THE UNIVERSITY OF MICHIGAN
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Technical Report

ENGINE PERFORMANCE WITH VARIOUS COMBINATIONS OF DIRECT DRIVE SUPERCHARGERS AND TURBOCHARGERS

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OBJECT

The object of this analysis is to establish the expected engine performance when both engine-driven superchargers and turbochargers are combined in series, and to extend the performance data of Ref. 1 when the supercharger is driven in ways which permit some responsiveness to be achieved.

ABSTRACT

The purpose of this report was to examine the ways and means by which the disadvantages of the turbocharged engine (low maximum power output at low speeds plus smoke during acceleration) might be overcome by combining it with the displacement compressor, and to examine the means by which the part-load economy of a direct-drive, positive-displacement machine could be improved without sacrificing its high torque at low speed.

Means were also examined by which some additional degree of responsiveness could be obtained with the displacement machine without sacrifice at all other load conditions.

The results obtained show that the above objectives can be achieved, but that the turbocharged engine still gives the best overall fuel economy. Secondly, responsiveness plus increased economy at part-load can be achieved when using the positive-displacement supercharger, provided means are incorporated for variable-speed operation relative to the engine of the supercharger.

INTRODUCTION

The methods employed to obtain the data for this report are those given in Refs. 1 and 2. Examples of the application of these methods are given in those references and the details of the calculations will not be repeated here, only the final results will be recorded.

The report will be divided into two sections: Section I. The mechanically driven supercharger arranged for responsiveness; Section II. Combinations of the mechanically driven supercharger with turbochargers.

I. THE MECHANICALLY DRIVEN SUPERCHARGER ARRANGED FOR RESPONSIVENESS

In this section a comparison will be made between engines fitted with a compression type displacement machine, such as the Bicera and Lysholm, driven in the following manner:

Scheme (a) Direct constant-ratio drive

Scheme (b) Direct variable-ratio drive (by fluid coupling)

Scheme (c) Direct responsive-ratio drive (by fluid coupling).

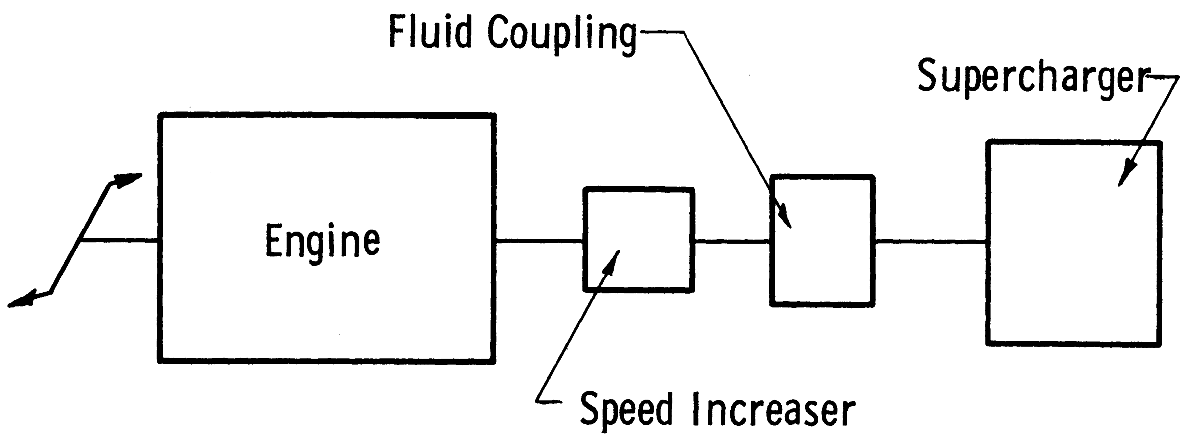

Fig. 1. Diagrammatic arrangement of variable-ratio supercharger drive.

Figure 1 is a diagram of the scheme. It consists of an engine driving the supercharger through any desired step-up ratio, which in turn drives one element of a fluid coupling, the second element of which drives the compression displacement supercharger. With this arrangement, Scheme (a) can be obtained by locking the fluid coupling, while (b) can be achieved by varying the amount of oil in the coupling; Scheme (c) also can be obtained by a variable amount of oil, but in a different manner from that of (b), as will be seen later. This type of drive has been selected because it is simple, small, and light weight (when coupled to the high-speed shaft); it is well known because of the years of experience with it in similar engine drives and its maintenance requirements are also well known. As a result of these factors, little development work will be required.

The arrangement is, of course, still subject to the limitations given in Ref. 1 for this type of supercharger, viz.,

1. Relatively low isentropic efficiency

2. Relatively low maximum speed of rotation

3. Pressure ratios not exceeding 2.0 to 3.0:1 approximately.

A. METHOD OF CALCULATION

In Scheme (a) the gear box ratio is selected to give the required speed of rotation of the supercharger when the fluid coupling is locked. For an engine speed of say 3000 rpm, a 3 to 4:1 ratio would be about the highest employed if the best of the supercharger characteristics are to be achieved. For the same reason, when employing Scheme (b) or (c) the same ratio of 3 to 4:1 would be necessary for the peak performance at maximum rpm of the engine. At part load, however, in Scheme (b) the fluid coupling would be allowed to slip in order to reduce the engine load at any given engine speed, i.e., road speed of the vehicle in any given gear. This means a reduction of the power required to drive the compressor, but also some loss due to slippage of the coupling; however the total power delivered by the engine to the supercharger will be reduced below that of Scheme (a) as the engine load is reduced for any given engine speed. The coupling's slippage, which increases as load is reduced, means a reducing manifold pressure in place of a substantially constant one, as occurs in the case of direct drive. The result is not only reduced work of compression but some lowering of frictional losses also.

In the operation according to Scheme (c) the reverse would be the case, slippage will exist at full speed and load so that the additional losses due to slip are encountered for short periods only or for a different gear ratio. At slow engine speed the relative supercharger speed will be higher, giving increased manifold pressure and thus some degree of responsiveness at low speed.

In order to effect this, the gear box will be provided with a gear ratio which will give, for example, a supercharger speed of 1.5 times that employed in Scheme (a), but the fluid coupling will slip with the engine at 3000 rpm to give the same 1:1 ratio of Scheme (a), followed by no slip at low speed to give the effective ratio of 1.5:1 at such low speed.

These three cases can be covered with one set of nomenclature:

γ_g = built-in ratio of gear box between engine and supercharger without slip.

S = slip ratio of fluid coupling = $\gamma_g N - n/\gamma_g N$.

γ_e = effective gear ratio between engine and supercharger

$= n/N = \gamma_g(1-S)$

n = supercharger speed rpm

N = engine speed rpm

$\gamma_g N$ = maximum compressor speed (no slip)

Then the work of compression can be given by:

$$\text{Work of Compression} = \frac{2\pi\gamma_e Nt}{33000} = \frac{wC_pT_1(R^{k-1/k}-1)}{\eta_c\eta_m}$$

t = compressor torque in lbs ft

T_1 = inlet air temperature °abs

R = compression ratio of supercharger

w = lbs of air per min

C_p = specific heat of air = 0.241 Btu/lb/°F

η_c = isentropic efficiency of compressor

η_m = mechanical efficiency of compressor

The work to be provided by the engine to the driving element of the coupling with slip thus becomes:

$$\text{Work from engine} = \frac{2\pi\ Nt}{33000} = \frac{1}{\gamma_e} \times \text{Work of compressor}$$

$$= \frac{1}{\gamma_e}\ \frac{w\ C_pT_1(R^{k-1/k}-1)}{\eta_c\eta_m}$$

It must be remembered that there is always some slip in a fluid coupling even if completely filled with fluid. This in general amounts to about 3%. It follows that for the fully engaged condition, when an effective gear ratio of say 3:1 is desired, the design ratio γ_g must be 3.09 in place of 3.0, and the slip ratio at maximum compressor speed becomes S = 0.03, which in turn affects the work to be supplied the compressor by the engine, even in full-speed condition.

