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DIESEL ENGINE NOISE UNDER STEADY STATE AND TRANSIENT CONDITIONS

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RICARDO CONSULTING ENGINEERS

INTRODUCTION

Continuing development of automotive diesel engines requires an increasingly detailed understanding of the processes occurring within the engine. This paper reviews some of the techniques in use and being developed in the authors' company and results which have been obtained through their use.

The reduction of engine noise cannot be studied in isolation. Both the combustion process and the total weight of the power unit are under extreme limitations imposed by requirements for maximum fuel economy and minimum exhaust emissions. Strategies therefore must be developed which will be successful in reducing noise levels whilst satisfying all the many other limitations. In order to achieve this result, development techniques must be available to readily provide understanding and to quantify the benefit to be obtained from a proposed modification.

The end point of most work on vehicle exterior noise reduction is to comply with legislation, which assesses the vehicle noise radiation under acceleration conditions. This paper deals with some work on engine noise radiation under such transient conditions and also indicates some areas of work, involving the application of digital techniques.

1. VEHICLE NOISE GENERATION

The noise radiated by the whole vehicle is comprised of many parts - figure 1 gives an outline of how these are related. Techniques of vehicle noise reduction may be divided into categories dependent on the area of action within the model. Thus fundamental or primary reductions can be obtained through modification of the combustion process, further reductions obtained by modifications to existing components or palliative cures adopted such as shields or enclosures (1)*.

In most current well-developed trucks it is the engine sources which represent the major contribution to the overall vehicle noise level. This in part is due to the more difficult challenge presented by the engine and also due to time and cost considerations when major changes are required.

2. ENGINE NOISE GENERATION

Two primary sources exist in the engine - combustion and mechanical. Both of these will be reduced by either palliative treatments or by structural modifications to reduce the response of the engine to internally generated forces. It is clear that a knowledge of which of these sources is dominant is

*Numbers in parentheses designate references at end of paper.

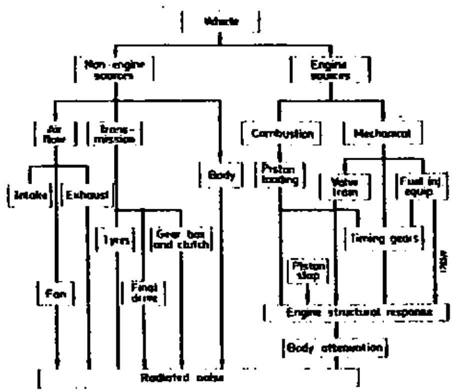

Fig. 1: Vehicle Noise Sources

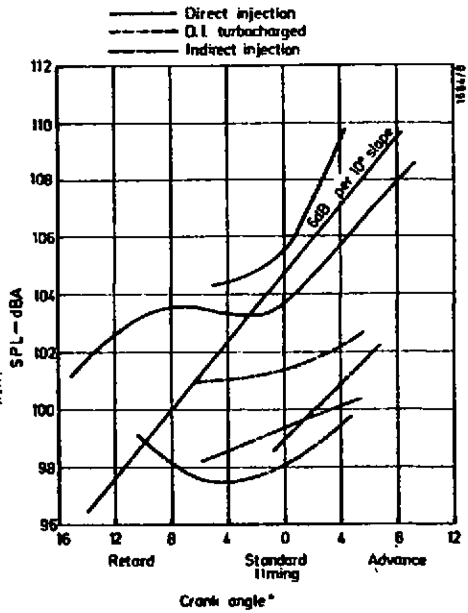

Fig. 2: Effects of Injection Timing

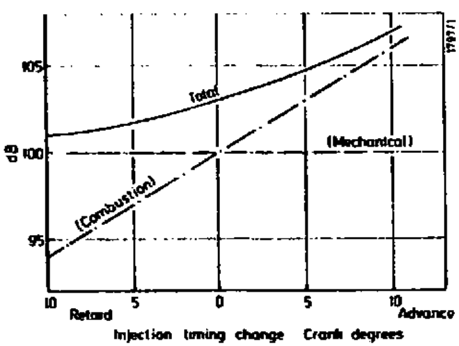

Fig. 3: Engine Noise Separation

of great value when considering action at the primary level.

A qualitative understanding can be obtained by observation of noise levels radiated by the engine as the injection timing is varied. If large changes are observed then it is likely that the combustion process is dominant; conversely if only small changes result then mechanical forces are probably dominant. In a similar manner the rate of change of noise level with speed can be an indicator of the dominant mechanism to an experienced assessor. A quantitative method to identify these components has been developed and is explained below.

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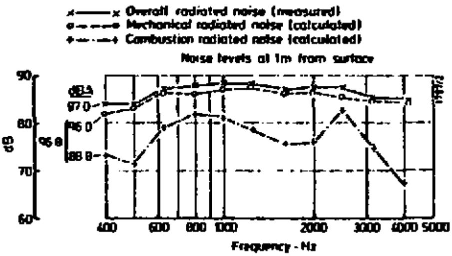

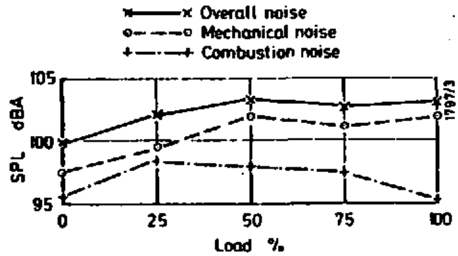

Fig. 4: Separation Frequency Spectra   Fig. 5: Separation Over Load Range

3. SEPARATION OF COMBUSTION FROM MECHANICAL NOISE

If the structural response of the engine is assumed to be linear (and all work to date confirms that this is so) then the two force inputs into the structure can be summed linearly to produce the response. It is relatively straight forward to measure the combustion excitation with a high quality pressure transducer in the cylinder head, but essentially impossible to measure the mechanical excitation since this is dispersed at many points. The response of interest is the radiated noise of the engine. By modifying the combustion input - most easily by changing the injection timing in diesel engines - the change in both the combustion force input and the response can be measured. There will not be a linear relationship between these two since the mechanical input has been included. The susceptibility of radiated noise is markedly different between various forms of diesel combustion systems as is shown in figure 2. The variation in characteristic may be due either to different susceptibilities of the combustion systems to injection timing or to varying levels of mechanical noise in the individual engines or to the combination of both effects. The general, simplified form of relationship is illustrated in figure 3, where a constant level of mechanical noise has been assumed, together with equal contributions from combustion and mechanical excitation at the standard injection timing.

