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COLLEGE OF ENGINEERING
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Technical Report

A REVIEW OF THE DIFFERENTIALLY SUPERCHARGED DIESEL ENGINE

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OBJECT

The object of this investigation is to examine the material recently published on the subject of the differentially supercharged diesel engine in the light of the requirements of the U. S. Army. The material of Ref. 1 forms the basis for the work.

ABSTRACT

This report examines the principle and operating characteristics of the Differentially Driven Supercharger and compares it with other combinations of displacement compressor drives.

The special torque characteristic achieved is due to the combination of torque converter and transmission placed behind the differential gear, rather than to the engine performance itself. A turbo-charged engine fitted with similar 2-speed transmission and converter would provide equal or superior torque curves at low vehicle speeds; at high vehicle speed the curves would be slightly lower and the engine would have far superior fuel economy.

INTRODUCTION

During the last year or two the subject of flexible versus responsive engines has been a project of analysis under the terms of the subject contract.

The material covered by Ref. 1, "Some Experiences with a Differentially Supercharged Diesel Engine," is pertinent to this contract. The paper does present data and tests which, at first sight, show a practical application in the field of responsiveness employing more or less conventional engines, superchargers, etc.

Since the interaction of the various units in the proposed scheme of operation is somewhat complicated, a careful review of the subject matter presented in Ref. 1 was carried out in order to understand completely the various interrelated factors producing the final results plus the contributions of the different items of the transmission to these results.

DESCRIPTION

The equipment finally employed by the authors of Ref. 1 consists of a typical diesel engine developed for operation at the rating desired, a compression-type positive displacement supercharger of the required capacity, and a planetary gear set connecting the engine, supercharger, and output shaft of the power package. In the output shaft of the planetary gear was included a conventional torque converter for multiplication of the torque output, followed by a two-speed gear.

The engine crankshaft was connected to the epicyclic gear via the planet carrier, while the output shaft was driven by the ring gear, with the supercharger connected to the sun gear through a gear train to give the desired supercharger speed. Since there was no fixed element in the epicyclic gear, the speeds of rotation of the various units is indeterminate unless complete details of the input torques and speeds of two of the three elements are given.

Under the conditions of operation for any given engine speed and load, the epicyclic gear becomes a torque divider, and the engine output torque is divided proportionally between the supercharger and the output shaft. The speeds of these two units then adjust themselves until the required torque conditions are met, in conjunction with the resulting speed of the vehicle.

The complete diagrammatic arrangement of the epicyclic drive and torque converter is shown in Fig. 1 (taken from Ref. 1).

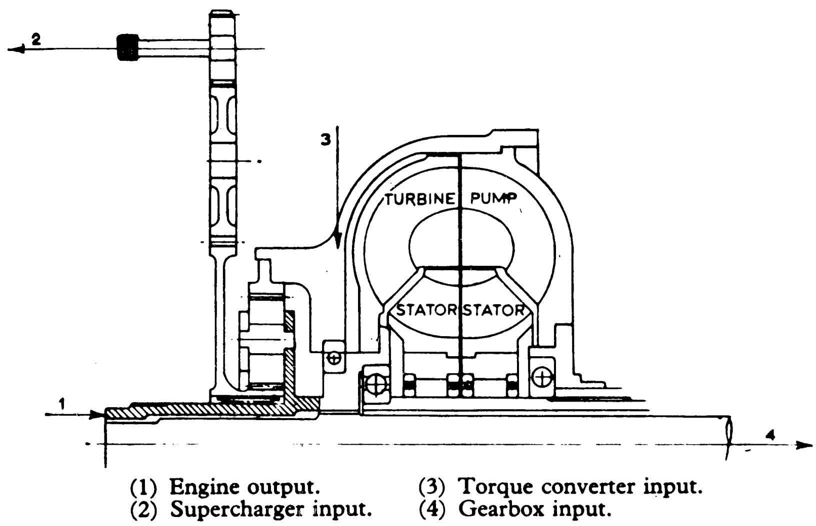

Fig. 1. Combined differential and torque converter arrangement (reproduced from Fig. 8 of Ref. 1).

PART I

DIFFERENTIALLY DRIVEN SUPERCHARGER

A. OPERATION

When applied to a road vehicle the operation depends upon the following factors:

(1) Tractive Effort

When operating on a definite gradient, the road speed and vehicle resistance will determine the output and speed requirements from the torque converter. This will control the input to the converter, which in turn is the output power of the epicyclic gear. It follows that the torque converter can be neglected in the approach to the analysis of the engine, supercharger, and output shaft combination of the planetary gear and their speeds and torque relations.

However, to solve the problem for any one road speed, the tractive effort is an essential requirement to any solution of the distribution of load and speed to the various units. Alternatively it would be possible to solve the problem for the complete range of engine speeds and loads capable of being produced by the power plant, thereby obtaining a performance map from which any unique solution could be obtained.

(2) Supercharger

In any case, if the complete details of any one set of engine or road conditions are specified, a second set of parameters must be fed into the problem to obtain the solution for the speeds and powers of the three units. The two sets of parameters could be the engine performance and supercharger maps (which would tie up the supercharger torque with the engine pressure ratio, the speed developed, and the air flow requirements of the engine); or the engine and road tractive effort (which would set the supercharge conditions). With a given supercharger the former seems preferable.

(3) Engine

As indicated earlier, the vehicle road speed and tractive effort could be the starting point, in which case the solution obtained would be a definite point on the performance curve. Alternatively the engine speed and load could be assumed and its output torque established for any assumed pressure ratio of the charger. This would determine the torque and air flow requirements of the latter, and would in turn (via the supercharger map) establish the speed of the supercharger. The output shaft torque is also known as soon as engine torque is established; and when the engine and charger speeds are known the output shaft speed follows. The output torque and speed would then be fed to the converter and compatible road speed conditions would have to be determined.