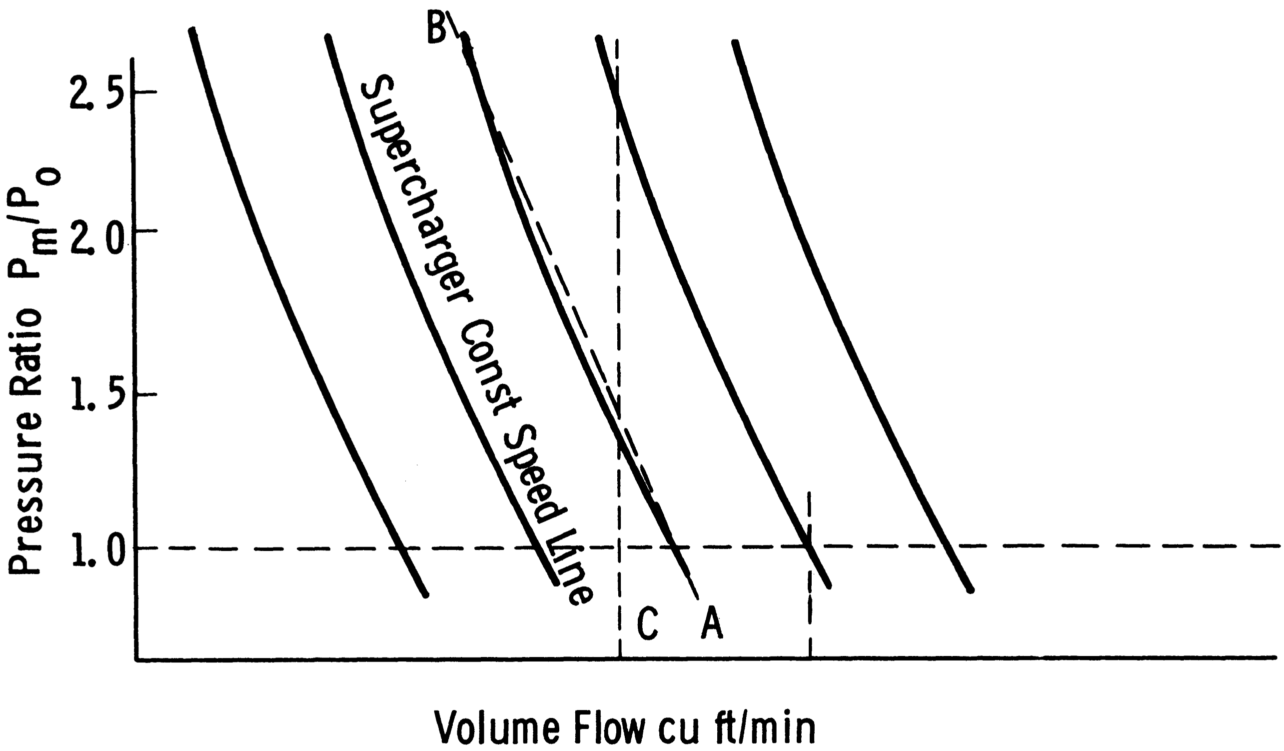

Fig. 2. Typical performance map of a displacement supercharger with compression.

The displacement compressor has a small variation of mass flow as the delivery pressure varies at constant speed, as can be seen from Fig. 2. If it is assumed that the volumetric efficiency and leakage due to pressure are constant, then the delivery would also be constant for any given speed and, to a first approximation (for relatively moderate engine-speed changes), the volume flow in cu ft per min of air to the compressor would be proportional to the compressor rpm.

Accepting these rather limiting assumptions initially, an approximation to the three schemes can be made with a moderate amount of calculation, and one can then judge if a more comprehensive method is justified.

Consider an engine of displacement D cu ins/cyldr, having "n" cylinders operating at N rpm, with a compression ratio of 17:1, then the air required to fill the cylinder and scavenge the clearance space is given by

$$\text{Air flow} = (D + \frac{D}{16}) \times \frac{N}{2} \times n \text{ cu ins/min}$$

$$= 0.53 \text{ nDN cu ins/min}$$

$$= 0.0184 \text{ nDN cu ft/hr}$$

$$= 0.0184 \text{ nDN } \rho_m \text{ lbs/hr},$$

where

ρ_m = density of air in manifold

P_m = pressure of air in manifold psia

T_m = temperature of air in manifold °abs.

It follows that the volume flow at the supercharger inlet becomes

Flow to supercharger at $P_1 = 14.7$ and $T_1 = 540°$ $= 0.0184 \text{ nDN } \frac{P_m}{14.7} \frac{540.0}{T_m}$

$$= 0.676 \text{ nDN } \frac{P_m}{T_m} \text{ cu ft/hr.}$$

With an aftercooler giving $T_m = 200°F$, the flow is simplified to:

Flow to supercharger at inlet $= 0.001025 \text{ nDNP}_m$ cu ft/hr

$$= 0.01506 \text{ nDN}(P_m/P_o) \text{ cu ft/hr.}$$

The lines of constant supercharger speed in Fig. 2 can then be replaced by straight lines, as shown by the dotted line AB. Using this approximation, one can then calculate the performance over varying conditions. Assuming that the supercharger flow is also proportional to the rpm, the performance diagram of the compressor then becomes that shown in Fig. 3, plotted on a percentage basis, where AB represents the volume flow of the compressor at some constant speed.

One can now superimpose on Fig. 3 the engine air flow requirements and, assuming constant volumetric efficiency for any speed (reasonable for highly supercharged engines) with aftercooling to a constant temperature, the engine flow also becomes a straight line. The engine flow lines for different per-

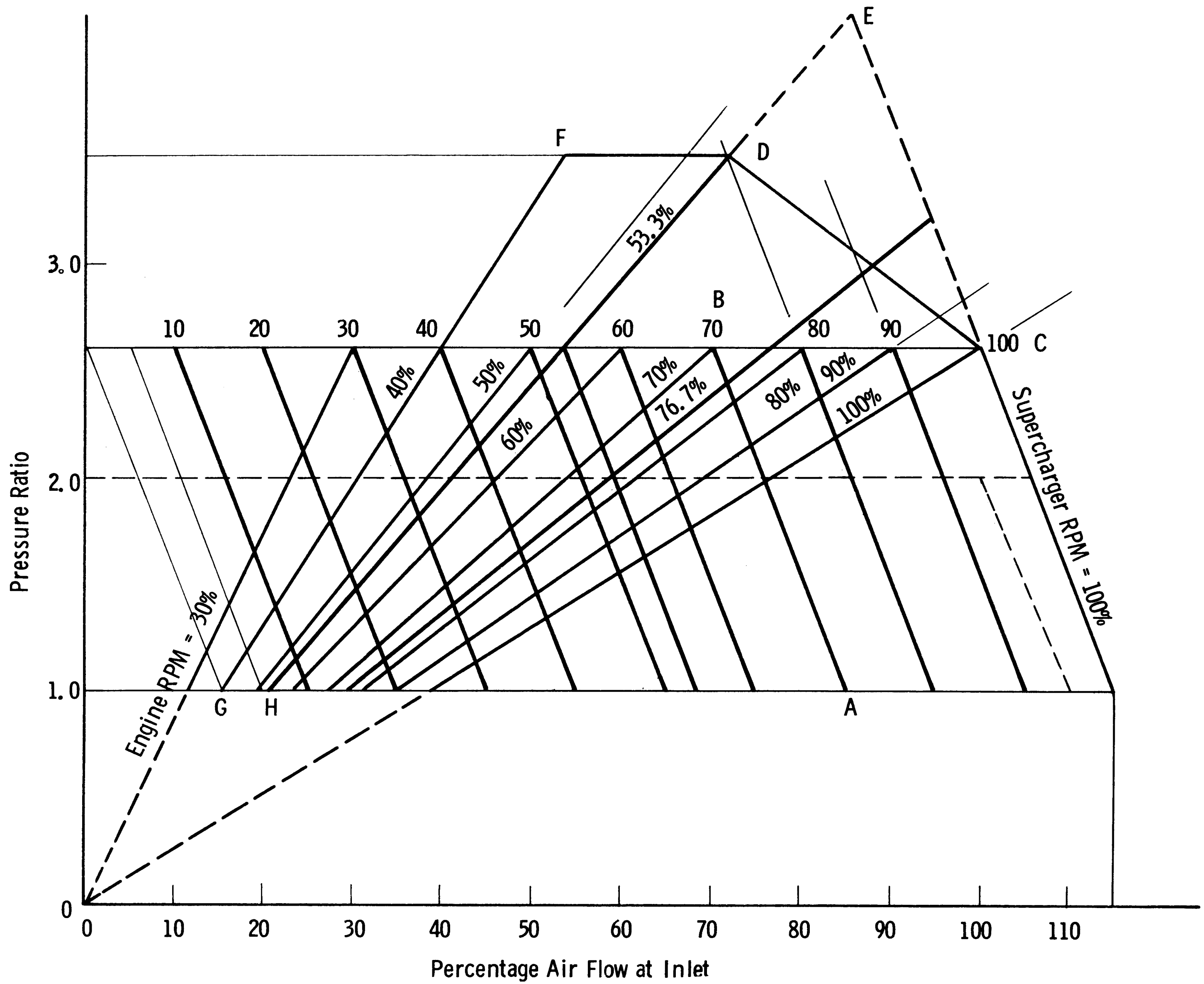

Fig. 3. Supercharger characteristics plotted on a percentage basis.

centages of full speed then become as shown in Fig. 3, the 100% engine flow intersecting the 100% supercharger flow line at the desired pressure ratio (2.6:1 in Fig. 3).

With the assumptions made, the manifold pressure will now remain constant as engine speed is varied if the supercharger is directly geared to the crankshaft with a constant speed ratio. This is indicated in the diagram by the intersection of the 50% engine-speed line with the 50% supercharger speed line at P_m/P_o = 2.6:1 as for full speed, similarly for other speeds.

When using the variable fluid coupling, one can operate the engine at 100% speed while the supercharger is at 50% speed only. In this case, Fig. 3 indicates that the manifold pressure ratio will drop from 2.6:1 to 1.55:1 and the work of the compressor will be reduced, as well as the engine effective mean effective pressure developed. The required question is: Has the fuel consumption of the engine improved for this load condition at 1.55:1 relative to the same load at 2.6:1, as it does with constant compressor speed?

B. CALCULATED RESULTS

A series of calculations was made for pressure ratios of 1.0 to 2.6:1 for each of three engine speeds—3000, 2300, and 1600 rpm—at a constant fuel air ratio of 0.047 (representing full throttle conditions at each speed), and with a constant air temperature of 200°F. The variation in pressure ratio was achieved by variable slip in the fluid coupling drive of the charger. The results are recorded in Table I and Fig. 4.