Early work in this area (2,3,4) used large values of injection advance in an attempt to bring the combustion noise well above the mechanical noise 'floor' and hence simplify calculation of the engine response. If the latter can be obtained simply by subtraction of the measured radiated noise from the measured in-cylinder combustion noise, then calculations back at normal timings are relatively simple. In practice, this technique has limitations on accuracy due to mechanical noise increasing at the extreme advance conditions as P_{max} increases and in cases where combustion and mechanical noise are nearly equal.

The current technique still assumes that the mechanical noise remains constant over a timing swing, but the swing is reduced to a smaller value and a check on the calculated mechanical noise is also made to pick up any errors in the initial assumption.

Details of the separation technique are given in Appendix I. The results of the application of this technique in third octave frequency bands from 400Hz to

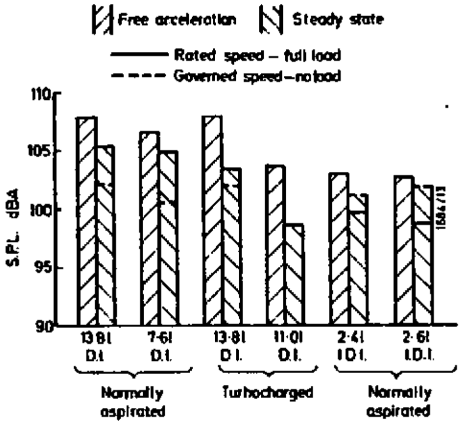

Fig. 6: Effect of Transient Operation

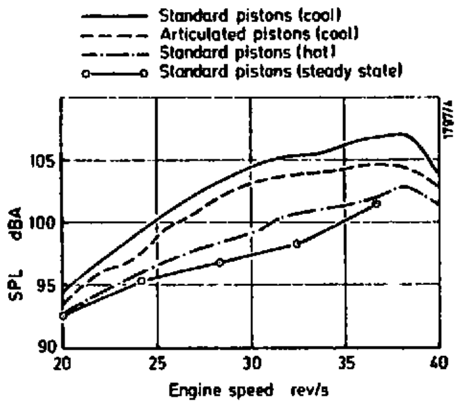

Fig. 7: Noise During Acceleration

4kHz have proved to be adequate for most engine development work. Examples of the results produced by this technique are given in figure 4, for an indirect injection engine and in figure 5 for a larger direct injection engine. These results were obtained from steady state measurements and therefore represent only stabilised conditions.

4. TRANSIENT OPERATION

It has been known for some years how significant the effects of transient operation can be on the radiated engine noise. This is illustrated for a number of engines with different combustion systems in figure 6. When accelerating up to the maximum governed speed the noise levels may increase by 2-3dBA for naturally aspirated engines, or by 5-6dBA for turbocharged engines, above the steady state maximum noise level.

This order of increase is significant, particularly when it is remembered that the legislative test procedure is a transient acceleration. The rate of acceleration depends on the power/weight ratio of the vehicle and the details of the instantaneous torque curve. The rate can be high, however, in European and USA tests, since these use the unladen vehicle for the testing, unlike the Japanese procedure, which uses a fully laden condition.

Attempts to simulate the vehicle acceleration rate on the test bed can become complex, but Ricardo have developed two simple tests which provide some indication of the scope of the problem. Noise measurements are plotted on a high speed X-Y recorder such that the instantaneous noise level can be plotted against engine speed during the transient. The first type of transient (COLD) is run against no load on the dynamometer, starting with an engine speed of say 20 rev/s and the rack control is then quickly moved to full, accelerating the engine to maximum governed speed. This is termed a 'cold' transient, since the engine temperatures at 20 rev/s, no load, are relatively low. The second type of transient (003) is obtained by starting with the engine running at full load at 20 rev/s and, when stable operation is obtained, releasing the brake load instantaneously so that the engine accelerates up to the maximum governed speed. This test is termed a 'hot' transient, since it starts from a high temperature (full load) condition.

An example of the different results obtained from these tests, together with the effects of reduced piston clearance, can be seen in Figure 7.

In order to examine the causes of this increase in radiated noise under transient conditions in more detail, it would be useful to separate the combustion from the mechanically induced noise during the acceleration. This has been achieved and the method used is very similar to that described above for the steady state case and uses some of the data obtained from that calculation.

It is possible, though tedious, to duplicate the entire measurement methodology of running transient accelerations at various timings, but an alternative approach can also be used. This involves measuring the instantaneous combustion excitation during an acceleration transient and re-working the original combustion/mechanical separation calculation. This simplifies the procedure considerably since from the original data for an engine the apparent attenuation of block structure can be derived and then applied to the measured combustion excitation data. This will provide a value for the radiated combustion noise, which when compared with the measured total radiated noise can provide an indication of the mechanical noise.

The validity of using this 'attenuation' method was checked by applying the calculated mean attenuation figures to the steady state data when agreement to within 1dB was obtained for calculation of combustion and mechanical noise.

The major difficulty is thus to obtain a measurement of the combustion noise during the transient.

4.1 Measurement of Transient Combustion Noise

Two methods for measuring the transient combustion noise have been examined, one based on an analogue instrumentation chain, the other totally digital.

With the analogue approach the filtered pressure transducer output of the individual 1/3rd octave band filters was fed to an X-Y recorder during the transient acceleration, and this process repeated for each band from 400 to 4kHz. The time constant used in this processing was IEC Fast. As an example of the output from this, Figure 8 clearly shows the increasing level against speed with the oscillation in level due to individual firing strokes during the acceleration superimposed. The level of interest is the peak of each firing stroke so that in order to smooth the data a curve was fitted to the peaks in each plot. The CPL for any engine speed and any band within the measured range could then be obtained for use in the further calculation. The repeatability of these measurements was ca. 0.5dB.

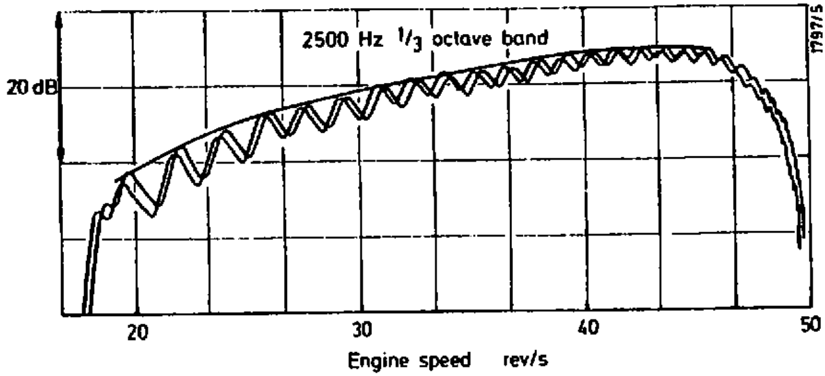

Fig. 8: Cylinder Pressure Level During Acceleration

The digital analysis used an HP 5451C FFT analyser, with the ADC trigger pulses obtained from an engine speed signal via a converter to produce 1024 pulses per engine cycle. In this way truncation errors were avoided whilst maintaining an adequate digitisation of the cylinder pressure signal. In order to obtain a measurement of instantaneous engine speed during the transient acceleration a constant frequency square wave was fed to a second channel of the analyser. Using the 'Throughput' mode of operation of the analyser the input was transferred from the ADC direct to the disc storage in real time.