By the above means a complete map of the unit's performance could be obtained, from which any actual set of road conditions could be read off. Whatever approach is used, however, more information must be available to make a detailed study of this drive than is provided by Ref. 1. If the epicyclic drive was set up on a computer, the overall map could be obtained easily.

In order to establish the best procedure for obtaining the complete operating range of such a unit, it appears that some first-hand experience of the methods would be necessary; and that to start with, a few sample conditions would have to be solved.

B. ENGINE TEST RESULTS

The final engine performance curves, Figs. 11 and 12 of Ref. 1, have been replotted in Figs. 2 and 3. Other interesting data calculated from the material available has been added to these figures, such as Supercharger Power and Torque, Supercharger Efficiency, Engine Output Torque, Input Torque to Converter, Overdrive Ratio, and specific fuel consumption based upon net engine output. The fuel consumption curves have been changed from pints/B.H.P./hr to lbs/B.H.P./hr, using a fuel of S.G. = 0.86; this permits greater ease in comparing them with typical U.S. figures.

Since both engine and gear output values are shown in Figs. 2 and 3, the corresponding points have to be numbered; point (1) on engine corresponds to (1) on output and so on.

Looking at these results first from the basis of engine performance, one is immediately struck by the very low specific fuel consumption of this rather small engine for such a high speed of rotation, e.g. 0.394 lbs/B.H.P./hr at 3800 R.P.M. when using a mechanically driven displacement type supercharger with a 1.55:1 pressure ratio. At the other end of the scale there is a S.F.C. of 0.318 lbs/B.H.P./hr at 1350 R.P.M. with a supercharge ratio of 2.45:1.

The above values on a B.H.P. basis were obtained by taking the S.F.C. as given for the output shaft and transposing to the engine input power to

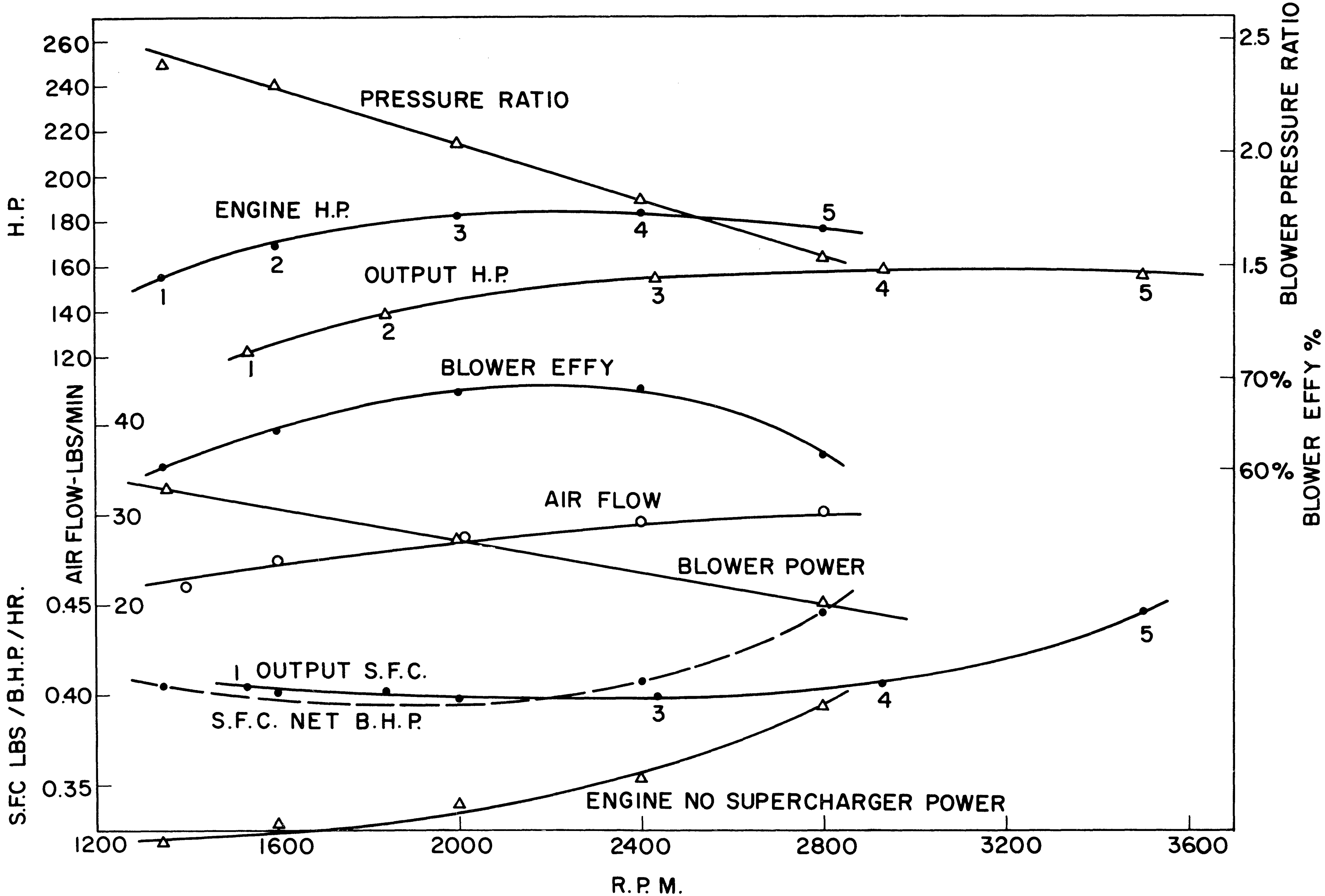

Fig. 2. Differentially charged engine performance.

