DIESEL ENGINE TUNING

A NEW APPROACH

BY

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This article first appeared in the January/February, 1964, issue of The English Electric Journal *and is reproduced by permission of the Editor.*

From the earliest days of the oil engine, the majority of Diesel engineers have considered it axiomatic that the best conditions for their engines were secured by meticulous adjustments relating to individual cylinders so as to balance firing pressures and exhaust temperatures.

This emphasis on individual adjustment has survived in medium-speed and slow-speed engines throughout more than half a century of progress in power, efficiency and production techniques. On the other hand, the practice has been abandoned in many high-speed engines, which employ a bench-calibrated block pump and are statically timed. This is generally looked upon as a concession from desired higher standards, yet it is seldom that such an engine suffers thereby, despite the achievement today of very high ratings.

This article reviews what tuning has to deal with, how far it succeeds, and whether its aim can be better achieved in other ways. Such a review has general application, but the examples cited relate to a medium-speed range of four-stroke engines.

The Problem

In the initial phase of designing a Diesel engine it will be so proportioned that it can fulfil the rating required at minimum cost of manufacture or upkeep. The timing and air supply giving the best compromise between economy and stress will be chosen.

A certain peak cylinder pressure and a certain exhaust temperature will be arrived at, whereas the engine will produce a scatter of values in each case. This scatter has several causes and in determining what is acceptable the following questions arise:

(1) How much can scatter be accounted for by manufacturing tolerances?
(2) How much is due to dynamic behaviour?
(3) Is tuning a valid approach to the reduction of scatter?
(4) What is the best method of dealing with scatter?

Manufacturing Tolerances

The four factors on which manufacturing tolerances have a bearing are compression ratio; trapped air charge; static timing; and equality of fuel injection.

Compression ratio

Consider as an example an engine having a stroke of 12 in. and a compression ratio of 11·7 : 1. The exact value of the compression ratio is dependent upon over a dozen separate dimensions, ranging from a ¼ in. to 2 ft 6 in. or so, whose tolerances, without individually exceeding 0·010 in., may total as much as 0·060 in. The resultant variation in compression pressure would be 40 lb per sq in., and in compression temperature 30 degrees F., starting with equal masses of air at equal manifold temperatures. For exactly equal injection quantities, equally timed and identically burnt, the variations in firing pressures and exhaust temperatures would be very nearly the same.

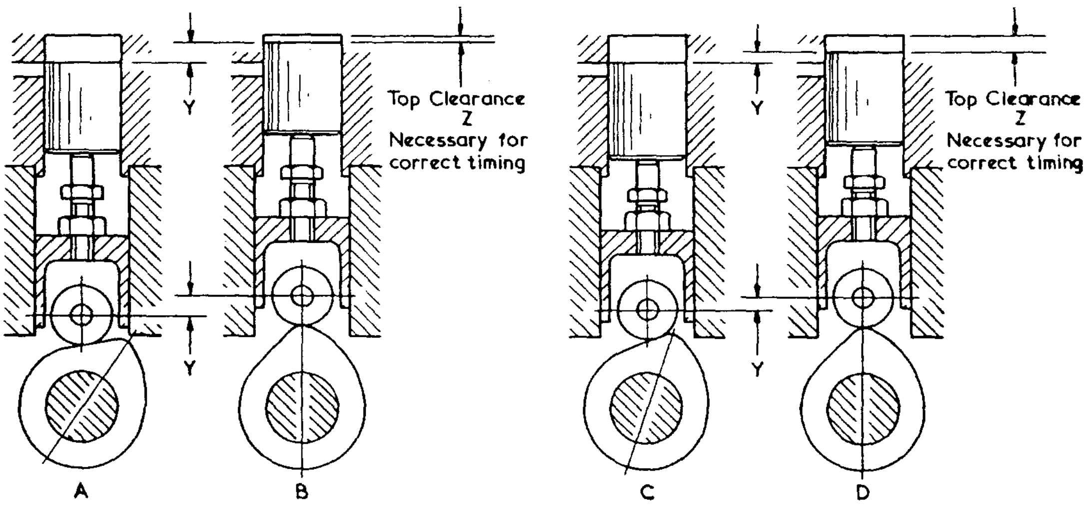

FIG. 1—CONFIGURATION OF CAM, TAPPET AND PUMP AT (A AND C) COMMENCEMENT OF INJECTION; AND (B AND D) MAXIMUM CAM FOLLOWER LIFT. IN EACH CASE THE PUMP IS SET CORRECTLY FOR THE REQUIRED INJECTION POINT.

