency vibrations and will damp low frequency vibrations Above all, it is cheap !

Manufacturing Tolerances

In liaison with manufacturers of things other than engines (I'm not mentioning airframes), I have been pleasantly surprised at the readiness with which initial agreement is reached on close tolerances, such as affect dimensions jointly concerning them and the engine manufacturer In one well-remembered case, the promise was made that the dimension would be " spot on " Well, the engine didn't fit, and in the course of the " inquest " it emerged that the vital dimensions were fixed by means of an ordinary steel rule, it being contended that on such a large dimension extreme accuracy was not essential In this case, the result of their departure from the engine makers' stipulation was an error in shaft alignment of $\frac{5}{16}''$!

Tolerances might be regarded as a nuisance, but a full appreciation of dimensional problems involves a painstaking investigation of the tolerances affecting the assembly This is the best road, in fact the only road to accuracy There is no such thing in our industry as " spot on " accuracy and it is difficult to over-emphasise the importance, firstly, of determining suitable tolerances, and secondly, of ensuring that components or assemblies are produced to within those tolerances

Conclusion

Now, for the purposes of this Paper, I have studiously avoided reference to such technical features of the drive as are covered in the text book I have confined my remarks mainly to some of the more advanced mechanical problems with which I have been confronted from time to time In this particular field of design there is almost unlimited scope for ingenuity, but the enthusiasm of the genius must be tempered by a thorough appreciation of the mechanical problems involved However, to achieve real progress it is necessary to push out beyond the fields of existing engineering accomplishment and this must be done with courage

The technical requirements can always be resolved mathematically and in the laboratory, but the ultimate answer will only be provided by practical test Between these two aspects there will be a gap The size of the gap will be directly related to the magnitude of our courage In the course of time this gap will doubtless be bridged but, to hasten the process it is essential to employ the right type of brain one who can devise and create one who was born with a spark of inventive ability He may not possess the highest of academic qualifications but he has a predisposition to follow a sound course—a predisposition which almost amounts to intuition