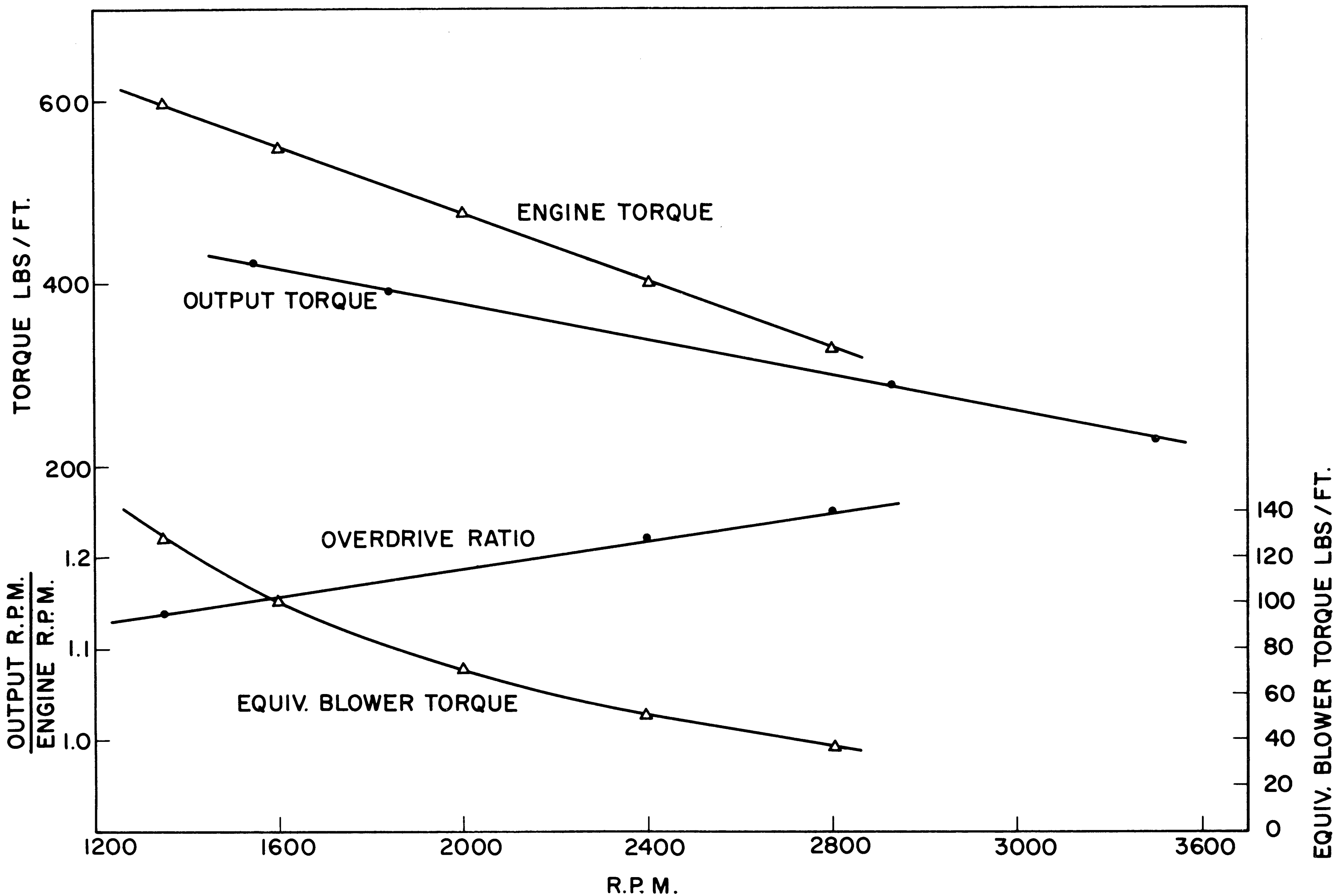

Fig. 3. Output from differentially driven unit.

the epicyclic gear, so that the engine S.F.C. was put on a B.H.P. basis. Fuel consumption values of the order of 0.315 lbs/B.H.P./hr have been achieved in larger engines operating at lower speeds, but not with the power-consuming charger of the type used here or with such high pressure ratios.

Looking over the general engine arrangement it is seen that the engine output in B.H.P. is fed to the epicyclic gear and that the power required to drive the supercharger is taken off this gear box rather than directly off engine. One is thus inclined to assume that what is meant by engine B.M.E.P. or B.H.P. is the gross power without the loss due to the charger. If this is the case then a second S.F.C. curve can be added to Fig. 2 and labeled "S.F.C. net," which is perhaps a better representation of the actual engine performance from a fuel standpoint. This approach is confirmed if it is remembered that the efficiency of the epicyclic gear is high; then if this efficiency is assumed to be 100%, the following relationship exists:

Engine Input = Output H.P. + Blower H.P. For the conditions given in Fig. 11, Ref. 1 for the 1350 R.P.M., this relationship becomes 155 = 122 + 33 at 2800 R.P.M., 176 = 156 + 20; in other words, this equality is satisfied as closely as readings of small diagrams permit. It follows that the correct S.F.C. curve for the engine should be that dotted in Fig. 2 and labeled "net"; this now corresponds with the curve for the output shaft of the differential except that the R.P.M. is reduced to that of the engine. This performance curve is more acceptable than the one given and fits in with the other data presented.