A shows the case for an over-retarded cam and C that for an over-advanced cam within permitted limits of top clearances. In either case Y (the residual lift) + Z = the timing dimension of the pump.

The Trapped Air Charge

Apart from dynamic effects (which are considered later), manifold design, equality of air density throughout the manifold, and valve timing—particularly at inlet valve closure—all affect the trapped charge.

With normal manufacturing quality, the 'as made' variations between manifold sets will not make a significant difference, nor will normal valve timing accuracy up to even $\pm$ 2 degrees from nominal. With a multi-blower layout, however, a boost difference may arise between manifolds feeding different groups of cylinders which may reach an extreme of 2 in. Hg, at which cylinder filling may vary by 4 per cent and peak pressures by 25 lb per sq in. This effect too can be easily eliminated by coupling the manifolds to permit equalization of boost density.

Static Timing

Assuming that timing figures take into account the dynamic behaviour, how closely can the engine be timed?

The answer has been much obscured by the favour accorded in the past to window timing. This was evolved to give the tuning advocates a datum, from which to start tuning, which was less tedious than spill timing. The best standard of accuracy of the window method is from $\pm 1\frac{1}{2}$ degrees to $\pm$ 2 degrees of crank angle, and the spill method, if applied by a skilled and conscientious operator, can obtain a reliable $\pm \frac{1}{2}$ degree. However, both of these methods are rendered obsolete by the simple and foolproof practice of timing by plunger fall. Referring to FIG. 1, the timing dimension of the pump (spill port cut-off to top of barrel) is controlled to within $\pm$ 0·001 in. ($\pm$ 0·002 in. in some designs). The flywheel is set to the injection point required and the cam follower lift from that setting to its maximum is measured with an accuracy of 0·001 in. The difference is the required top clearance, which can also be set to $\pm$ 0·001 in., giving a total timing accuracy of 0·0035 in., equivalent to $\pm \frac{1}{4}$ degree to $\pm \frac{1}{2}$ degree. If there is any risk of a camshaft standing high in any bearing clearance it should be nipped otherwise a retardation of about one degree may be introduced.

The effect of injection timing on exhaust temperature is, within the usual working range, only of the order of 5-10 degrees F. per degree crank, whereas its effect on firing pressure is approximately 20 lb per sq in. per degree crank—more if the timing is badly retarded, and less it if is advanced beyond optimum.

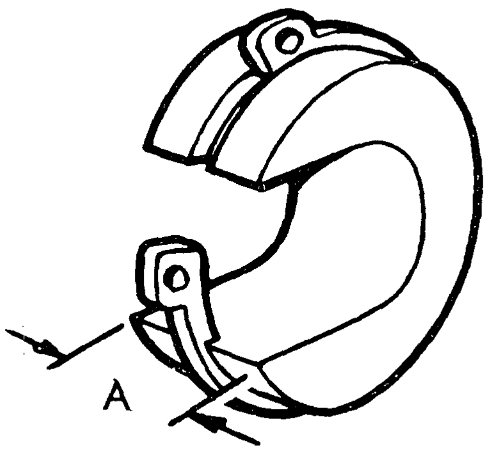

FIG. 2—A STANDARD RACK SETTING GAUGE

Equality of Fuel Injection

Equality of fuel injection depends on the pump and injector, and given the usual care in manufacture of the high-pressure fuel system, and the setting of the injector, it depends on the pump alone. It is assumed that aeration in the fuel supply would be eradicated as soon as it arose, since it reveals itself quite distinctly either in persistent unbalance of end cylinders, or as spasmodic flickering of the governor linkage or of the wattmeter.