To avoid aliasing errors a low pass filter was used for the measurements, with

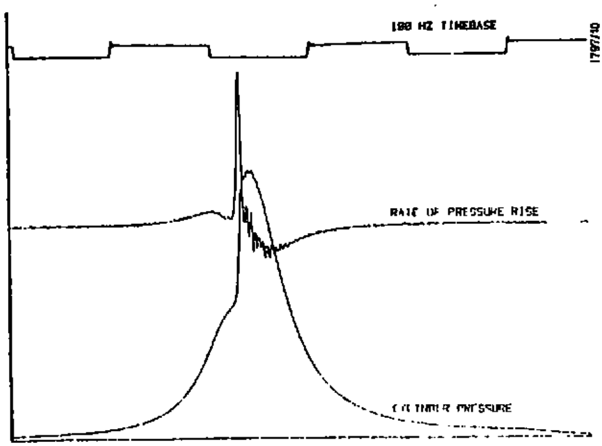

Fig. 9: Cylinder Time History During 'Cold' Acceleration (N.A., D.I., 30 rev/s)

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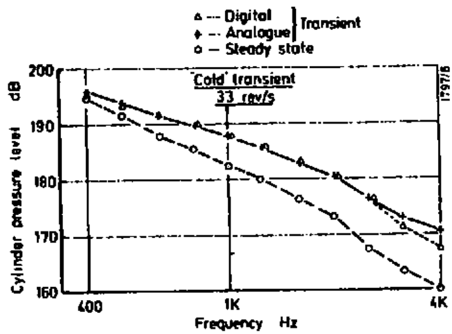

Fig. 10: Cylinder Pressure Spectra Comparison

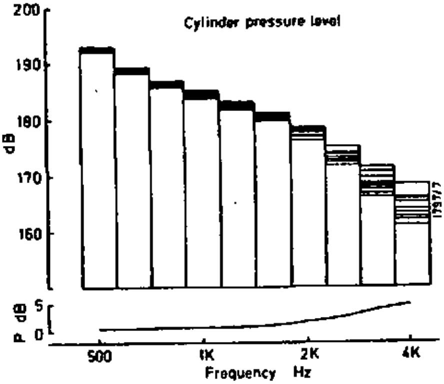

Fig. 11: Variation at Steady State

the roll-over point set to 5kHz. In order to obtain sufficient dynamic range for cylinder pressure spectra a high-pass filter was also used, eliminating signals of less than 250Hz. This latter filter was not used where the rate of pressure rise was to be examined.

Each record of 1024 points represented one complete cycle in the monitored cylinder and could be frequency transformed. From the second channel of data the engine speed (averaged over the cycle) could be found and thus the frequency axis scaled correctly. An example of the data obtained is given in figure 9, where the cylinder pressure diagram and the rate of change of this is shown, together with the 100Hz marker signal. Narrow band data, as obtained from FFT processing, is somewhat too detailed to enable convenient comparisons to be made, so that data was converted to equivalent 1/3rd octave band levels (all cylinder pressure levels are referred to 2 x 10^{-5} Pa, the same reference pressure as noise levels).

A comparison of the two techniques and the agreement between them for the transient cylinder pressure level at one speed in the acceleration is shown in figure 10, both of which are compared with the steady state results for one particular engine. The overall agreement between the two techniques is excellent with some deviation at 3.15 and 4kHz. This latter may be due to repeatability of the engine combustion itself, which is known to have greater variance at higher frequencies. This is shown in figure 11 which compares the frequency spectra of 10 successive cycles of the same engine at 40 rev/s full load. Calculating the dB spread for 68% probability from:

$$P = 20 (\log (\bar{x} + \sigma) - \log (\bar{x} - \sigma))$$

where $\bar{x}$ and σ are the linear mean and standard deviation of the original pressure signal, gives the curve against frequency shown in the lower part of the figure. This confirmed the variability of the higher frequency bands, with a value of 4.8dB at 4kHz, which is in agreement with other published data (5). Although this variation can be averaged out at steady state conditions by the

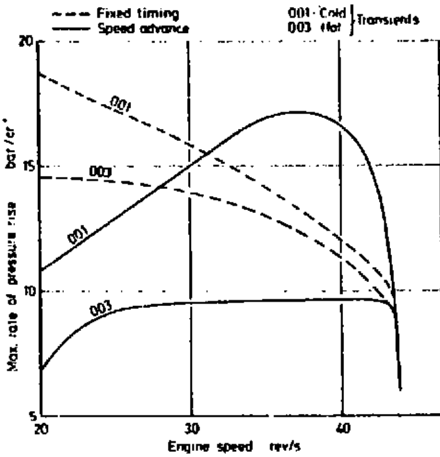

Fig. 12: Effect of Fuel Pump Type

use of longer averaging times, this is clearly not possible for transient studies.

Results obtained using this technique have shown that the original steady-state balance between combustion and mechanical noise is apparently retained during the transient for naturally aspirated engines. Where combustion noise dominates at steady state it also does during the transient. Where an equal balance is found, this is also found under transient conditions - both sources increasing by the same amount. Other work has shown similar results (4). Further work is required on turbocharged engines, where mechanical noise dominates, to assess the effect of turbocharger lag. It is suspected that this is an area where large differences may be found between nominally similar engines.

The type of fuel pump governor fitted to the engine can have a dramatic effect. Figure 12 illustrates the differences between an engine with a fixed timing pump and one with a speed advance unit. Although no direct correlation exists there is a strong relationship between the maximum rate of cylinder pressure rise and the combustion noise. It may appear to be an advantage to have fixed timing of fuel injection, but in general a better optimum timing plan can be achieved with a speed advance unit. This compensates for the fixed chemical reaction times having to take place in a shorter time interval as engine speed increases. This affects the delay period (between injection of fuel and the start of combustion); the longer the delay period the greater the amount of unburnt fuel to be ignited simultaneously so that the rate of pressure rise increases with increasing delay period. Under transient conditions the delay period can increase by up to 30% of the steady state value.