C. EPICYCLIC GEAR

The presence of the epicyclic gear, as has already been pointed out, acts as a torque divider and in the case under consideration also acts as a speed step-up, giving an output shaft speed of 3500 R.P.M. at an engine speed of 2800 R.P.M., and an output shaft speed of 1535 R.P.M. at an engine speed of 1350 revs. This means that the output torque applied to the torque converter is less than it would be if the engine was directly connected. Thus at first sight the presence of this gear seems to be a disadvantage, since the engine torque needs amplification, not reduction; and the torque converter has a bigger job to do as a result. This epicyclic gear is more important for what happens to the supercharger drive than for the torque output. Its presence enables the manifold pressure ratio developed to increase from 1.5 approx. at 2800 R.P.M. of the engine to about 2.5:1 approx. with an engine speed of 1350 R.P.M. This increasing ratio is in place of the more or less constant pressure ratio that would result if a similar fixed drive supercharger was employed.

As a result of this speed change of the supercharger, the engine air flow rate varies from 30 lbs/min at 2800 R.P.M. to 23 lbs/min at 1350 revs. This is in place of the rate of 30 to 14.5 lbs/min that would result if the ratio had been kept constant. Since horse power and air flow are almost synonymous, the result is a power increase at 1350 R.P.M. of about 60% for a given speed compared with a fixed speed drive; this also means a 60% increase in torque at the engine output shaft.

This change in engine speed from 2800 to 1350 R.P.M. is accompanied by a reduction in the converter output shaft speed relative to the input of 1.25 to 1.14:1, which means a greater variation in torque at the output shaft. The change in engine output can be measured by the change in B.M.E.P. for the range of engine speed given. Inspection of Fig. 11 of Ref. 1 indicates that this change of equivalent B.M.E.P. at the output shaft is from 98 psi at 2800 R.P.M. to 179 psi at 1350 revs. In other words, the percentage increase in the output torque is 85% approx.

The main function of the epicyclic gear is considered to be that of providing a torque increase to the supercharger drive as the road torque requirement increases. This increase in supercharger torque (and accompanying speed) is responsible for the production of an increasing manifold pressure ratio as road torque increases.

An additional set of conditions resulting from use of the epicyclic gear needs investigation; these are the conditions of stall by various obstacles which can occur with military vehicles under combat conditions, a type of service which is not encountered in normal road vehicles.

If the details given in Ref. 1 regarding the epicyclic gear are assumed and the condition of stall of the output shaft is investigated, it will be found that the sun gear driving the supercharger will run at 3.33 times engine speed. The sun gear speed is stepped up to the required supercharger speed which, for an engine speed of 2800 R.P.M. with 3500 R.P.M. of the output shaft, has a speed of 4500 R.P.M. approx. (obtained by scaling Fig. 3 of Ref. 1). This magnitude is about what would be expected and will be accepted as close enough to the actual value. It follows that if the output is stalled with the engine at 2800 R.P.M. the supercharger speed will be about 56,000 R.P.M.; with an engine speed of 1400 R.P.M. it will be about 28,000. These speed conditions would not be achieved in actual operation except under lock-up. Normally stall would release the lock-up device and the torque converter would permit some rotation of the epicyclic output shaft. Under such conditions, with the output shaft speed pulled down to say 500 R.P.M., then for an engine speed of 1400 R.P.M. the supercharger speed would still be at about 20,000 R.P.M.—practically double its speed of 10,200 R.P.M. during normal operation at 1400 R.P.M.

This condition, met in military operations, would need serious consideration from all aspects if such a drive was contemplated. The completely stalled speeds are far higher than this type of charger will tolerate; even the 20,000 R.P.M. under low engine speed conditions is generally too fast for such a compressor.

D. TORQUE CONVERTER

The presence of the torque converter on the output shaft of the epicyclic gear has no function as far as the responsiveness of the engine is concerned. In fact, if the same engine performance could be obtained without the epicyclic gear being used, then a torque converter of about 2.5:1 ratio in place of the 3.6:1 of the present set-up would provide the same range of torques as that of Ref. 1. The torque converter provides the necessary range of torques which would have to be produced by a transmission otherwise.

The converter introduces to the system some additional losses while it is in action, but the provision of the lock-up device cuts out the action at about 1/3 speed apparently. This point of lock-up is perhaps suitable for a road vehicle but an off-the-road vehicle would probably need greater range.

The range of torques required to give the tractive effort curve of Fig. 15 of Ref. 1 are proportional to the effort, viz., 1000 to 6500 in high gear and 1800 to 11800 in low gear a ratio of 6.5:1 approx.; in each case this is the ratio of full speed to stall torque. At the speed of maximum engine torque (1350 R.P.M.), which corresponds to 27 mph in high gear and 15 mph in low, the torque ratios are 1.8:1 approx. This ratio of course corresponds with the gear ratio employed and indicates that the torque converter has a 3.6:1 ratio as stated.

E. TRACTIVE EFFORT

The tractive effort curve of Fig. 15 of Ref. 1 gives a very interesting picture of the responsive abilities of the engine and transmission combination.

It is now proposed to examine what part of this effect is due to the differentially supercharged engine and what part is due to the transmission. As pointed out above, the engine is responsible for a torque ratio of 1.8:1 and the converter for a 3.6:1 step-up in torque. This indicates that the converter is the main item in the drive that accounts for the shape of the performance curves.

Let the performance of the turbo-charged 5.95/1 engine shown in Fig. 15 of Ref. 1 be examined on the basis that it was fitted with the same ratio torque converter and only two speeds were employed, as with the differentially charged engine. The top gear of 1/1 and the second gear of 3.78:1 are selected for investigation. Taking the tractive effort at the low-speed end of the dotted curves of Fig. 15 (i.e. 20 mph in top gear and 5 mph in 2nd) the tractive effort would become 3250 lbs in top gear and 11500 lbs in 2nd gear when fitted with the converter. The corresponding values for the differentially charged engine are 2200 lbs in high gear and 7000 lbs in low gear. In other words the turbo-charged engine of Ref. 1 fitted with a 3.6:1 torque converter would out-perform the differentially charged engine fitted with the same torque converter. In addition to this its fuel consumption would be at a considerably lower value, as will be shown later.