Now whereas small differences may exist between the characteristics of nominally identical pumps, it is always possible to adjust the calibration of the control rod to the element so that, at a nominated balance point, at or near full load, the outputs of all pumps of a given pattern give equal deliveries within a tolerance of $\pm$ 1 per cent or better. It is not difficult to maintain this accuracy in servicing given the necessary calibration rig and the observance of a simple routine. Block pumps, of course, are phased and balanced together at the time of manufacture or of servicing, but individual pumps have to be calibrated to some external feature, such as a graduated scale or a shimmed collar. In turn, when the pump is fitted to an engine, the correct setting has to be retrieved from the scale or the collar. The reading accuracy obtained with rack and pointer, $\pm \frac{1}{4}$ mm, prevents any reduction of overall variation in output between cylinders below $\pm 4\frac{1}{2}$ per cent, whereas the shimmed collar, used with standard slip gauges, with a setting accuracy of $\pm$ 0·001 in. can reduce this to $\pm 1\frac{1}{3}$ per cent. The effect in these two cases on exhaust temperature at full load would be 30-40 degrees F. or 10-15 degrees F. respectively. The effect on peak pressures would be about 5 lb per sq in. in the first case, and negligible in the second.

FIG. 2 illustrates a standard slip gauge.

Dynamic Behaviour

The three factors which require consideration here are: trapped air charge, scavenge air, and dynamic timing.

Trapped Air Charge

Trapped air charge variations may arise repeatedly along a given manifold design at a given engine rating, due to patterned differences in the behaviour at each cylinder. For instance, the cylinders near the turbo-blower end of a uniform manifold have to deflect air from a faster flowing stream than those remote from the blower, and also the mean density during a cylinder's admission period varies according to the location of the cylinder preceding it in firing order.

It can be calculated that a 5 per cent variation in the trapped charge in two otherwise identical cylinders will affect the compression—and hence the firing—pressure by about 30 lb per sq in., and that this will in turn affect the exhaust temperature by about 30 degrees F.

Analysis and correlation of patterns of exhaust temperature and peak pressure variations for a range of engines reveals that, at ratings demanding about 1·5 boost pressure ratio or more, $\pm$ 2 per cent variation may be experienced, but that with lower pressure ratios, particularly at the highest speeds, the variation may be double this.