The required slip in the coupling is determined from Fig. 3, e.g., for 100% engine speed (3000 rpm) and a pressure ratio of 2.0:1, the charger must operate at 72.5% of its 100% speed, because at these two conditions the air flow to the engine is equal to the air delivered by the compressor.

In order to represent completely the operating conditions over the complete range of HP required by the vehicle, the data of Table I and Fig. 4 must be extended to a lower HP range. The low pressure ratio of 1.0:1 of Table I represents the condition of the supercharger slipping to such an extent that it is merely "windmilling" to pass the requisite amount of air to the engine with no increase in pressure. Further reduction in HP can still be achieved by reducing the fuel flow below the F/A ratio of 0.047 by closing the fuel pump throttle. This part-throttle performance with slip is shown in Table II.

In order to compare these data with normal engine operation employing a fixed-ratio drive to the charger (Scheme (a)), the data of Table III were calculated. In this case, the operating line of the charger was that at constant pressure ratio of 2.6:1, since at 60% engine speed, for example, the

TABLE I

FULL-THROTTLE PERFORMANCE WITH SLIP

Manifold Temperature = 200°F (F/A = 0.047)

	Pressure Ratio		1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6
	IMEP (gross)	psi	117	142	167	194	219	244	272	294	324
	Compressor Work	psi	3.3	5.0	10.3	14.7	17.2	25.2	32.9	41.5	52.0
Engine rpm = 3000	Comp. rpm	percent	23.5	33.0	42.5	52.0	62.0	72.5	80.5	91.0	100
	Eff. Ratio (γ_e)		0.235	0.33	0.425	0.52	0.62	0.725	0.805	0.91	0.97
	Work With Slip	psi	6.0	15.1	24.2	28.3	27.8	34.7	40.9	45.6	53.5
	Friction and Cooling Loss	psi	43.0	44.0	45.0	46.0	47.0	48.0	49.0	50.0	51.0
	BMEP (net)	psi	68.0	82.9	97.8	119.7	144.2	161.3	182.1	198.4	219.5
	BHP		161.0	196.0	231.0	283.0	341.0	382.0	428.0	469.0	520.0
	Specific BHP	lb/sec	307.0	318.0	318.0	345.0	368.0	369.0	377.0	379.0	386.0
	Specific IHP (net)	lb/sec	500.0	485.0	464.0	478.0	488.0	479.0	481.0	475.0	476.0
	Air Flow	lb/sec	0.524	0.618	0.726	0.827	0.928	1.035	1.135	1.237	1.345
	Fuel/IHP/hr		0.339	0.350	0.363	0.354	0.346	0.353	0.353	0.354	0.355
	Fuel/BHP/hr		0.552	0.534	0.53	0.466	0.460	0.458	0.451	0.444	0.438
Engine rpm = 2300	Comp. rpm	percent	15.0	22.5	30.0	37.5	45.5	53.5	62.0	68.5	76.6
	Eff. Ratio (γ_e)		0.196	0.294	0.392	0.49	0.594	0.698	0.81	0.89	0.94
	Work with Slip	psi	5.0	17.0	26.3	30.0	29.0	36.1	40.6	46.6	53.5
	Friction and Cooling Loss	psi	29.5	30.1	30.9	31.6	32.3	33.0	33.6	34.3	35.0
	BMEP (net)	psi	82.5	94.9	109.8	132.4	157.7	174.9	197.8	213.1	235.5
	BHP		150.0	172.5	199.5	240.0	286.0	318.0	359.0	388.0	428.0
	Specific BHP	lb/sec	375.0	361.0	359.0	380.0	402.0	401.0	411.0	408.0	416.0
	Specific IHP (net)	lb/sec	509.0	476.0	460.0	471.0	484.0	477.0	481.0	474.0	478.0
	Air Flow	lb/sec	0.40	0.477	0.556	0.632	0.712	0.793	0.874	0.95	1.03
	Fuel/IHP/hr		0.333	0.356	0.368	0.359	0.35	0.355	0.352	0.357	0.354
	Fuel/BHP/hr		0.451	0.469	0.472	0.446	0.422	0.423	0.412	0.416	0.407
Engine rpm = 1600	Comp. rpm	percent	8.5	12.0	18.0	23.5	29.5	36.5	42.0	47.7	53.3
	Eff. Ratio (γ_e)		0.16	0.225	0.338	0.44	0.553	0.684	0.787	0.894	0.97
	Work with Slip	psi	4.0	22.2	30.5	33.4	31.1	36.8	41.8	46.4	53.5
	Friction and Cooling Loss	psi	18.0	18.6	19.3	19.9	20.5	21.1	21.8	22.3	23.0
	BMEP (net)	psi	95.0	101.2	117.2	140.7	167.4	186.1	208.4	225.3	247.5
	BHP		120.0	128.0	148.0	178.0	212.0	235.0	263.0	285.0	313.0
	Specific BHP	lb/sec	435.0	386.0	383.0	403.0	428.0	426.0	434.0	431.0	436.0
	Specific IHP (net)	lb/sec	518.0	457.0	446.0	460.0	480.0	474.0	479.0	474.0	476.0
	Air Flow	lb/sec	0.276	0.331	0.386	0.441	0.496	0.551	0.606	0.661	0.717
	Fuel/IHP/hr		0.327	0.37	0.379	0.368	0.353	0.354	0.353	0.357	0.355
	Fuel/BHP/hr		0.389	0.438	0.442	0.42	0.396	0.398	0.390	0.393	0.388

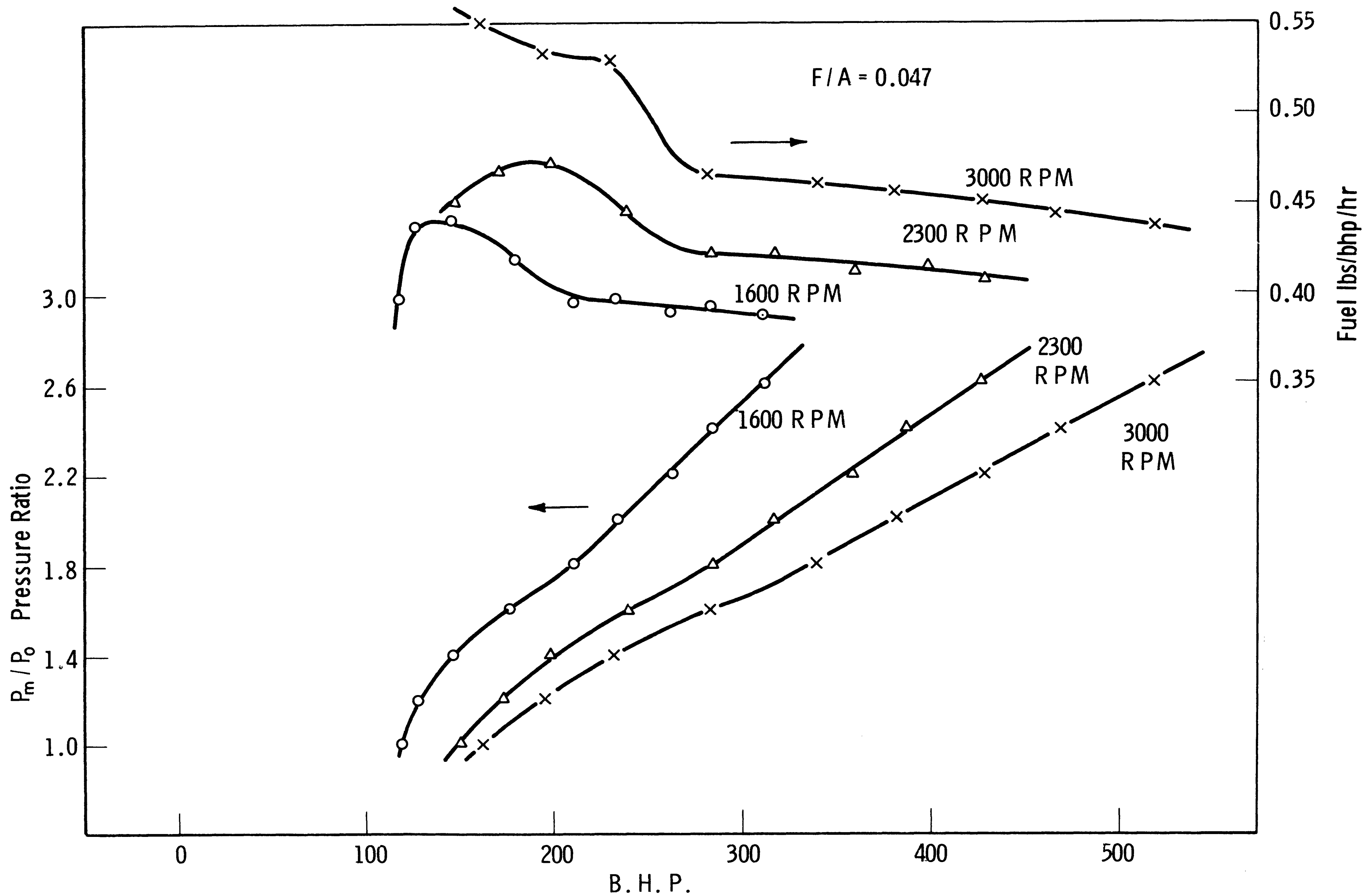

Fig. 4. Performance with slip at full throttle.