It is true that in lock-up condition the tractive effort at full speed would be a little lower, but as shown by Fig. 15 of Ref. 1 the top speed available is only some 3 or 4 mph lower. Without a complete plot of the torque converter performance the exact relationship between the two is difficult to establish; however, the tractive effort of the turbo-charged engine has been established at between 1.5 to 1.6 times that of the differentially charged engine with the converter in full operation. It follows that superior performance will be achieved over some range of speeds before the curves will cross over and reverse the trend. The point of lock-up would of course definitely switch the performance from one set of curves to the other.

F. CONCLUSIONS

It is possible to draw certain conclusions from this partial analysis of the differentially supercharged engine. These are as follows:

1. The use of a displacement type blower for supercharging has some special features, generally well known, for maintaining tractive effort at low speed at the expense of fuel consumption.

2. The speed increase of the displacement charger to produce any desired increase in supercharge ratio is low, in place of a high speed change for the centrifugal type.

3. The displacement blower geared to the epicyclic gear, as given in Ref. 1, provides the necessary automatic speed increase of the charger for high manifold pressure ratios as tractive effort increases.

4. As tractive effort reduces, the speed of the charger would slow down, eliminating some of the losses that would occur

with a fixed drive supercharger. Part load conditions would need to be examined to establish the exact savings if any.

5. The engine arrangement as shown in Ref. 1, still needs the application of the torque converter to provide the necessary output torque for satisfactory operation over the desired speed range.

6. A typical present day turbo-charged engine fitted with a two-speed gear box and torque converter, similar to that provided for the displacement machine, would give at least the same type of tractive effort curve; in fact it would be superior over the low-speed, high-torque range but a little inferior at high speed.

7. The turbo-charged engine would have improved fuel economy relative to the displacement machine.

8. The combination of engine, supercharger, converter, etc. of Ref. 1 was probably selected to give a rather special performance characteristic under average conditions over a given typical paved highway operation.

9. As a result of (8), direct relation with Army equipment in off-the-road application cannot be drawn exactly.

PART II

A TURBO-DRIVEN VERSION OF THE DIFFERENTIALLY DRIVEN SUPERCHARGER

It is now proposed to examine what would happen if a turbine was added to the system to recover the exhaust gas energy and feed it back into the system. This will be done in two ways:

1. A turbine will be geared to the system as set forth in Ref. 1, without any other charge.

2. The displacement charger will be removed from the epicyclic gear and driven by an exhaust gas operated turbine. The epicyclic gear is then removed from the drive shaft and replaced by a two-speed gear box and torque converter of comparable ratio with those employed in Ref. 1.

A. TURBINE ADDITION

In this case no change will be made to the power package of engine, gear, and converter except to add the available energy of the exhaust in the system. To do this, extra cost would be involved, but little extra weight or bulk. The main cost increase would be in the addition of the turbine and necessary gears.

It will be assumed that the exhaust gas temperature and mass flow will remain the same as that given in Fig. 12 of Ref. 1, as will the inlet manifold pressure. The exhaust back pressure will be increased to a value of $P_e = 0.85\ P_m$ where P_e = exhaust manifold pressure and P_m = pressure in the intake. This increase in exhaust pressure will reduce the available m.e.p. by a definite amount given by:

$$\left.\begin{matrix}\text{Engine B.M.E.P.}\\ \text{with turbine}\end{matrix}\right\} = \left\{\begin{matrix}\text{Engine B.M.E.P.}\\ \text{with atmospheric exhaust}\end{matrix}\right\} - (0.85P_m = P_o)$$

where P_o = atmospheric pressure.
(see page 13 of Ref. 4 for this relationship)

The data of Fig. 11 of Ref. 1, will now be corrected in this manner, the available turbine work in the gases calculated, and new S.F.C. figures obtained. The resulting data is shown in Table I and Fig. 4, where comparison of the system of Part I can be made. It is seen that at any engine

TABLE I

PERFORMANCE WITH TURBINE ADDITION

Engine, rpm	Output, rpm	Corrected		Exhaust Gas Temp., °R	Total Gas Flow, lb/sec	Isentropic Turbine Power, hp	Turbine Efficiency, %	Turbine Output, hp	Supercharger Power, hp	Net Output with Turbo, hp	Net hp Diff. Super.	SFC, lb/BHP/hr		F/A Ratio	Output Torque, lb/ft	
		BMEP	Cylinder hp Output									Diff. Super.	Turbo Super.		DDS Shaft	Turbo Shaft
1400	1605	235	148.5	1300	0.389	32.4	78.0	25.3	32.5	141.3	123.5	0.405	0.359	0.037	410.0	530.0
1600	1840	221	158.5	1350	0.423	34.2	79.0	27.0	30.7	154.8	138.0	0.397	0.354	0.037	390.0	508.0
1800	2110	207	165.5	1400	0.466	36.2	80.0	29.0	29.0	165.5	149.0	0.396	0.352	0.0364	363.0	482.0
2000	2430	193	173.5	1448	0.480	35.2	81.0	28.5	27.0	174.0	155.0	0.395	0.352	0.0374	336.0	458.0
2200	2650	178	174.5	1498	0.493	34.8	82.0	28.5	25.5	177.5	157.0	0.398	0.352	0.0374	317.0	424.0
2400	2930	164	175.5	1545	0.502	30.2	83.0	25.0	23.5	177.0	158.5	0.408	0.365	0.0387	289.0	387.5
2600	3210	149	174.0	1593	0.511	26.0	82.0	21.3	21.7	173.6	158.0	0.423	0.385	0.039	260.0	352.0
2800	3500	135	169.7	1644	0.519	21.8	81.0	17.6	20.0	167.2	156.0	0.445	0.415	0.0388	229.6	317.0