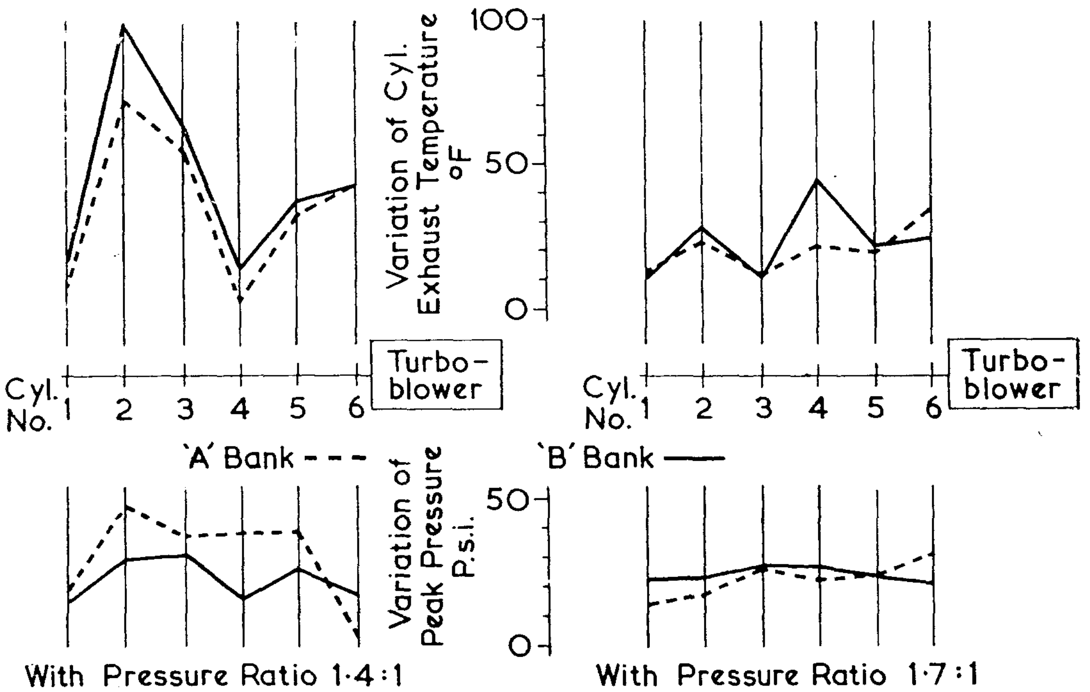

FIG. 3—AVERAGE CYLINDER READINGS OF GROUPS OF SIMILAR TWELVE-CYLINDER ENGINES WITH DIFFERENT BOOST PRESSURE RATIOS

FIG. 3 depicts exhaust temperatures and peak pressures for engines of similar build and different boost pressure ratios and uniform timing. Naturally aspirated engines would manifest these variations to a lesser degree.

Scavenge Air

The effect of the scavenge variation between cylinders of a supercharged engine may again be assessed by consideration of engine behaviour patterns. It is not hard to see that there can be no effect on peak pressure due to this factor alone, but it can produce differences of exhaust temperature of 80 degrees F., particularly if the boost pressure or manifold volume is not great enough to prevent variation in air manifold density, or if pulse reflections in the exhaust pipe reduce the net instantaneous pressure drop across the cylinder during the scavenge period.

FIG. 3 illustrates this point. No. 2 cylinder on each bank is displaying appreciably higher exhaust temperatures due to scavenge air deficiency, yet its peak pressure indicates the trapped charge to be as good as its neighbours.

Dynamic Timing

Dynamic timing, particularly in an engine having individual pumps, may be affected by camshaft twist under load, and the behaviour of camshaft drive chains (if used). It has been found that a long camshaft may, at its extreme end, oscillate by up to $\pm$ 2 degrees under the influence of the cyclic fuel pump driving torque. In practice, however, this fluctuation is at about its zero value at the moment when injection commences, and it is therefore not so much the timing as the rate of injection which is affected. The effect of this on performance is not enough to be discernible.

The correction of the effects of camshaft chain drives on timing does call for a little ingenuity, particularly on some vee engines. With in-line engines, its only outward effect is the gradual retardation tendency with age which can easily be corrected by the methods of timing now available. In vee engines having one chain, with the jockey sprocket between the camshafts, the reverse torque from the trailing camshaft after one of its cams has passed its peak tends to advance