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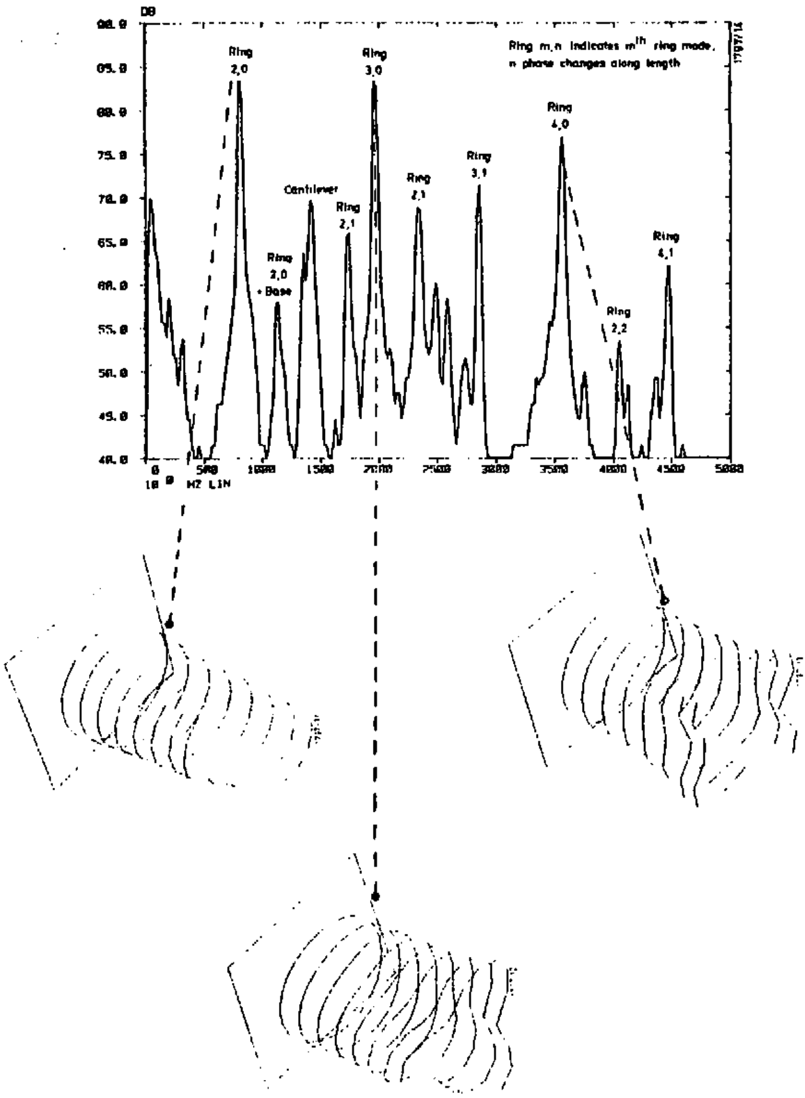

Fig. 13: Analysis of Clamped Cylinder Liner

5. COMPUTER ASSISTED ANALYSIS

5.1 Structural Vibrations

Many techniques have been used over the years to analyse and understand engine vibration behaviour. The complex structure of an engine block is a challenge to any approach. Current work at Shoreham uses a combined approach of finite element modelling and modal analysis, both of which techniques rely on the availability at reasonable cost of modern digital computers and developed methology. This approach is showing promise and a dual result is expected; the finite element modelling techniques in use are aimed at using a reasonable model of an engine structure which can be used to examine design alternatives ab initio; the modal analysis data for an existing structure, in addition to proving the finite element model, can provide the input for further synthesis work. Some examples of modal analysis applied to a cylinder liner, as a demonstration of the technique in engine-like structures, are given in figure 13. Ring modes are clearly identified together with higher orders where phase changes occur in ring modes down the length of the cylinder. The liner for this work was clamped to a stiff base plate to simulate the installation in an engine and the effect of this base on some modes was evident. The visualisation of these complex high frequency modes is a valuable facility.

5.2 Piston Slap

A mathematical model of piston motion (6,7,8) is useful when examining the optimum design for engine applications. The use of such models in noise and vibration reduction is fraught with difficulties since the minutiae of the piston motion can be extremely important in determining engine vibration. The determination of various parameters in the calculation, such as piston pin and skirt friction, is difficult and empirical values must be used. An example of the application of such a program is given in figure 14 for the effect of piston clearance. As can be seen a reasonable correlation is obtained in this case where changes in noise and vibration are compared with the predicted energy loss around the firing stroke TDC. This data was obtained from a lightly turbo-charged engine with low combustion noise levels. It should be stressed that this example covers a range of relatively small clearances whereas different results and better overall correlation may be found for larger clearances.

The selection of optimum pin position is less straightforward and phenomena which are outside the range of the mathematical model may be important. It has been shown that some reductions can be obtained where the piston transverses the bore clearance space with the skirt parallel to the cylinder wall (as shown by calculated impact times for upper and lower skirts being very close together), where oil film effects may dominate the running engine condition. This is another area where further work is needed to provide a more precise model for prediction.

5.3 Acoustic Intensity

The use of the cross spectrum technique for source identification on engines to determine major sources of acoustic emission holds great promise - an

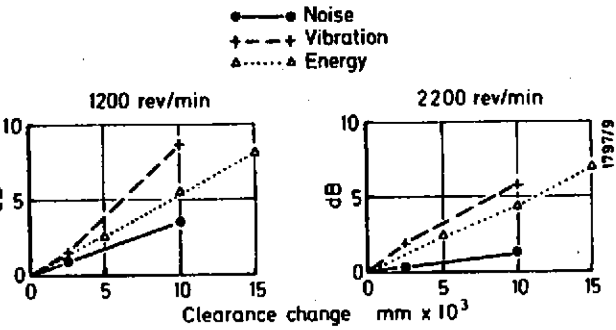

Fig. 14: Effects of Piston/Liner Clearance

alternative to the lead covering technique which retains sufficient accuracy would be welcomed by many. Initial work at Shoreham has confirmed the results of Chung et al (9) on simple sources and work is in hand on the acid test of a complex engine analysis. This technique is only possible due to the advent of high speed FFT processors.