TABLE II

PART-THROTTLE PERFORMANCE WITH SLIP

	F/A Ratio	0.047	0.04	0.035	0.03	0.025	0.02
	IMEP (gross)	117.0	112.0	99.0	85.0	68.0	
	Compressor Work	6.0	6.0	6.0	6.0	6.0	
P_m/P_o = 1.0, rpm = 3000	Friction and Cooling	43.0	42.5	42.0	41.5	41.0	
	BMEP (net)	68.0	63.5	51.0	37.5	21.0	
	BHP (net)	161.0	150.4	121.0	88.8	49.7	
	Specific BHP/lb	307.0	287.0	231.0	169.5	94.8	
	Specific IHP/lb	500.0	479.0	421.0	357.0	280.0	
	Air Flow lb/sec	0.524	0.524	0.524	0.524	0.524	
	Fuel/IHP/hr	0.339	0.301	0.299	0.302	0.321	
	Fuel/BHP/hr	0.552	0.502	0.545	0.636	0.948	
	F/A Ratio	0.047	0.04	0.035	0.03	0.025	0.02
	IMEP (gross)	117.0	112.0	99.0	85.0	68.0	42.0
	Compressor Work	5.0	5.0	5.0	5.0	5.0	5.0
P_m/P_o = 1.0, rpm = 2300	Friction and Cooling	29.5	29.0	28.5	28.0	27.5	27.0
	BMEP (net)	82.5	78.0	65.5	52.0	35.5	10.0
	BHP (net)	150.0	141.8	119.0	94.5	64.5	18.2
	Specific BHP/lb	375.0	354.0	297.0	236.0	161.1	45.5
	Specific IHP/lb	509.0	486.0	426.0	363.0	286.0	168.5
	Air Flow lb/sec	0.40	0.4	0.4	0.4	0.4	0.4
	Fuel/IHP/hr	0.333	0.296	0.296	0.298	0.315	0.427
	Fuel/BHP/hr	0.451	0.406	0.425	0.459	0.56	1.58
	F/A Ratio	0.047	0.04	0.035	0.03	0.025	0.02
	IMEP (gross)	117.0	112.0	99.0	85.0	68.0	42.0
	Compressor Work	4.0	4.0	4.0	4.0	4.0	4.0
P_m/P_o = 1.0, rpm = 1600	Friction and Cooling	18.0	17.7	17.3	17.0	16.7	16.5
	BMEP (net)	95.0	90.3	77.7	64.0	47.3	21.5
	BHP (net)	120.0	114.0	98.2	80.8	59.7	27.2
	Specific BHP/lb	435.0	413.0	355.0	293.0	216.0	98.5
	Specific IHP/lb	518.0	489.0	434.0	371.0	293.0	174.0
	Air Flow lb/sec	0.276	0.276	0.276	0.276	0.276	0.276
	Fuel/IHP/hr	0.327	0.295	0.29	0.291	0.307	0.414
	Fuel/BHP/hr	0.389	0.353	0.355	0.368	0.415	0.732

TABLE III

PART-THROTTLE PERFORMANCE, NO SLIP

	F/A Ratio		0.015	0.02	0.025	0.03	0.035	0.04	0.047
	Pressure Ratio		2.6						
	IMEP (gross)	psi	63	133	188	228	258	288	324
	Compressor Work	psi	52.0	52.0	52.0	52.0	52.0	52.0	52.0
Engine rpm = 3000	Friction and Cooling	psi	46.0	47.0	48.0	49.0	50.0	51.0	51.0
	BMEP (net)	psi	0	34	88	127	156	185	221
	BHP			81.0	209	301	370	438	524
	Specific BHP	hp/lb		60.2	156	224	275	326	390
	Specific IHP	hp/lb		143.5	241.0	311	363	416	479
	Air Flow	lb/sec	1.345	1.345	1.345	1.345	1.345	1.345	1.345
	Fuel/IHP/hr			0.502	0.374	0.347	0.347	0.347	0.354
	Fuel/BHP/hr			1.19	0.578	0.481	0.458	0.443	0.436
	Pressure Ratio		2.6						
	IMEP (gross)	psi	63	133	188	228	258	288	324
	Compressor Work	psi	52.0	52.0	52.0	52.0	52.0	52.0	52.0
Engine rpm = 2300	Friction and Cooling	psi	33.0	33.5	34.0	34.5	35	35.5	36.0
	BMEP (net)	psi	0	47.5	102	141.5	171	200.5	236
	BHP			86	185	254	311	365	429
	Specific BHP	hp/lb		83.5	179	249	302	354	416
	Specific IHP	hp/lb		142.5	239	310	364	417	479
	Air Flow	lb/sec	1.03	1.03	1.03	1.03	1.03	1.03	1.03
	Fuel/IHP/hr			0.506	0.577	0.348	0.346	0.346	0.353
	Fuel/BHP/hr			0.864	0.503	0.433	0.417	0.407	0.407
	Pressure Ratio		2.6						
	IMEP (gross)	psi	63	133	188	228	258	288	324
	Compressor Work	psi	52.0	52.0	52.0	52.0	52.0	52.0	52.0
Engine rpm = 1600	Friction and Cooling	psi	20.0	20.5	21.0	21.5	22.0	22.5	23.0
	BMEP (net)	psi	0	60.5	115.0	154.0	184.0	213.5	249.0
	BHP		0	76.5	145.5	195	233	270	315
	Specific BHP	hp/lb		106.8	203	272	325	377	439
	Specific IHP	hp/lb		143	240	310	364	417	480
	Air Flow	lb/sec	0.717	0.717	0.717	0.717	0.717	0.717	0.717
	Fuel/IHP/hr			0.503	0.375	0.348	0.346	0.346	0.353
	Fuel/BHP/hr			0.674	0.444	0.397	0.388	0.382	0.385

charger speed was also 60%, giving an intersecting point at 2.6:1 ratio from Fig. 3. The data of Table III record the results obtained for part-load performance without slip; these data are also plotted on an F/A ratio base in Fig. 5.

In order to simplify the evaluation of fixed and variable drive displacement superchargers, the data of Tables I, II, and III have been plotted on a BHP base in Fig. 6, where the solid lines record the fixed drive conditions and the dotted lines the variable drive conditions. The dotted lines show the data of Table I obtained by slipping the coupling until the pressure ratio reaches 1.0:1, then adding the data of Table II as the throttle is closed (AB represents slip at 3000 rpm, and BC throttle closing at 3000 rpm, similarly for the other speeds).

Figure 6 reveals that the hypothetical 500-HP engine showed a slight loss in fuel economy, amounting to a maximum of 2.3% at 3000 rpm in the range of 350 to 500 HP; but from 350 HP down, there are savings in fuel for a given HP from zero at 350 BHP to 40% at 100 HP. At 2300 rpm there are savings of 42% at 75 HP reducing to zero at 225 HP, while at 1600 rpm there is a 45% reduction in fuel flow at about 60 BHP, reducing to zero at 160 HP. At all speeds there is a loss of 2-3% in the higher HP ranges.

The problem now arises as to the average conditions under which operation occurs. Assuming that the power requirements under the conditions encountered in service do not exceed 50% to 60% of the maximum at any engine speed, the operation of the engine with a slipped compressor would show considerable improvement in fuel consumption relative to a similar compressor driven at a fixed ratio to the engine; and at the same time there is equal full-load HP available for all emergency conditions.

C. RESPONSIVENESS

By accepting the idea of a fluid coupling drive, one can then design into the compressor drive a two-speed arrangement to allow for normal power conditions, employing a low-speed supercharge drive with slip for economy, plus a high-speed drive for high power with responsiveness. Figure 7 is a diagram of one such a device. A compound wheel, D, driven by the engine crank can rotate two halves of the fluid couplings, A and B, at two speeds in the ratio of say 1.0 to 1.5. The other halves of the couplings are located back to back in casing C surrounding the couplings. Facilities will be provided (shown only for coupling A) to supply oil to either coupling as desired. When coupling A is filled with oil, the high-speed gear ratio will be engaged and the supercharger will run at high speed with some slip in coupling B. Because this coupling will be drained of oil, there will be only the small drag caused by the low oil flow required to lubricate the parts. By emptying A and filling B, the low speed is engaged for normal