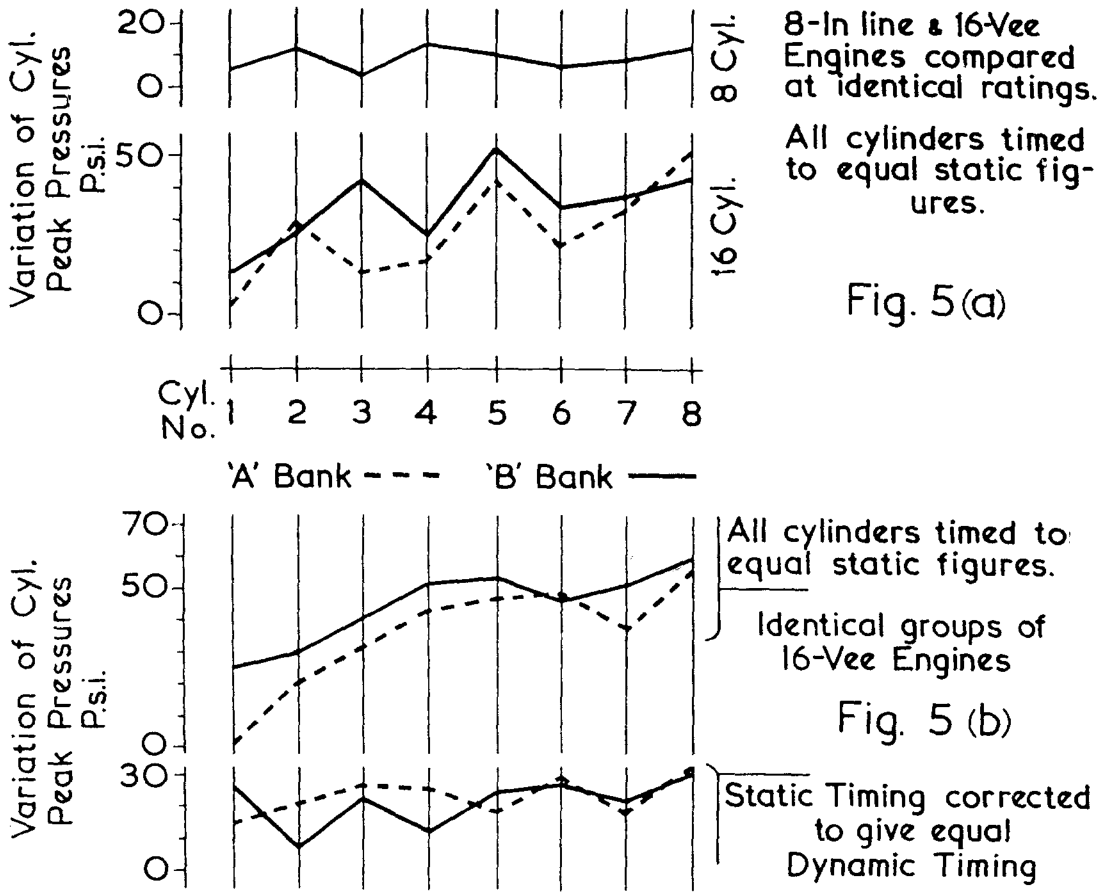

FIG. 5—EFFECT OF (*a*) CAMSHAFT CHAIN DRIVE IN VEE ENGINES, AND (*b*) CORRECTION OF DYNAMIC TIMING

the following cam on the leading shaft, whereas the leading shaft is unable to help the trailing one, since the slackening of the chain over the jockey sprocket at this stage merely allows the centrifugal force to gather the slack and actually tends further to retard the trailing camshaft. The degree of variation depends on which pair of cylinders is involved, and is most felt at the remote end of each camshaft. On the medium-speed engines considered here, and with a correctly tensioned chain, it usually amounts to a degree difference between the banks with up to an extra $1\frac{1}{2}$ degrees retardation for the end two-three cylinders of the 12 vee and 16 vee engines. FIG. 5 illustrates these points.

Chain stretch or bedding requires adjustment of chain tension. An increase in side-play of one inch at the middle of a two-foot run will affect all timings by about $\frac{1}{2}$ degree (in opposite senses on the two banks in vee engines). Although this affects the relationship of cams to flywheel, this is of no importance if the timing method described above (i.e. by plunger fall) is followed.

Taking into account all these points and with reference to average actual performance, it should be possible to set the static timing on any engine type so that the dynamic timing will not vary among cylinders by more than $\pm\frac{1}{2}$ degree crank, giving a total effect on peak pressure of 15-20 lb per sq in.

How Valid is Tuning?

Before answering this question it is necessary to consider where the engine is vulnerable. In the design stage, the engine is proportioned to withstand pressures well in excess of the normal anticipated maximum—certainly beyond the range of variation mentioned below.