5.4 Statistical Energy Analysis

Another technique relying heavily on FFT and digital processing is that

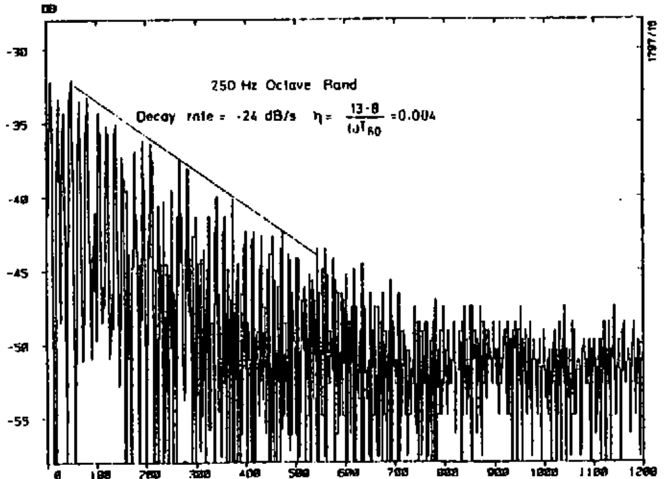

Fig. 15: Component Vibration Decay

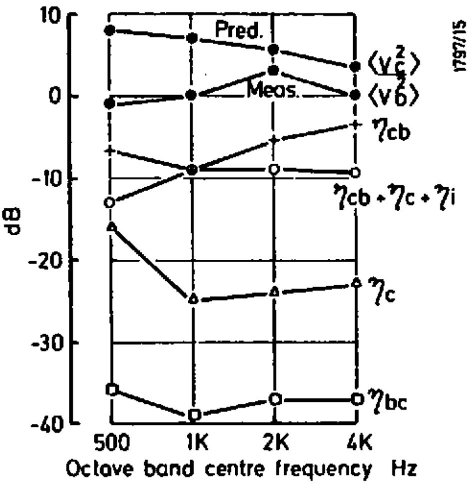

Fig. 16: Statistical Energy Analysis Example

pioneered by Lyon and DeJong (10). This technique is of most value when applied to the light weight covers of engines and to the fundamentals of vibratory energy power flow within engines. Work using this technique is now in progress.

The underlying assumption is that the energy flow within a system can be described by measurements of the acceleration levels and the component loss factors and coupling loss factors. To ensure reasonable statistical reliability a sufficiently large number of modes of vibration must be averaged. This implies the use of octave bands. Loss factors are evaluated through band filtered transient vibration decay measurements. Coupling loss factors can be calculated from point mobility measurements (see ref. 10).

An example of the application of this technique to a stiff engine cover is given in figure 16. These results are clearly approximate but indicate the general form of the technique. The high level of η_{cb} indicates that for this component the coupling between cover and block is the dominant term, any increase in η_c, the cover damping, would have to be very large to have any significant effect on the cover vibration. The levels also indicate that additional damping may be required if the cover were isolated. This additional damping may, in practice, be present in the isolation system.

6. CONCLUSIONS

6.1 Transient acceleration data can be analysed to provide information on the instantaneous combustion and mechanical noise elements. Both are evidently increased under transient conditions when compared with stabilised running. Methods of control must therefore act on both sources to obtain optimum results.

6.2 Computer assisted calculations such as finite element dynamic modelling and modal analysis can both be applied to engine vibration studies and work to provide such analysis at reasonable cost and within a sensible budget in order to refine structures for minimum noise and weight.

6.3 Mathematical modelling techniques for piston slap and energy flows in engine structures extend the understanding of the existing problems and provide direction indicators to improvements.

7. ACKNOWLEDGEMENTS

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APPENDIX I

Derivation of separation technique

Since the model of the radiated noise is the sum of the combustion noise and the mechanical noise:

$$SPL_A = CN_A \overset{\log}{+} MN_A \qquad (1)$$

$$SPL_R = CN_R \overset{\log}{+} MN_R \qquad (2)$$

where

SPL - sound pressure level

CN - combustion noise

MN - mechanical noise

(suffices A and R being at advanced and retarded conditions respectively)

It is assumed that the mechanical noise remains constant over the limited timing swing, so that:

$$MN_A = MN_R \qquad (3)$$

The structural attenuation of the engine to combustion forces is assumed to remain constant (linear system), so that:

$CPL - CN = R$ where CPL is the Cylinder Pressure Level, and so

$$CN_A - CN_R = CPL_A - CPL_R \qquad (4)$$

Rewriting equations (1) and (2) in terms of pressure gives:

$$P^2_{ref}10^{\frac{SPL_A}{10}} = P^2_{ref}10^{\frac{CN_A}{10}} + P^2_{ref}10^{\frac{MN_A}{10}} \qquad (5)$$

$$P^2_{ref}10^{\frac{SPL_R}{10}} = P^2_{ref}10^{\frac{CN_R}{10}} + P^2_{ref}10^{\frac{MN_R}{10}} \qquad (6)$$

(for clarity P_{ref} will be omitted from here onwards).

Subtracting (6) from (5) gives:

$$10^{\frac{SPL_A}{10}} - 10^{\frac{SPL_R}{10}} = 10^{\frac{CN_A}{10}} + 10^{\frac{MN_A}{10}} - 10^{\frac{CN_R}{10}} - 10^{\frac{MN_R}{10}} \qquad (7)$$

using equation (3), this reduces to:

$$10^{\frac{SPL_A}{10}} - 10^{\frac{SPL_R}{10}} = 10^{\frac{CN_A}{10}} - 10^{\frac{CN_R}{10}} \qquad (8)$$

now considering equation (4):

$$CN_R = CN_A - CPL_A + CPL_R \qquad (9)$$

or

$$10^{\frac{CN_R}{10}} = 10^{\frac{CN_A - CPL_A + CPL_R}{10}} \qquad (10)$$

or even

$$10^{\frac{CN_R}{10}} = 10^{\frac{CN_A}{10}} \times 10^{\frac{CPL_R}{10}} \Big/ 10^{\frac{CPL_A}{10}} \qquad (11)$$

Using equation (11) to substitute for CN_R in equation (8), this gives:-

$$10^{\frac{SPL_A}{10}} - 10^{\frac{SPL_R}{10}} = 10^{\frac{CN_A}{10}} - 10^{\frac{CN_A}{10}} \times 10^{\frac{CPL_R}{10}} \Big/ 10^{\frac{CPL_A}{10}} \qquad (12)$$

or

$$10^{\frac{SPL_A}{10}} - 10^{\frac{SPL_R}{10}} = 10^{\frac{CN_A}{10}} \left[1 - 10^{\frac{CPL_R}{10}} \Big/ 10^{\frac{CPL_A}{10}}\right] \qquad (13)$$

or

$$10^{\frac{SPL_A}{10}} - 10^{\frac{SPL_R}{10}} = \left[10^{\frac{CN_A}{10}} \Big/ 10^{\frac{CPL_A}{10}}\right]\left[10^{\frac{CPL_A}{10}} - 10^{\frac{CPL_R}{10}}\right] \qquad (14)$$