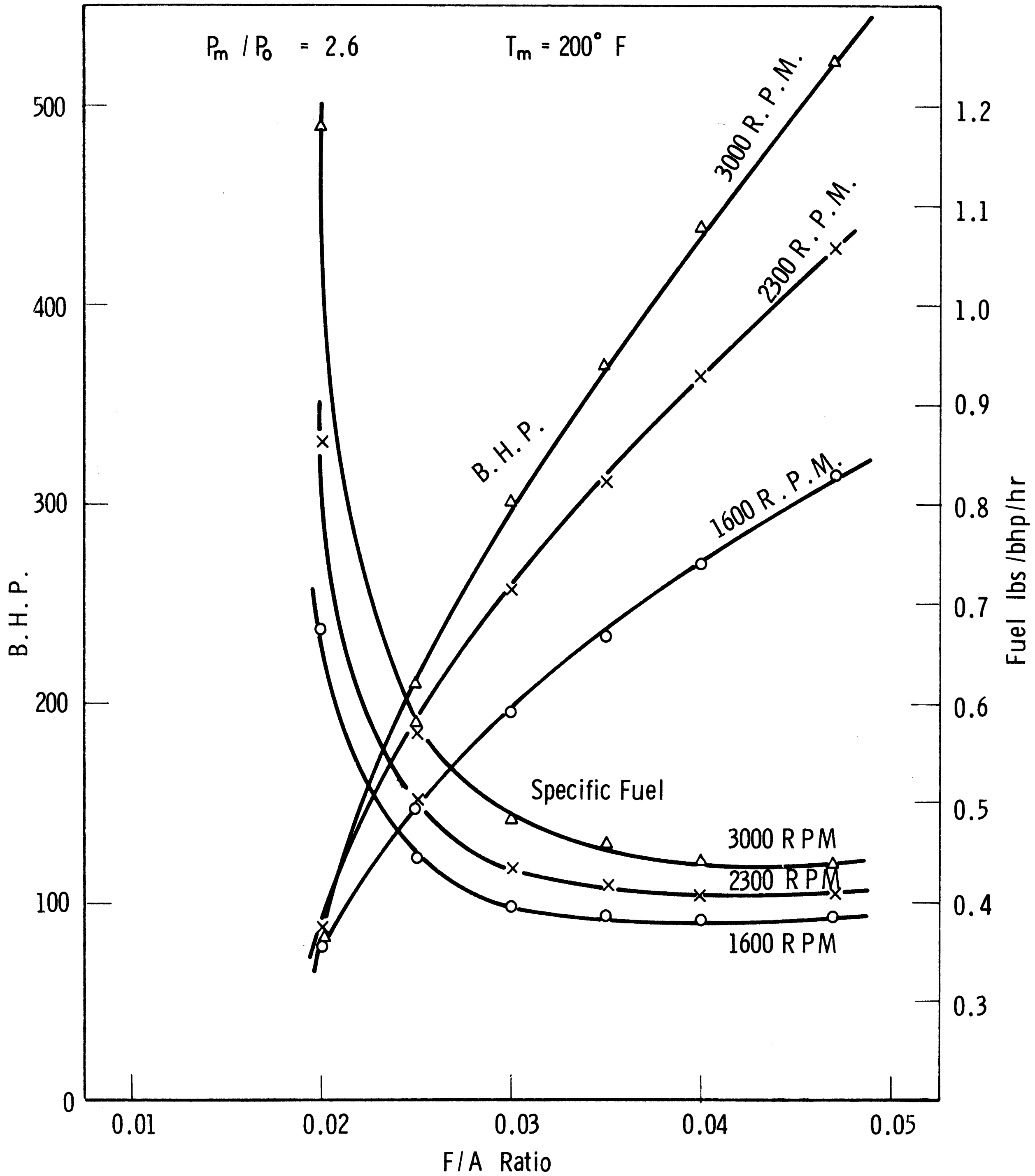

Fig. 5. Performance with fixed drive to supercharger.

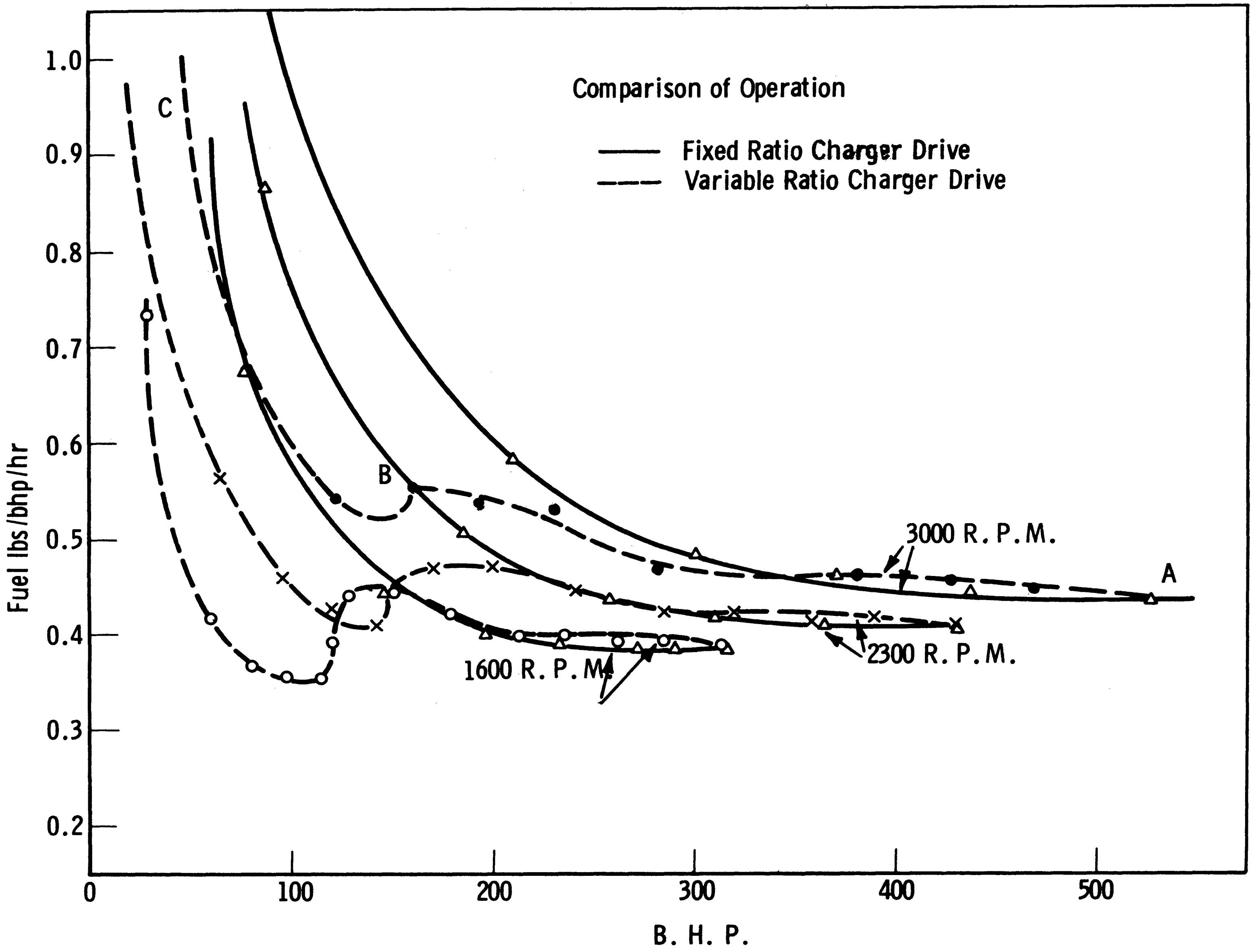

Fig. 6. Comparison of fixed and slipping drive performance.

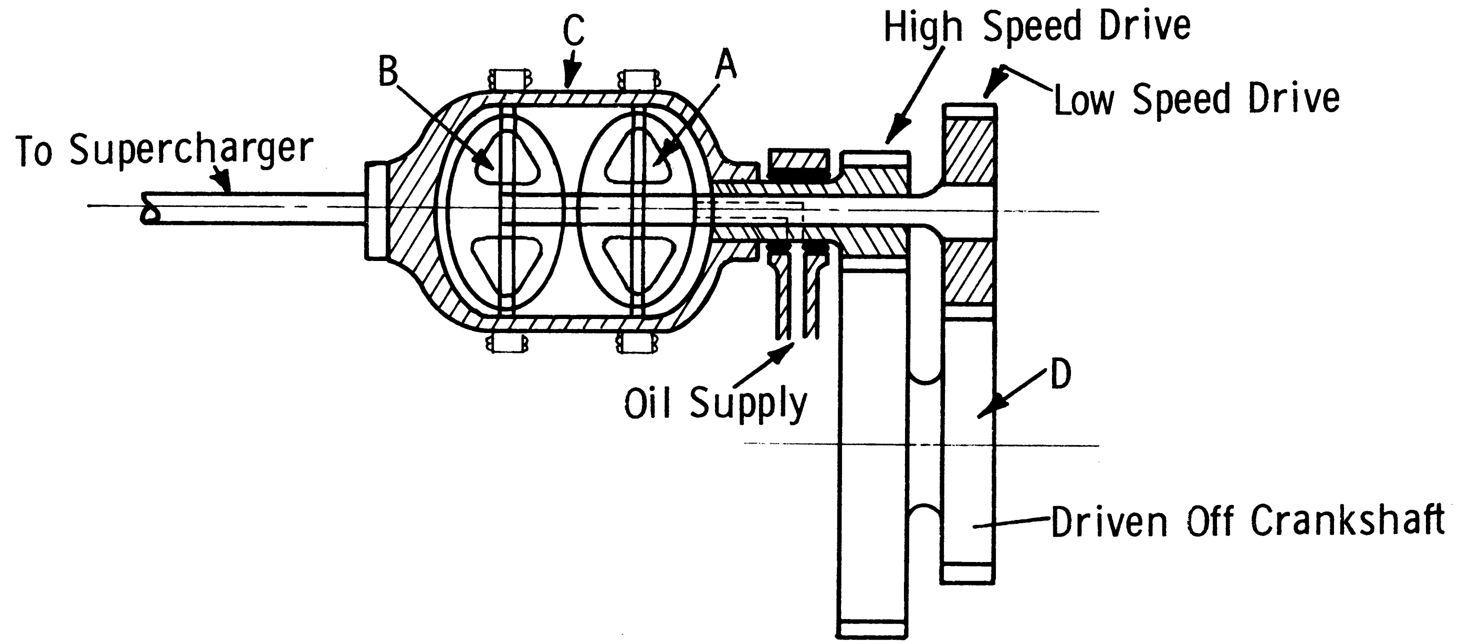

Fig. 7. Supercharger drive for responsiveness.

operation. For the slip condition at low power, the coupling B can be drained to give the desired supercharger speed.