TABLE I—*The influence of Exhaust Temperature, Peak Pressure, and H.L.F. of normal range of variation of factors affecting these quantities.*

	*Scavenge deficiency**	*Cylinder charge* ($\pm$ 3%)	*Compression ratio* ($\pm$ 3)	*Pyrometer errors* ($\pm$ 20 *deg. F*)	*Errors in fuel pump setting* ($\pm$ $1\frac{1}{3}$%)	*Pressure reading errors* ($\pm$20 *lb/sq in.*)	*Errors in timing* ($\pm$ $\frac{1}{2}$ *deg. F*)
Exhaust temp. $\pm$ (deg. F.)	40	18	15	20	5	0	3
Peak Pressure $\pm$ (lb/sq in)	0	20	20	0	1	20	10
Heat Loading Factor $\pm$	0	0·8	0	0	0·4	0	0·25

TABLE II—*Effect on H.L.F. of levelling Exhaust Temperature, Peak Pressure, or both, to eliminate the variations indicated in* TABLE I (*Effects are evaluated for each column separately*)

	*Scavenge deficiency**	*Cylinder charge* ($\pm$ 3%)	*Compres-ratio* ($\pm$ 3)	*Pyrometer errors* ($\pm$ 20 *deg. F*)	*Errors in fuel pump setting* ($\pm$ $1\frac{1}{3}$%)	*Pressure reading errors* ($\pm$ 20 *lb/sq in.*)	*Errors in timing* ($\pm$ $\frac{1}{2}$ *deg. F*)
Exhaust Temp. only	2·2	0·05	0·6	1·1	0	0	0
Peak Pressure only	0	1·2	0·5	0	0	0·5	0
Both	2·4	0·6	1·0	1·2	0	0·5	0

*To give the effect of $\pm$ 40 deg. F. on exhaust temperatures.

The branch outlet pyrometer measures a mean temperature of 600-900 degrees F., but the actual temperature may vary from the order of 1500 degrees F. at exhaust valve opening to 100 degrees F. during the scavenge period, so that mean exhaust temperature indications are a poor guide to maximum exhaust temperature. Moreover, the exhaust valve is invariably of at least an austenitic steel, and is quite capable of dealing with the maximum temperature in a Diesel engine operating on a Class A or B fuel, even with an otherwise unacceptable deterioration in combustion, provided that the seat is in reasonable condition. Much the same applies to the supercharger turbine (where applicable) where the temperature variation in the gas is less extreme.

In the great majority of engines, it is in fact the piston which will succumb first to deterioration in designed thermal conditions. Naturally some of the factors which add to the piston thermal loading will also reflect in the mean exhaust gas temperature, but several others affect the piston while having no direct influence on any other component or variable. Oil cooling of the piston will reduce the gap between the piston limit and the valve limit, but only very rarely will the valve become the limiting factor. Even then improved valve material will usually raise the operating limits again.

In considering the piston, it is necessary to introduce a relatively new concept, that of a heat loading factor or h.l.f. The h.l.f. is derived mathematically from consideration of five different parameters. It has been found to vary widely between engines having very different ratings but almost identical exhaust temperatures, and it correlates well with experimental piston temperature measurements. It is beyond the scope of this article to explain in detail, but the use of heat loading factors will assist the interpretation of the behaviour discussed here.

The following relates particularly to an engine having an uncooled aluminium alloy piston.

The usual heat loading factor for a charge cooled engine of medium speed would be 20 to 30, on an arbitrary scale where 15 represents the usual safety limit. Therefore a worsening of the heat loading factor by two numbers on this scale represents an encroachment of 15-40 per cent of the safety margin.

TABLE I shows the effect on peak pressure, exhaust temperature and h.l.f. of average incidence of a number of factors. The extremes of scatter seldom act together, the normal total scatter with average production engines being ± 15 to ± 40 lb per sq in. and ± 20 to ± 50 or ± 70 degrees F. Naturally aspirated engines exhibit lower ranges of scatter since some variables do not apply.

Now the conventional procedure when confronted with such inequality of readings would be to advance or retard ± 2 degrees and to alter the fuel pump rack settings by ± 1½-2 mm, i.e. by ± 12-15 per cent of the full load quantity of fuel. TABLE II shows the effect on the h.l.f. of such adjustments.

Referring to the tables, it is seen that in the case of the timing alteration, factors have been taken into account which together could have been responsible for 25 per cent only of the discrepancy, and these have been altered to offset factors which have no effect on timing. Moreover, it may well be unwise to advance the timing to compensate for a low compression pressure since if the engine has to run through a variable speed change with fixed timing, the bearing pressures may be left damagingly high at the low speed.