Taking logs and multiplying by 10 (to convert from pressures to pressure levels i.e. dB) gives:-

$$10 \log \left[10^{\frac{SPL_A}{10}} - 10^{\frac{SPL_R}{10}}\right] = 10 \log \left\{\left[10^{\frac{CN_A}{10}} \Big/ 10^{\frac{CPL_A}{10}}\right] \times \left[10^{\frac{CPL_A}{10}} - 10^{\frac{CPL_R}{10}}\right]\right\} \qquad (15)$$

or

$$10 \log \left[10^{\frac{SPL_A}{10}} - 10^{\frac{SPL_R}{10}}\right] = 10 \log 10^{\frac{CN_A}{10}} - 10 \log 10^{\frac{CPL_A}{10}} + 10 \log \left[10^{\frac{CPL_A}{10}} - 10^{\frac{CPL_R}{10}}\right] \qquad (16)$$

Rewriting equation (16) in the form of equations (1) to (4) gives:-

$$SPL_A \overset{\log}{-} SPL_R = CN_A - (CPL_R - (CPL_A \overset{\log}{-} CPL_R)) \qquad (17)$$

From equation (17) a value of CN_A can be obtained, as all other quantities in the expression are known measured values. Whence by back substitution in equation (4) a value of CN_R can be calculated. Substitution of CN_A and CN_R into equations (1) and (2) yields values of MN_A and MN_R; which will in fact be identical by definition.

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GEAR IMPACT NOISE IN DIESEL DRIVES

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Summary

The causes of loss of contact in gear drives and the effects on noise emission are discussed. Identification of loss of contact is not clearcut since conventional approaches are of little help. A possible measurement technique is described and some simple methods for attempting prediction of contact loss are given.

Introduction

Increasing emphasis on noise and the corresponding improvements in engine noise levels have focussed attention on the noise from the remainder of the drive and in particular gear noise and chain drive noise.

If the gear system remains linear i.e. the teeth do not lose contact, the system is relatively simple to analyse and the conventional approaches work well to locate troubles and hence to eliminate them. The excitation for the system is provided by the gear errors which remain constant in amplitude but whose frequency varies as speed varies while the gearcase, engine block and surrounding supports and structure behave as a complex but consistent system whose response can be described as a conventional frequency response or in terms of an impulse response. Provided that tests may be carried out over a range of speeds, preferably at constant torque, a 'waterfall' plot may be displayed showing the variation of noise power over the frequency range as speed varies (1). When sophisticated computer displays are not available the same essential information can be obtained by using a variable frequency filter set to a multiple of tooth frequency to determine system resonance and a fixed frequency filter to determine the frequency content of the forcing excitation from the gears.

Once the system response is known any troublesome resonances can be identified and eliminated and checks on single flank grating equipment (2) will show quickly whether there are gear errors which are unacceptably large. In cases where there are isolated errors such as single damaged teeth, time averaging techniques are better than the Fourier analysis approach and will locate the damage easily (3).

If loss of contact occurs between gears, analysis becomes much more complicated. Since the system is no longer linear it is not possible to regard it as having a simple input-output response and the techniques described above do not work correctly and in many cases give deceptive answers. For example gear errors at tooth frequency may give strong resonances at half tooth frequency or two or three times tooth frequency but little response at tooth frequency itself.

GEAR IMPACT NOISE IN DIESEL DRIVES

Causes

The simplest type of loss of contact occurs due to engine speed variation. Under idling conditions the energy required to accelerate and deccelerate the pistons and to provide compression in a 4-cylinder engine gives a speed variation of the order of 1.5% corresponding to a torsional vibration amplitude of the order of 0.5 at twice engine rotation frequency. Unless drag torques are exceptionally high the driven gears will not decelerate with the engine and loss of contact will occur. This type of rattle is relatively easily identified since load application stops it immediately.

A gear drive itself gives loss of contact when it provides a speed variation due to design or manufacturing errors greater than the variation which the system dynamics can accommodate; some very simple estimates can give an idea of the quantities involved. When tooth frequency is much greater than the main resonance frequencies of the system, the gears do not respond and so loss of contact occurs if the errors are greater than the elastic tooth deflections. A loading of 175 N/mm (1000 lbf/inch) facewidth gives an elastic deflection of 12.5 μm assuming the standard tooth contact elasticity of 1.4×10^{10} N/m/m, so that an error in excess of this value will give loss of contact.

Observed Effects

Loss of contact gives rise to very heavy impact loadings when the gear teeth meet again; these high loads may have effects on gear life but are usually troublesome due to noise radiation through the structure. It is however rather difficult to determine whether or not the gears are losing contact in an installation. The most obvious difference in the response of a non-linear system is shown in Fig. 1 where the characteristic 'jump' effect is shown. Some car gearboxes show this effect fairly clearly; the vibration level at once-per-tooth frequency drops very suddenly at a point in the range and there can easily be 10 dB differences between the power levels at that frequency dependant on whether speed was increased or decreased to the test point (4). If a 'jump' occurs it is a clear indication of non-linearity but the converse is not true.

In a gear mesh, in addition to regular errors at once-per-tooth frequency and harmonics there are lower frequency errors associated with pitch errors or synchromesh problems. These errors may be of sufficient magnitude to disturb the regular response from point A to point B in Fig. 1 or vice versa so that the response keeps changing between the two states. This is sometimes observed as a "warbling" sound if the changes are infrequent. An additional effect of a non-linearity as indicated in Fig. 1 is that there is no longer a clear resonant frequency but increased amplitudes occur over a wide range of speed, sometimes down to half the 'natural' frequency of the system.

A regular once-per-tooth error which gives rise to loss of contact may give a response shown diagrammatically in Fig. 2 where the response is also periodic at once-per-tooth frequency and the gears bounce elastically on the rising portion of each sinusoid. In this figure the full line is the disturbance in the system due to errors; this is normally called the 'static' transmission error. Transmission error for a gear drive is defined as the difference between the actual position of the output and the position the output should be if the drive were perfectly smooth with no errors or deflections. The dotted line represents the actual position of the output relative to its correct position; at impact the lines cross due to elastic deflection of the teeth. The most obvious effect of loss of contact in this case is the increase in the levels of second and third harmonic vibration from the impact. It is thus reasonable to assume that if harmonic levels increase as load is reduced that loss of contact has occurred. Unfortunately this is not always true as is shown diagrammatically in Fig. 3; in this case under high loads high harmonic levels are associated with following the transmission error while under low loads the high frequency components of error are avoided.