This two-speed drive via a fluid coupling, then, would be superior to a low-speed drive with clutches, since the fuel savings indicated by Fig. 6 can still be achieved by slipping the coupling.

One could employ just one coupling with the high-speed ratio, letting it slip sufficiently to give part-load performance. In this case, the work of compression would be high and the slip great, thereby increasing the fuel consumption to a value higher than that for the two couplings in one casing. Detailed analyses of these two methods would be necessary to make a final selection.

With the preceding rough indication of how to design such a drive, calculations were made to ascertain the engine capabilities with such a two-speed drive. These calculations are shown in Table IV, and the speed and flow conditions for the engine and charger are shown in Fig. 3. The action is as follows:

The low-speed condition at full HP without slip is still represented by the intersection of the 100% engine and supercharger lines, and normal performance with slip at this gear drive is shown in Fig. 6 by the dotted lines.

When the vehicle encounters extreme resistance, the high-speed coupling is filled as necessary to increase supercharger speed, and simultaneously the fuel supply is increased proportionally. The result is that as the engine speed is reduced due to increasing load, the supercharger ratio will increase along the line CD, reaching 3.5:1 at 1600 rpm of the engine, permitting a greatly increased torque development; this is in the case where the two drive gears are in the ratio of 1.5:1. If a 1.88:1 ratio were employed, the line CE would be followed, reaching a supercharger pressure ratio of 4.1:1.

If desired, one could also apply the high-speed ratio at 3000 rpm, however the strain on the engine would be great, and the 1.88:1 speed ratio would still employ only the 100% supercharger speed. Thus, increased loads due to high speed on this unit would not be encountered.

Admittedly, such increase in manifold pressure as would be achieved with the high gear ratio would impose additional gas loads on the engine. Using the V.C.R. principle of Continential Aviation and Engineering Corp. would avoid such increases.

In Table IV, Columns 1 to 7 show the performance to be expected from 3000 to 1600 rpm along the increased pressure-ratio lines CD and CE of Fig. 3, and Columns 8 to 19 record performance along constant-ratio line DF; DH and FG indicate the portion where the fuel/air ratio is gradually reduced

TABLE IV

RESPONSIVE PERFORMANCE WITH TWO-SPEED FLUID DRIVE DISPLACEMENT COMPRESSOR

Pressure Ratio	2.6	2.6	3.0	3.5	3.2	4.1	4.1
IMEP (gross)	324	324	375	425	385	500	500
Compressor Work		52.0	61.6	74.6	67.2	88.0	
Engine rpm	1200	3000	2300	1600	2300	1600	1200
Compressor rpm	1.0	1.0	1.2	1.5	1.31	1.88	1.88
Work with Slip	52.0	53.5	77.0	77.0	77.0	91.0	91.0
Friction and Cooling Losses	16.0	51.0	38.0	26.0	38.0	28.0	20.0
BMEP (net)	256	219.5	260	322	270	381	389
BHP (net)	242.5	520	472	407	490	481	368
Specific BHP	451.0	386	398	419	386	426	434
Specific IHP	478.0	476	456	453	440	457	456
Air Flow	0.538	1.345	1.185	0.97	1.27	1.13	0.848
Fuel/IHP/hr	0.354	0.355	0.371	0.374	0.385	0.37	0.371
Fuel/BHP/hr	0.375	0.438	0.426	0.404	0.439	0.398	0.39
F/A Ratio	0.047	0.047	0.047	0.047	0.047	0.047	0.047
Pressure Ratio	3.5	3.5	3.5	3.5	3.5	3.5	3.5
IMEP (gross)	385	350	330	250	185	425	
Compressor Work	74.6	74.6	74.6	74.6	74.6		
Engine rpm	1600	1600	1600	1600	1600	1200	
Compressor rpm	1.5	1.5	1.5	1.5	1.5	1.5	
Work with Slip	77.0	77.0	77.0	77.0	77.0	77.0	
Friction and Cooling Losses	25.5	25.3	25.0	24.7	24.5	18.5	
BMEP (net)	282.5	247.7	228	148.3	83.5	329.5	
BHP (net)	357	313	288	187	105.5	312	
Specific BHP	368	323	297	193	109	429	
Specific IHP	401	356	329	225	141	453	
Air Flow	0.97	0.97	0.97	0.97	0.97	0.728	
Fuel/IHP/hr	0.359	0.354	0.329	0.40	0.51	0.373	
Fuel/BHP/hr	0.392	0.391	0.365	0.367	0.66	0.394	
F/A Ratio	0.04	0.035	0.030	0.025	0.02	0.047	
Pressure Ratio	3.5	3.5	3.5	3.5	3.5	3.5	
IMEP (gross)	385	350	330	250	185	147	
Compressor Work							
Engine rpm	1200	1200	1200	1200	1200	1200	
Compressor rpm	1.5	1.5	1.5	1.5	1.5	1.5	
Work with Slip	77.0	77.0	77.0	77.0	77.0	77.0	
Friction and Cooling Losses	18.0	17.5	17.0	16.7	16.5	16.0	
BMEP (net)	290	256	236	156.3	91.5	54.0	
BHP (net)	275	243	224	148	86.8	51.2	
Specific BHP	378	334	308	203	119	70.4	
Specific IHP	401	357	331	225	140	91.2	
Air Flow	0.728	0.728	0.728	0.728	0.728	0.728	
Fuel/IHP/hr	0.359	0.353	0.326	0.40	0.512	0.69	
Fuel/BHP/hr	0.381	0.378	0.35	0.444	0.603	0.894	
F/A Ratio	0.04	0.035	0.03	0.025	0.02	0.0175	

from the full-load value of 0.047 to 0.0175, at which point the HP has been reduced to about 50 at both 1600 and 1200 rpm—the part-throttle performance.

Figure 8 plots this responsive operation at full throttle, where HP, torque, and fuel consumption are shown for the 0.047 F/A ratio relative to the condition with zero-percent slip investigated in Table I. The original data of Table I indicate a gradually increasing torque from 1.0 to 1.18; with $\gamma_e = 1.5$, the ratio changes from 1.0 to 1.48; and with $\gamma_e = 1.88$, it changes from 1.0 to 1.78—a 78% increase in torque at 1200 rpm relative to that at 3000 rpm. This is not a high degree of responsiveness, but an examination of the outputs shows that the HP change at 1200 rpm is from 240 to 380, a fairly substantial increase at such a low speed.

The fuel consumption data show a small increase in specific fuel consumption as γ_e changes from 1.0 to 1.88. At 3000 rpm, all three conditions are the same because the blower speed also is constant; the relative speed ratio γ_e does not reach 1.5 or 1.88 until the engine speed has been reduced to 1600 rpm. The maximum increase in specific fuel is 8.5% approximately and occurs at about 2300 rpm. When maximum responsiveness occurs, at 1600 rpm, the increase in specific fuel flow is only about 6% for a torque increase of 37% to 51%.

Figure 9 shows the part-throttle performance for these two responsive conditions at 1600 and 1200 rpm; DH represents operation at 1600 rpm, and FG operation at 1200 rpm (from Fig. 3). The dotted fuel-consumption line in Fig. 9 represents performance at full throttle and 1600 and 1200 rpm, when $\gamma_e = 1.88$, giving 4.1:1 ratio, indicating the shift toward higher HP of the two part-throttle lines with little or no change in SFC. The degree of responsiveness is shown by the fact that 520 HP is developed at 3000 rpm and 480 HP at 1600 RPM if the 1.88:1 value of γ_e is achieved.

The problems of developing a supercharger, of the type being considered, to produce a pressure ratio of 4.1:1 would be great, though indications are that it would not be impossible. Its characteristics, as represented in Fig. 3, show that large speed increases are not necessary for a given size blower to achieve high pressure ratio, in contrast to the centrifugal machine. As already stated, the 4.1:1 ratio can be attained if, by means of the coupling, blower speed is kept at 100% while the engine speed is reduced to 53.3% of full speed, as shown by point E of Fig. 3.