Much of the same applies to the alteration of fuel pump settings. Here the variables considered could only have provided 8-10 per cent of the discrepancy, and the harm which could result is greater. For instance, the cylinder which now has 12-15 per cent extra fuel is in fact carrying a substantial permanent overload without the benefit of the necessary extra air. Its thermal loading will be significantly worse and its combustion will be prejudiced. On the other hand, the cylinder whose fuel input has been cut will be unable to support combustion when running at no load, particularly on a constant-speed engine where the no-load injection rack opening is often only 1½-2 mm above the 'dead rack' position. Any cylinders which do not fail to fire at idling will be substantially cooler, so that the quantities of oil passing the piston rings will be two or three times greater than at full load. With no gas pressure to oppose it and with no firing heat to burn it, much of this oil will escape into the exhaust manifold. When the engine next reaches half load the heat of the exhaust gas will char this oil and burn it out, thus producing objectionable smoke.

The quantity of oil which might be thrown into the exhaust system in this way from a 2000 h.p. engine running at 750 rev/min on no load if the cylinders have a rack scatter of 3-4 mm might be only two pints per hour, but two pints lying in the exhaust system when load was re-imposed could make thick yellow smoke which would not clear for 5 to 15 minutes, depending on the length of the exhaust system.

With balanced pumps, maximum idling scatter is equivalent to only ± ¼mm of rack scatter, and is insufficient to prevent a cylinder from firing.

What is the Correct Treatment?

Obviously there will always be some differences between cylinders due to manufacturing tolerances, and even quite expensive attention to the true causes would not eliminate them, so let us consider the implications of leaving them alone.

The compression ratio effect is not likely to provide the last straw either for the bearing and crankcase structure or for the exhaust system. Its effect on fuel economy will be all but negligible, and it does not harm the piston.

The charge air variations also will not affect economy to any significant extent and will not over-stress the engine or over-heat the exhaust. The piston does suffer slightly, but not as greatly as it would if timing settings had been adjusted to equalize peak pressures. Actually, in this case, if fuel pump output alone was adjusted to equalize the exhaust temperatures, the piston conditions would be restored to equality. The difficulty is that an operator would not be able to pick out this solitary justifiable cause for adjusting output from among half a dozen possible explanations for the same symptoms. The greater risk of oil throwing on low load would still exist.

The scavenge variation is in fact completely harmless. Scavenge air in the Büchi system is provided for the turbocharger's benefit primarily, to a small extent for the valves and piston, and in general to flush out the cylinder. The variations being discussed here, 80 degrees F. range of exhaust temperature, will not have a greater effect on the piston than $\pm$ 5 degrees F. Moreover, scavenge deficiency seldom if ever leads to a deficiency in cylinder filling and is not, given even ordinary care in design and development, sufficient to prevent a thorough expulsion of residual gases from the cylinder.

The timing errors themselves produce less than the effect of the 'tuning margin' usually allowed when tuning on conventional lines is carried out, while the pump output variation is even closer than would have been required. The greater the care taken in making these settings the more will this be so. The optimum may not give identical peak pressures on each type of engine in a range at equal rating and speed, due to differences in such factors as cylinder filling, but it remains illogical to depart from the chosen optimum on this account by the necessary 2-3 degrees.

A point which should be borne in mind is that, whereas timing and pump output can be set by accurate static measurements, the measurement of peak pressures and exhaust temperatures relies heavily on such factors as the accuracy and consistency of the instrumentation and associated indicator passages. Obviously mal-functioning of these items can lead to quite heavy spurious adjustments if too great a reliance is placed upon them.

It is also clear that no alterations should be made which simply reflect the shop ambient conditions at the time of test. It is clearly ridiculous that two otherwise identical engines, possibly operating together under the same conditions, should have variations in their timing up to $\pm$ 2 degrees solely because one was built in summer and the other in winter.