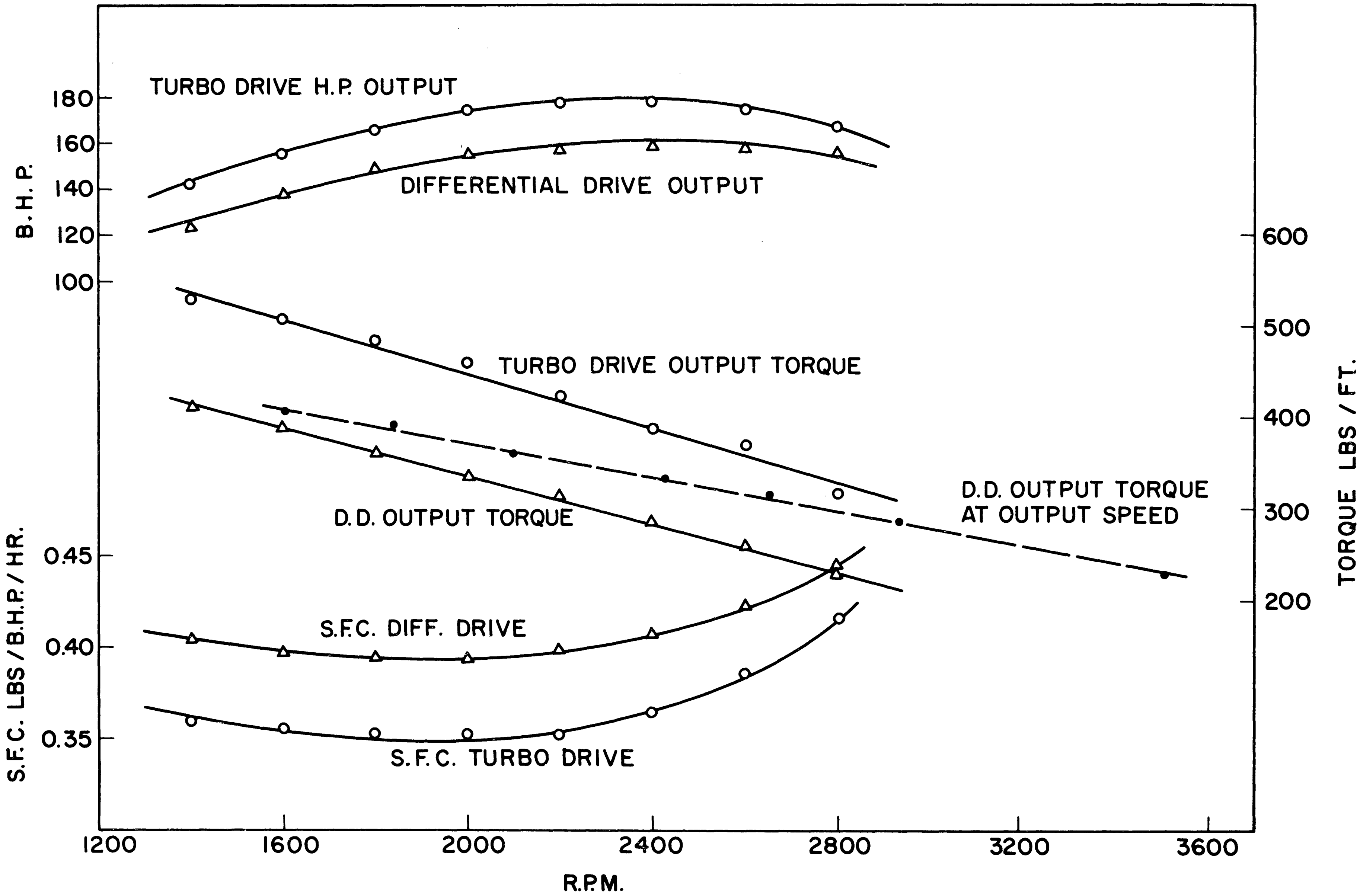

Fig. 4. Comparison of differential and turbine applications.

speed the S.F.C. is some 8 to 10% lower, accompanied with an 8 to 12% increase in horsepower in favor of the turbo application.

On Fig. 4 the differentially charged engine has been shown in two ways: (1) the output shaft torque plotted on an equivalent engine R.P.M. base, and (2) the output torque plotted on output shaft speed. The latter of course is the plot to employ if H.P. is desired.

B. DISCUSSION

The data presented in Figs. 2 and 3 records the engine and differential drive gear performance material, but does not include the torque converter multiplication or the effect of either of the gear ratios employed in the transmission. This has been done since the converter or gear ratio can be equally applicable to any engine or epicyclic output. For example Ref. 1 states that the converter has a 3.6:1 ratio while the two forward speeds are of 1:1 and 1.8:1 ratio. It follows that the overall torque multiplication of the output is 3.6 or 6.48:1 plus reduction ratio of the differential axle given at 7.17:1. These values coupled with the 1.14:1 reduction of ratio built into the epicyclic gear give the differentially charged unit an overall torque multiplication of 22.6:1 when in high gear [based on engine torque developed at the point of maximum torque (1350 R.P.M. of engine)], and of 40.8 when in low gear at the same engine speed.

At the high engine speed, where the epicyclic provides a 1.25:1 ratio, these overall values become 20.65:1 and 37.2:1 respectively.