Frequency analysis of the noise or vibration is of very little direct use in diagnosis since it cannot show any essential difference between contact and non-contact conditions, particularly when there are additional random vibrations present from other sources. The time-averaging approach is more powerful and may typically give traces as shown in Fig. 4; again there is no fundamental difference between contact and non-contact conditions but if variation of load affects part of the trace only there is a strong possibility of non-linearity. It should be noted if frequency analysis methods are used with long averaging times that only multiples of once per revolution can exist in theory though in practice there may be clear vibrations at frequencies which are not exact multiples of once per revolution; the difference is one of definition only.

Measurement

Noise and vibration measurement will not directly determine whether contact has been lost but transmission error measurement at speed can show what is happening. Small radial gratings may be attached to the ends of shafts in Fig. 5; accuracy is relatively low, about 5 seconds of arc but is more than sufficient for automotive purposes and the grating can run up to 6,000 rpm (5). The principal limitation on performance is that the grating vibrates torsionally on the flexible connection to the shaft end and this imposes an upper limit of about 600 Hz for reliable information. At low speed the gratings measure the 'static' transmission error in the system, giving a good check on assembly and manufacturing errors, then as speed rises, the observed transmission error is distorted due to resonant effects and due to loss of contact. Since the transmission error measures the movement apart of the gear teeth it is possible to see how far the gear teeth separate and whether or not they travel fully across the backlash zone. The limitation to this measurement is due to the fact that the gratings only measure torsional movement but do not respond to lateral movements so that where high lateral movements are possible, as in a machine tool gearbox with long shafts, separation may occur without affecting observed transmission error.

Predictions

Since detection of separation is very difficult, if not impossible, it is usually necessary to attempt to predict whether or not it is occuring. This diagnosis is important because the effects of changing stiffness, masses, damping or accuracy will depend greatly on whether or not separation occurs; in particular decrease of stiffness will often greatly improve a linear system but will correspondingly make a non-linear system much worse. The information necessary for prediction is

(i) The 'static' or manufacturing transmission error. Ideally this should be the error under load but for most drives, particularly lightly loaded ones, the zero load error is sufficient. It is possible to deduce the loaded error from the no load error provided the gears are made accurately and consistently. If engine torsionals are involved these are usually well known.

(ii) The dynamic response of the gears and their supports to excitation at the mesh point. The response is usually reasonably linear but is complex since even a single pair of gears will normally have six degrees of freedom. A full analysis of the vibration response of a multi-degree of freedom will involve a laborious matrix approach to determine system response.

Fortunately the dynamic response can usually be predicted with sufficient accuracy by relatively simple methods at particular frequencies. Fig. 6 shows an idealised system for a gear pair and the response of the system to forcing is easily deduced.

It is the relative displacement between the teeth that is imposed by the transmission error but it is simplest to start by considering a force P between the teeth. The total deflection at the tooth is then the algebraic sum of the elastic tooth deflection, the response of the gear as an inertia on a torsional spring and the response of the gear as a mass mounted on a linear spring (6). The dynamic response of system X is then

$$x_p = \frac{P}{S_t} + \frac{P\, r^2_x}{K_x - J_x\omega^2} + \frac{P}{S_x - M_x\omega^2}$$

and correspondingly, with no tooth deflection term

$$y_p = \frac{P\, r^2_y}{K_y - J_y\omega^2} + \frac{P}{S_y - M_y\omega^2}.$$

The transmission error $\epsilon = x_p + y_p$, taking note of signs so that from a given ϵ we can deduce P and also deduce the forces which act upon the bearing housings; these are given by

$$F_x = \frac{S_x}{S_x - M_x\omega^2} \quad \text{and} \quad F_y = P \,.\, \frac{S_y}{S_y - M_y\omega^2}$$

For separation it is only necessary for the dynamic component of P to be greater than the steady imposed load. Typical figures for a medium size gear might be r = .075 m, J = .01 kgm^2, $S_t = 7 \times 10^8$ N/m, $S_x = 10^8$ N/m, $M = 10^5$ kg, $K_x = 5 \times 10^5$ Nm/radian. The corresponding deflections at a tooth mesh frequency of 800 Hz are a deflection per unit force of 1.4×10^{-9} due to the teeth bending, $- 2.2 \times 10^{-8}$ due to rotation effects and $- 6.5 \times 10^{-9}$ due to gear movement giving a total of $- 2.7 \times 10^{-8}$ m/N. The negative sign shows that the system is above a resonance i.e. is deflecting in the opposite direction to the instantaneous force. If this gear were mating with a similar gear whose response (without tooth deflection) was $- 2.85 \times 10^{-8}$ the combined response would be 5.56×10^{-8} N/m and so an error of 10 micron would generate 180 N force; this compares with the 7000 N force that would be generated if the gears could not move. With this error the forces at the bearings would be 118 N.

There are particular cases where a simpler approach is informative. The condition of tooth frequency much higher than dynamic response frequencies has already been mentioned and leads to a simple criterion that the elastic tooth deflection should be greater than the amplitude of transmission error to prevent separation. At frequencies much lower than resonance with low tooth deflections a criterion is that the nominal load on the system should be sufficient to accelerate gear inertias fast enough to maintain contact; this gives the criterion that $P(\text{nominal}) > Me(2\pi f)^2$ where f is tooth meshing frequency and M is the effective mass of the system; this assumes that the error is predominantly at tooth frequency.

Once separation has occured it is desirable to know whether the gears will cross the backlash zone completely. If this occurs the resulting bounce from the backs of the teeth can give very powerful bursts of vibration hammering across the backlash. It is not easy to predict hammer without a full non-linear model; even then results are not reliable since virtually nothing is known about loss of energy at tooth impacts. However an order of size estimate can be obtained by assuming gear contact for part of the mesh and hence deducing the relative velocity between the gears when contact is lost. This gives the time of loss of contact and the maximum separation.

As a simple example consider a pinion of inertia 10^{-2} kgm^2, pitch circle diameter 100 mm, torque 50 Nm, 2400 r.p.m., errors of 10 μm at 25 times per revolution and 80 μm at once per revolution and meshing with a large massive wheel; assume that the wheel does not deflect, that the pinion drive shaft is sufficiently flexible that the torque remains constant and that only torsional motion is involved. The figures are typical of a medium accuracy reduction gear suitable for general use for low speed machinery.