To demonstrate the relative performances, we have added to Fig. 9 one line representing a 3000-rpm performance with fixed drive ratio, employing the charger under discussion, and another line representing a fairly typical turbocharger performance. The former shows high specific fuel consumption at low loads due to the combination of high frictional losses plus constant compressor work, while the turbocharger suffers from low power output at low speed—approximately 90 BHP maximum at 1200 rpm for the centrifugal against

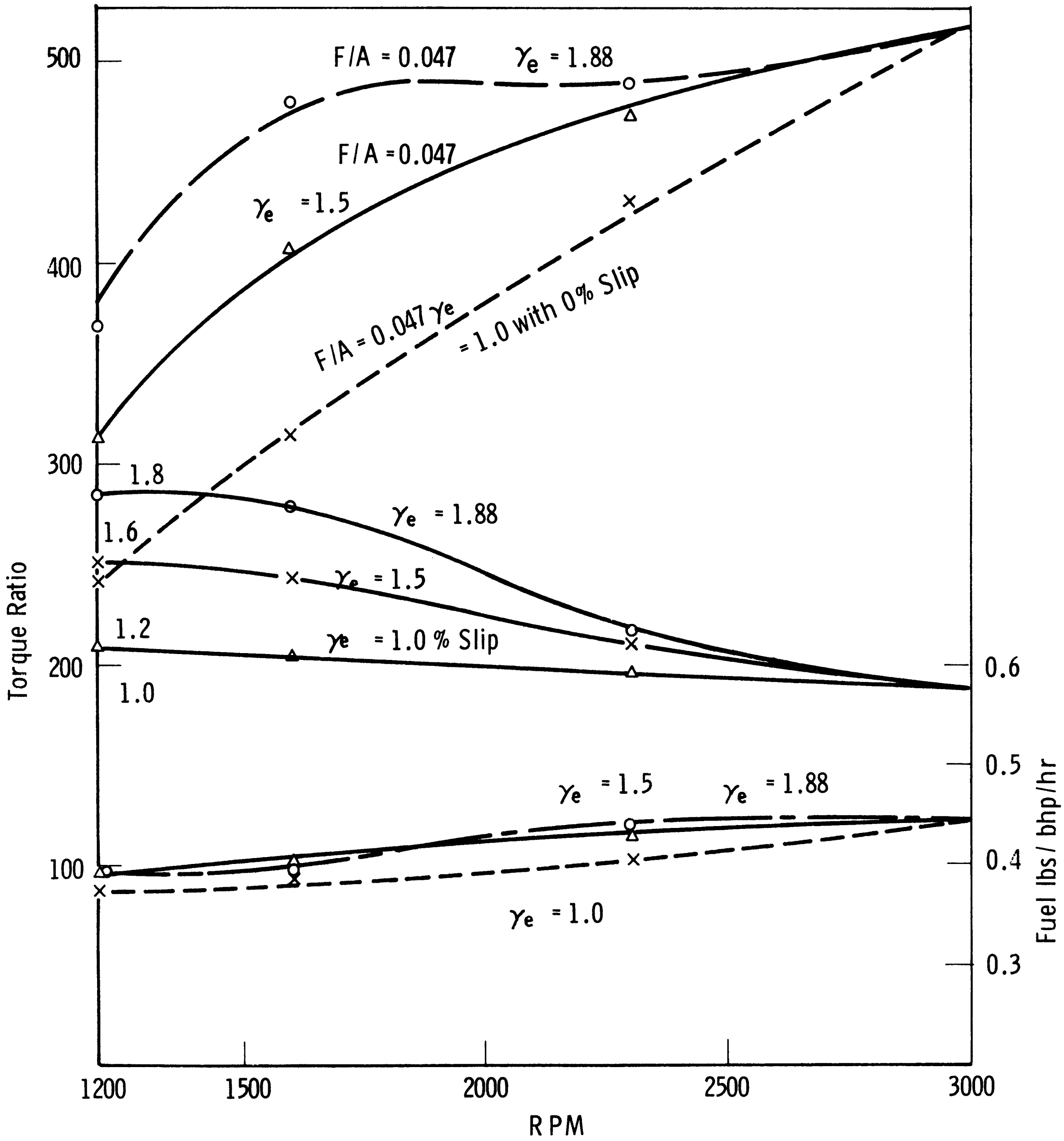

Fig. 8. Response operation at full throttle.

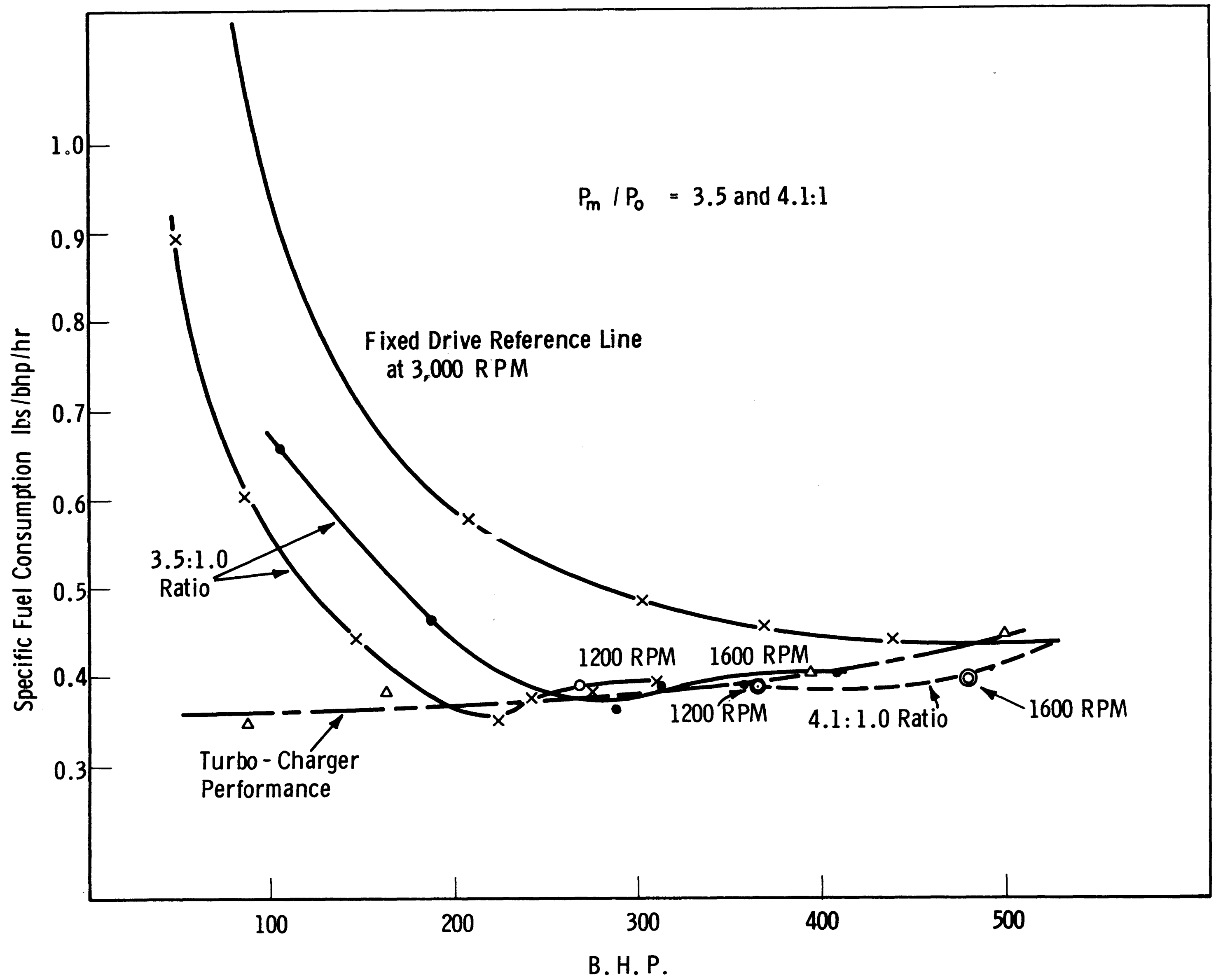

Fig. 9. Part-load responsive performance.

210 HP at the same engine speed for $\gamma_e = 1.0$, 240 with $\gamma_e = 1.5$, and 285 HP when $\gamma_e = 1.88$. The vehicle's performance in regard to acceleration, power to overcome obstacles, etc., would be greatly improved with the displacement blower at the expense of an increased fuel consumption in the normal power range up to about 200 to 250 BHP.

D. CONCLUSIONS

In normal operation on relatively level surfaces, a compression-ignition type engine fitted with a displacement supercharger with compression driven off the engine by means of a variable fill fluid coupling to permit speed-ratio variation possesses distinct advantages, as far as fuel flow is concerned, compared to the fixed drive ratio machine. The fuel flow up to about 250 HP, of a total of 500 HP, is superior by a relatively large margin, while in the 250- to 500-HP load range the loss is only a matter of some 2% relative to the fixed-ratio machine.

It follows that operating conditions would play a great part in determining the type of supercharger to be used in a vehicle. Low-speed operation in difficult terrain, where high torque is necessary at low vehicle speed, favors the slip-speed blower, because this unit can be operated at the lowest speed at which it can develop the required HP, but can be driven at overspeed to give a good torque increase if conditions warrant; this overspeed can be obtained with a fairly simple two-speed device.

When this type of supercharger is compared with the turbocharger, the two main differences are specific fuel consumption and power output or torque. The turbocharger engine has low torque at low speed, and thus low HP but good fuel economy, accompanied by some lack of acceleration and smoke when opening up from idle condition. At full power and speed, its power and economy are as good or better than the other designs, and at present it is capable of operating at higher manifold pressures than all other superchargers. The positive displacement machine will give superior low-speed torque and HP characteristics at the expense of considerably increased fuel flow, which can be reduced considerably if the slipping drive is applied. At the same time some degree of responsiveness is achieved.

II. COMBINATIONS OF THE MECHANICALLY DRIVEN SUPERCHARGER WITH TURBOCHARGERS

A. DISPLACEMENT AND TURBOCHARGERS COMBINED

This section will deal with the effect of combining a mechanically driven, positive displacement charger with a turbocharger. This combination is used because it can function turbocharged with maximum economy for normal operating conditions, and then for emergency conditions bring in the mechanically driven charger via a fluid coupling, or similar device, with only slightly increased fuel consumption.