If these same overall torque ratios were provided by the epicyclic gear when the turbine was added, then at maximum torque the torque multiplication factors would be 25.6:1 and 46.2:1 in place of the 22.6 and 40.8 for the differential charger. At the maximum engine speed these values would be 23.75 and 42.75 in place of 20.65 and 37.2 respectively. That is, the turbine addition to a charger of the same type, efficiency, etc. as that employed in Ref. 1 would give tractive effort performance which is superior to that being examined.

If the second combination was employed (i.e. the epicyclic gear was removed with the engine driving directly into the torque converter) while the turbine drove the existing supercharger through a suitable gear, then the performance would remain unchanged to all intents and purposes since there is enough power in the exhaust gases to drive the charger, and the losses in the epicyclic gear are low. However, the weight and bulk of the overall machine would be reduced greatly.

It should be noted, however, that Table I shows that at both ends of the speed scale (viz. 1400 and 1600 R.P.M. at one end and 2800 at the high speed end), the turbine power is insufficient to provide the work required by the charger. The reason for this is a relatively low turbine efficiency employed in the calculations, plus a low efficiency of the charger 60 to 61% (see Fig. 2). In making the calculations the additional power required for charger operation was subtracted from the engine output, hence the predicted performance is still one that could be achieved. In Fig. 5 is provided a modern S.R.M. compressor performance curve with a principle similar to that of Ref. 1, where efficiencies of 70 to 76% are recorded. If this range of values can be retained in the size required for the problem in hand, the deficiency in power would be more than overcome and operation under all conditions would then be a possibility. It is considered that the supercharger can be driven by a turbine and gear set if a suitable design is developed.

The use of the epicyclic gear seems to be based upon the need for a speed increase of the charger as load increases. It does, at the same time, provide a higher output shaft speed to the torque converter which in turn reduces the size of this element somewhat. The increased input speed to the converter means the need for a greater torque multiplication factor in the converter since the input torque reduces with speed increase. It follows that the turbine drive charger would not need so great a multiplication as the differential drive; this is shown in Fig. 4 by the increased torque values in each case when the same drive was employed.

It seems that the paper under discussion has demonstrated that the torque output of a diesel engine can be increased by 1.75:1 approx. as the engine speed reduces if the degree of supercharging is increased from a pressure ratio of 1.5:1 at full speed to 2.5:1 at roughly half speed. The differential gear is one means of securing the required speed change in the supercharger drive. The required speed change is a rather moderate amount when the displacement type charger is used. The same effect could be obtained with a centrifugal supercharger if the required speed range could be obtained. Alternatively a turbo-charger of conventional type could produce the same effect by variable turbine nozzles, provided a broad enough range of air flow without surge could be obtained.

The main disadvantage of the proposed scheme appears to be that the displacement supercharger, as fitted, is an uneconomical means of producing the required result, so far as S.F.C. is concerned. The selection of the means to be employed to produce the desired characteristic will thus depend upon the relative importance of fuel costs in the overall operation.

Leaving the general principle for a time and considering the details of the power train and their contributing effects, it can be stated that the torque converter plus transmission is still the most important contributor to the overall results. The engine provides a minor role since a fixed drive displacement supercharger would give a torque at half speed