Finally, a regime of static settings without further adjustment has the advantage of highlighting a cylinder that is misbehaving, whereas with tuning by individual pump adjustment readings were levelled whatever the cause of their variance and the bad was hidden along with the distortion of the good.

Conclusion

Conventional methods of tuning Diesel engines are not only unreliable, but sometimes dangerously misleading. The only two points which should be considered are (1) exactly equal injection quantities on all cylinders and (2) injection into every cylinder at exactly equal dynamic timing. Both of these requirements can be achieved by static settings on medium-speed and slow-speed engines, no less than in the high-speed designs, where this 'concession' was first introduced.

It has in fact been established that, far from being a concession, and given normal commercial accuracy of manufacture, static settings are the most scientific, logical and accurate way of setting the running characteristics of a Diesel engine.

Acknowledgement

Acknowledgement is gratefully made to English Electric Diesel Engines Limited for facilities for gathering the material for this article, and for permission to publish it, and also to C. A. V. Limited for their co-operation.

STEAM VESSELS

The following is an extract from the Introduction to The Steam Boat Companion; and Stranger's Guide to the Western Islands and Highlands of Scotland, *published by James Lumsden and Son in 1820.*

To many travellers, who resort to the facility of conveyance which steam vessels afford, it may not appear uninteresting, nor out of place, here to state a few particulars relative to the invention of those useful vehicles, which are known only to a few scientific men: So far back as the year 1733, a publication was circulated, setting forth the advantages that would result from the use of vessels to be impelled by wheels, or paddles, and describing the mechanism by which they were to be put in motion; a copy of which is preserved in the Advocates' Library at Edinburgh. The proposal seems to have been considered so chimerical, that no notice was taken of it; and the idea of such a power, for purposes of navigation, lay dormant for near half a century, till the late patriotic and enterprising Mr. Millar of Dalswinton, in Dumfries-shire, without any knowledge of the work alluded to, but with that genius and perseverance which marked his character, appears to have formed more correct notions of the practicability of propelling vessels by such means, and he set his mind to the construction of them upon scientific principles. Accordingly, in 1776, he completed a small boat, with a steam engine, which was set in motion on the lake of Dalswinton, in the presence of some ingenious persons of his acquaintance.

Ten years thereafter, in 1786, a large vessel was built under his direction at Grangemouth, the engine and machinery of which were made by Mr. Symington, civil engineer; and several experiments were made on the Forth and Clyde Canal, in presence of Sir Thomas Dundas of Carse, and several other gentlemen connected in the management of the canal, which gave entire satisfaction; but from the agitation of water which the action of the wheels produced, and the consequent risk of injuring the banks, this vessel was not considered as eligible for canal navigation. The late ingenious Earl of Stanhope afterwards made similar experiments in England, with the same satisfactory results.

Of these discoveries, we have seen an engraved plate, published in 1766, done from drawings by Mr. Naysmith, which delineates paddles of the very same construction, and applied on the same principle as those now in use.

From this, it appears, that the merit is not due to Mr. Fulton of America, who was on a visit to Scotland, and had, previously to his leaving this country, seen Mr. Millar's invention. But the greatest praise is due to Mr. Henry Bell, of Glasgow; who, with much labour and expense, invented several material improvements in the machinery, which now render steam vessels so complete, and of such general benefit to the public.

In 1800, Mr. Bell produced a large model of a steam vessel, 27 feet long, which he presented to the late Lord Melville, then at the head of the Board of Admiralty; but being overlooked by his Lordship, and his colleagues, was eagerly adopted by the American Government, and speedily put into successful practice upon the great rivers and lakes of that country.

In the present age, when discoveries in science, and improvement in the arts, meet with general encouragement, it is surprising, and to be regretted, that nothing has been done towards procuring Mr. Bell a permanent remuneration for the great advantages which this extensive community, as well as the whole kingdom, derive from his indefatigable exertions:— a consideration which becomes the more imperious, when we reflect on the unwearied toil, and anxiety of mind, as well as the loss of health and property, which he has experienced in the completion of so important a national discovery.