Calculations were made by combining the methods of Ref. 1 for turbo- and mechanically driven chargers. We assumed that the positive displacement machine was designed for various ratios—1.2, 1.4, to a maximum of 2.0:1—and that it was then combined with a turbo of ratios 1.0 to 3.0 to give mean effective pressures up to approximately 400 psi; the overall fuel consumption was determined as before. The data of Table V were obtained from these calculations; the data of Table VI are for a typical turbocharged engine alone.

Figure 10 plots the data of Tables V and VI on a base of BMEP for a constant engine speed of 3000 rpm; thus the base also represents HP to some scale.

The case of the combined displacement and turbochargers shows that for a given BMEP the specific fuel increases as the ratio of the displacement compressor increases. This would be expected from the relative efficiencies of the two units. Secondly, the maximum BMEP achieved increases with the overall manifold pressure, but the straight turbocharged engine develops the higher mean pressure for any given value of the overall manifold pressure ratio; this is accompanied by the lowest specific fuel consumption over the whole load range.

B. CONCLUSIONS

From the standpoint of either power output or fuel consumption, there seems little merit in combining the characteristics of the displacement compressor with those of the centrifugal compressor if the maximum manifold pressure is to be held at some constant value, i.e., at full load and high-speed operation.

TABLE V

PERFORMANCE AT 3000 RPM

Displacement Pressure Ratio, R_m	Overall Pressure Ratio	IMEP (net), psi	BMEP, psi	Fuel, lb/IHP/hr	Fuel, lb/BHP/hr
1.2:1	1.2	135.6	91.6	0.345	0.510
	1.68	194.2	147.2	0.329	0.432
	2.16	247.9	199.4	0.340	0.422
	2.64	305.5	254.5	0.338	0.405
	3.13	365.1	312.1	0.335	0.392
	3.6	425.8	370.8	0.330	0.378
1.4:1	1.4	163.6	118.7	0.334	0.461
	1.96	228.3	180.7	0.335	0.424
	2.52	289.9	239.9	0.340	0.410
	3.08	356.5	304.0	0.330	0.387
	3.64	417.0	362.0	0.341	0.393
	4.2	477.6	419.3	0.343	0.391
1.6:1	1.6	181.0	135.0	0.345	0.462
	2.24	253.5	204.5	0.346	0.428
	2.88	328.9	276.0	0.342	0.407
	3.20	363.7	310.7	0.344	0.403
	3.52	398.5	343.5	0.346	0.400
	3.84	438.3	382.3	0.350	0.402
	4.16	475.0	417.5	0.346	0.390
1.8:1	1.8	200.9	153.9	0.349	0.457
	2.52	282.3	228.0	0.348	0.431
	3.24	366.8	308.0	0.344	0.410
	3.6	406.6	346.2	0.346	0.406
	3.96	450.3	384.5	0.342	0.402
	4.32	492.1	425.3	0.343	0.396
	4.65	527.8	466.0	0.344	0.388
2.0:1	2.0	218.8	170.8	0.356	0.456
	2.8	308.3	256.7	0.355	0.426
	3.6	406.6	351.6	0.346	0.400
	4.0	449.3	392.3	0.347	0.397
	4.4	495.1	436.1	0.347	0.394
	4.8	548.9	486.9	0.342	0.386
	5.2	588.7	524.7	0.344	0.386

TABLE VI

TURBOCHARGED ENGINE

Manifold Pressure Ratio	IMEP (net)	BMEP	Fuel, lb/IHP/hr	Fuel, lb/BHP/hr
1.6	189	143	0.326	0.432
2.0	234	186	0.330	0.414
2.4	279	229	0.334	0.405
2.8	338	287	0.321	0.377
3.2	373	320	0.330	0.386
3.6	424	369	0.328	0.376
4.0	474	417	0.327	0.370
4.4	522	464	0.326	0.366
4.8	568	508	0.326	0.362

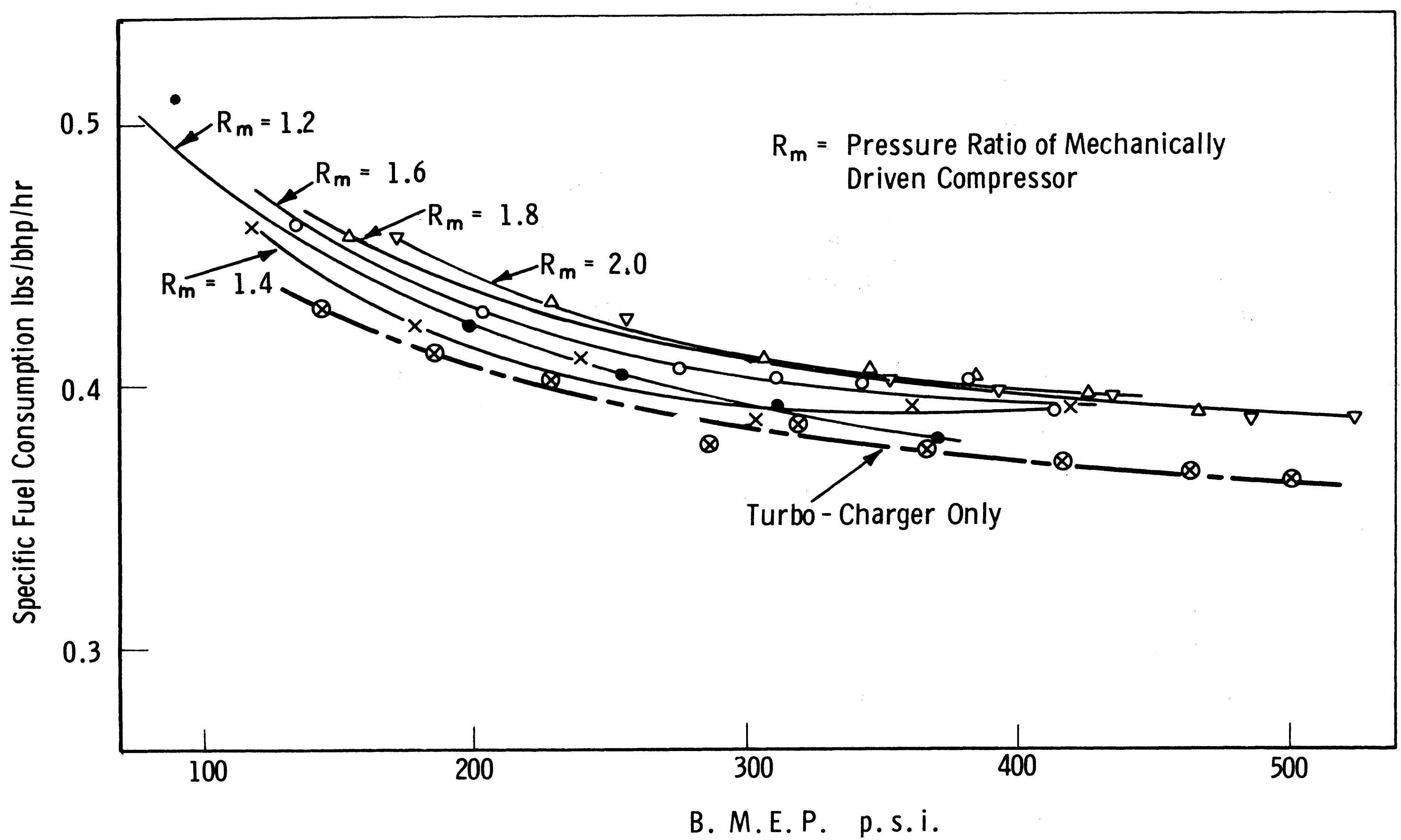

Fig. 10. Full-throttle performance at 3000 rpm of combined chargers relative to a turbocharger alone.

As shown in Fig. 9, at reduced speed and load, substantial fuel would be saved by using a turbo alone, providing the power capability of the engine so fitted is sufficient for the purpose, or gear ratios can provide torque requirements.

Due to the action of the positive displacement unit, the combined type of chargers would be of value only for increased torque capability at low speed. To capitalize on this, a drive should be provided to disconnect the displacement unit when not required, otherwise a heavy fuel flow penalty will result at part-throttle operation.

The type of drive required—fluid coupling, clutches, etc.—would add complications over and above the need for two superchargers. The result would be increased engine bulk and complexity, with compensating advantages for short periods of operation only.

The simpler, and thus generally more reliable, solution then would be to select (1), a high-ratio displacement compressor which produced high isentropic efficiencies, driven via a fluid coupling to give both high torque multiplication at low speed, coupled with slip for economy at part load, or (2), a turbocharger with variable turbine nozzles which would improve both low-speed torque and fuel economy over the whole range of operation. Admittedly, (2) would not provide as much torque multiplication at low speed as (1).

C. OBSERVATIONS

The positive displacement machine with variable drive via a fluid coupling is particularly interesting in view of the recently published article, Differentially Compounded, Supercharged Diesel Engines (Ref. 3). The next report will analyze the differential drive machine and compare it with the type discussed in Section I of this report. The two are quite similar as far as the objectives are concerned, and a detailed analysis of the part-load capabilities (not given in Ref. 3) as well as the full-load capabilities of the two systems must be made in order to evaluate the overall operation characteristics.

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