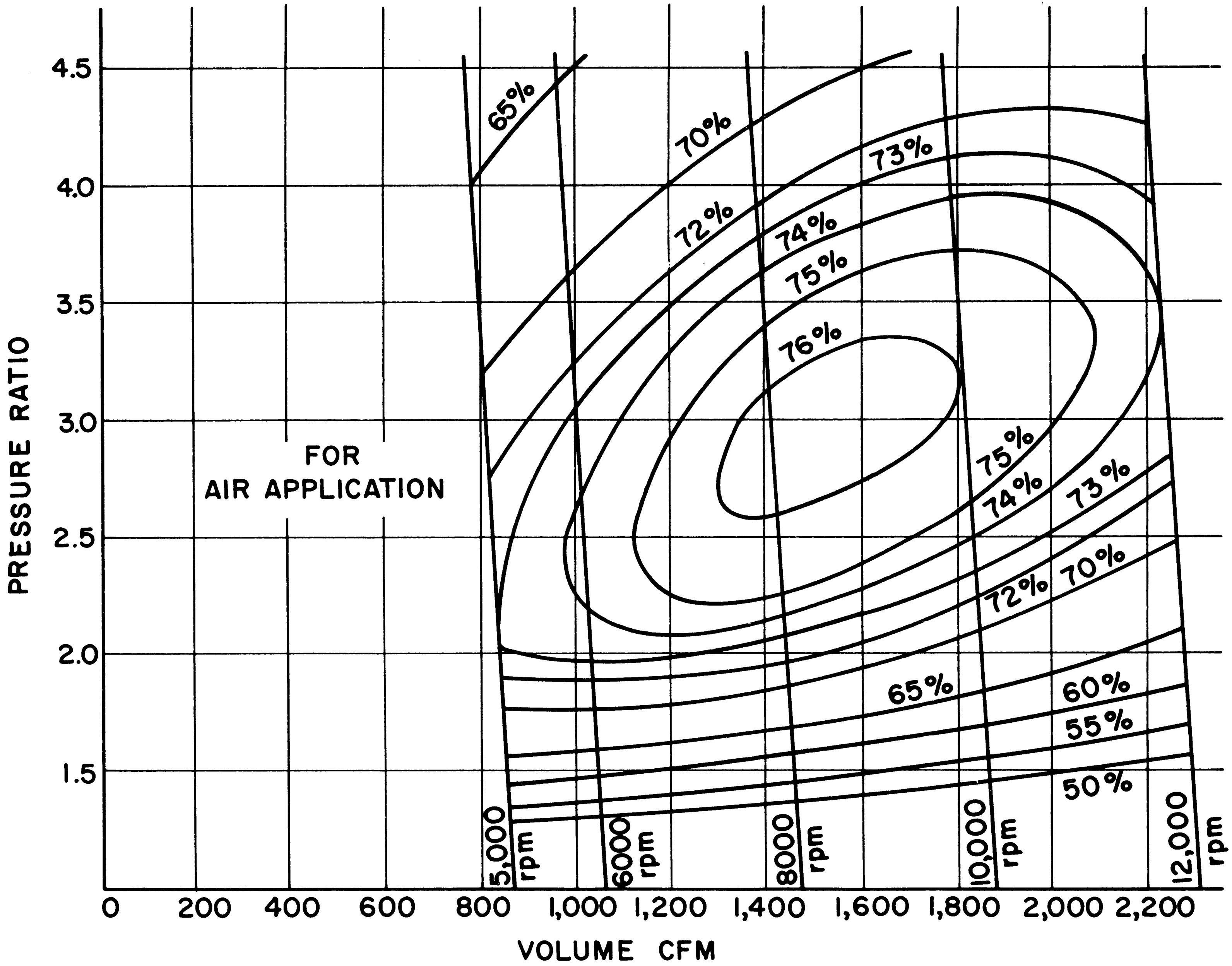

Fig. 5. Modern S.R.M. compressor performance.

of about 1.2:1 of that given at full speed as a result of its characteristics (see Ref. 2), against 1.75:1 of the epicyclic gear.

Again if the level of performance of the subject engine at full speed was satisfactory for a military application, then there would be no need for the use of the differential drive. The subject engine only produces a net B.M.E.P. of 98 psi at the output shaft for a manifold pressure ratio of 1.55:1. Now a naturally aspirated diesel engine of good design has no difficulty in producing 95 to 110 psi mean pressure, i.e., more than the supercharged one. The turbo charged at 1.55:1 (Ref. 3) gives a possible output of about 170 i.m.e.p or 140 b.m.e.p., i.e., 42% more torque to start with. At the present time the typical combat vehicle has a b.m.e.p of 200-500 psi, or more, at full speed. All that has to be done to be in competition with the engine being considered is to cut the full speed rating to say 100 psi and keep the 250 psi approx. at half load by turbo speed increase; this is done by the use of a variable nozzle turbine design. The penalty in using this scheme would of course be the large increase in bulk and weight of the power plant if the same maximum H.P. was desired at full speed as was presently available (700-1100 H.P.).

In considering the improved fuel economy shown by the turbo drive of Fig. 4, it should be remembered that no change has been made to the engine performance by the addition of the turbine, except the effect of the added back pressure. This is not a turbo-charged engine; it is still a positive displacement charger of exactly the same efficiency as that employed in Ref. 1. The change is that it is driven off the exhaust gases in place of the engine output. A turbo-charged machine would give still further reduction in fuel consumption. The turbine speed would remain relatively constant for the displacement machine, but would need to be increased for the turbo-charged one as engine speed fell; a wide flow range and variable nozzle would undoubtedly be necessary for this latter type. The Continental Motors Supercharger proposal is one which would approximate the necessary conditions.

One feature is important when the epicyclic drive is employed: the required change of speed of the supercharger would be almost instantaneous, following engine speed very closely, thus keeping air flow in step with requirements. In the case of the turbine drive with variable nozzle control there would be some lag of air flow behind engine needs and thus the possibility of smoke. However, this lag would be far less than that of the present fixed-nozzle turbo-chargers; moreover, high manifold pressure exists before throttle is changed.

C. THE DISPLACEMENT COMPRESSOR WITH SLIP

In considering the problem under discussion, the same degree of responsiveness gained by the epicyclic gear of Ref. 1 can be achieved more simply with a great reduction in complication, weight, and bulk. This method is discussed in Ref. 2, where it is shown that a 1.8:1 torque increase can be achieved by the simple addition of a variable fill fluid coupling to the conventional drive gears of a displacement compressor. In this case there is little or no additional weight or bulk involved, and at the same time there is improved economy at part load relative to the fixed drive charger.

D. CONCLUSIONS

This examination of the various factors entering into the performance characteristics of the differentially driven supercharger permits the following conclusions:

1. The overall system (engine, epicyclic gear, torque converter, and transmission) examined here does produce an operating characteristic having special advantages.

2. At the same time it seems that the design has been specialized to suit the application involved, whereas the competitive turbo-charged system to which it was compared was not as specifically tailored to the application.

3. The operating characteristics achieved were mainly the result of including the 3.6:1 torque converter rather than the result of engine responsiveness achieved.

4. The inclusion of a 3.6:1 converter into the turbo-charged engine performance shown would, when coupled to a two-speed gear of suitable ratios, almost duplicate the tractive effort of the differential drive unit plus its two-speed gear, and would have superior fuel economy.

5. The fuel economy with the displacement compressor is adversely affected with the unit as at present designed.

6. Improved power and torque output is possible by providing for exhaust energy recovery via a turbine driving to the supercharger in place of the epicyclic gear.

7. The slip drive technique of Ref. 2 should be considered in competition with the present scheme in order to evaluate their relative merits.

8. The differentially driven positive displacement machine has one important advantage, viz. it is free from the surge problems that plague centrifugal machines.

9. The turbine-driven displacement compressor, fitted with variable nozzles, is also free of surge.

10. The differentially driven machine as described has some special features which suit it to regular truck operation over the expressways, if the reduced fuel economy can be justified.

11. Unless a considerable increase in the level of manifold pressure ratio can be effected, the resulting power plant would be unsuitable for such military purposes as main battle tanks, due to the large bulk and weight.

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