A quick check shows that the torsional acceleration due to the once-per-tooth component is $10^{-5}\,(2000\pi)^2/.05$ which involve a torque of 10^{-2}. $10^{-5}\,(2000\pi)^2$ / .05 i.e. 79 Nm which is sufficient to give loss of contact. The maximum increase in speed above average for the pinion is $10^{-5} \times (2000\pi)/.05 + 8 \times 10^{-5} \times (80\pi)/.05$ which is 1.66 radians/sec. Whilst out of contact the deceleration involved is $50/10^{-2}$ i.e. 5000 rad/s^2. This 'take-off' velocity and deceleration combine to give an out of contact time of 0.66 ms and a maximum 'height' of 13.7 μm; this time estimate assumes that contact occurs at the same 'height' as loss of contact. When the time of lost contact corresponds to tooth frequency there is a strong possibility of the type of response indicated in Fig. 2. In this case the velocity would have to be about 50% greater for bouncing to occur powerfully and the bounce height would then be about 30 μm.

Noise Reduction

When the mechanism of noise generation has been identified it is possible to start assessing likely effects of possible modifications. The most fundamental choice is whether to alter design or attempt to improve manufacturing accuracy; inevitably economics must influence such choices very strongly. Within the design area two approaches may be used since it is possible either to minimise the likelihood of impacts occurring in the first place or to allow the impacts to occur but to isolate their effects from the remainder of the system. In the very few cases where gear damage is occurring the latter approach may, of course, not be used. There are unfortunately no general rules for deciding whether stiffness and masses should be increased or decreased to improve matters. It is however rare for additional damping to give higher noise levels.

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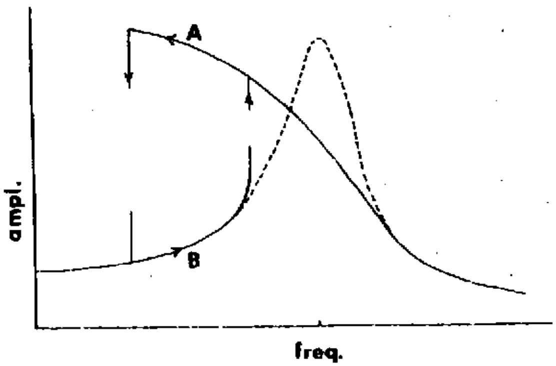

Fig. 1 Response curves for simple linear (dashed line)+ non-linear (full line) systems.

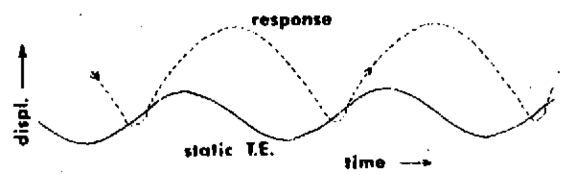

Fig. 2 Sketch of response of non-linear system to sinusoidal excitation. The full line indicates the static transmission error and the dotted line shows the response.

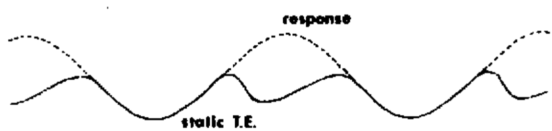

Fig. 3 Effect of load on system response with some harmonic errors. The full line indicates the static transmission error and dashed line is the response at low applied loads.

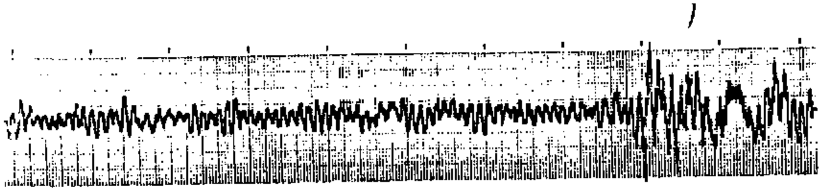

Fig. 4 Vibration trace with contact loss.

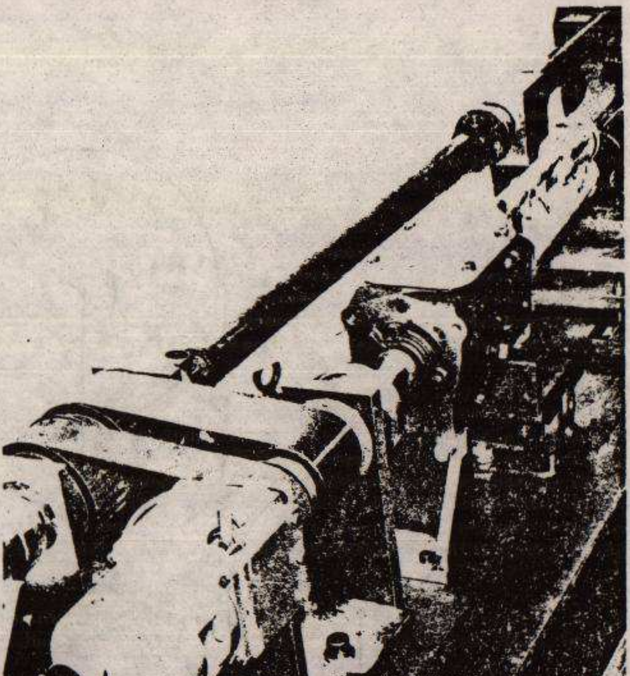

Fig. 5 Grating head in position on a test rig.

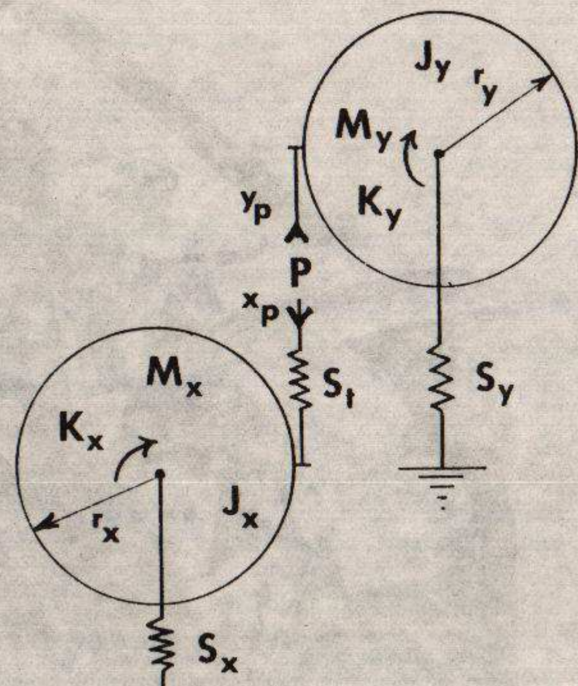

Fig. 6 Idealised gear model.