THE UNIVERSITY OF MICHIGAN
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Technical Report

THE POSSIBILITIES OF A RESPONSIVE TURBO-CHARGED COMPRESSION IGNITION ENGINE

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ABSTRACT

This report is an assessment of the possibilities of producing a responsive or partly responsive compression ignition engine operating at the highest mean pressures developed by recent single-cylinder research engines. A compressor map is presented which shows the results potentially obtainable by applying recent technical advances to produce a unit with high pressure ratio and high efficiency.

From these data are constructed the requirements that could now be met with regard to the achievable range of responsiveness, the type of controls necessary, the characteristics of the supercharger, and the exploitation of its flow characteristics.

The overall performance map will be evaluated in detail in a later report.

OBJECTIVE

This report examines the characteristics of tubro-chargers and the conditions under which they operate, with a view to determining whether any form of responsive operation can be developed through their use.

This report can be considered an extension of Ref. 4, an investigation of the minimum responsiveness that would match existing data on the performance of an engine having a transmission with a torque converter and a lock-up device. The present report looks at the maximum responsiveness that appears feasible at present, the necessary controls, etc., for satisfactory operation of a turbo-charged engine.

PRINCIPLES AND CHARACTERISTICS OF TURBO-CHARGERS

The turbo-charger consists of a centrifugal compressor driven by a gas turbine which is operated by the exhaust gases from the internal combustion engine. As a result, the performance characteristics of the engine depend upon those of both the centrifugal compressor and the gas turbine. The limitations of the former are perhaps the more serious; these limitations and their effects are discussed below.

THE CENTRIFUGAL COMPRESSOR

The unit consists of a high-speed impeller enclosed in a stationary housing fitted with a volute exit passage and sometimes with diffuser vanes. In some cases a concentric casing only is used.

The speed of rotation for a given pressure can vary widely, according to the diameter of the impeller. It is the blade tip velocity that matters, not speed of rotation; however, a slow speed makes large sizes necessary. For most engines, therefore, small size is of prime importance if the design is to be integrated into the general engine design. Thus for the present purpose, high speed of rotation—30,000 to 100,000 rpm—can be taken as conventional. The impeller tip speed, which determines the pressure ratio produced, will be of the order of 1,200 to 1,600 fps, depending upon the pressure ratio desired, the efficiency of the wheel, etc.

The air flow capability of the machine depends upon the passage area and its aerodynamic characteristics. In general, the flow capacity is large relative to the machine's overall dimensions, and the compressor is small. However, for a given diameter and speed there are certain limits of air flow possible within the framework of the size; therefore compressors of various sizes are available for differing air flow requirements.

It can be shown that the maximum work capacity of such a machine is given by:

$$\text{maximum work capacity} = \frac{w}{g}V^2 \text{ (ft lb/sec)}$$

w = weight flow (lb/sec)

V = tip velocity of impeller (fps)

When losses, etc., are included, the equations relating flow, speed, temperature rise, etc., are

$$\text{work} = JwC_p\ (T_2 - T_1)\ (\text{ft-lb/sec})$$

J = mechanical equivalent

C_p = specific heat at constant pressure (Btu's/lb)

T_1 = inlet temperature (° abs)

T_2 = outlet temperature (° abs)

An ideal machine is considered one in which compression is isentropic, in which case

$$\text{ideal work} = JwC_p\ (T_{is} - T_1)$$

$$T_{is} = \text{isentropic temperature of compression}$$

$$= T_1(R)^{\frac{k-1}{k}}$$

$$R = \text{pressure ratio}$$

$$= \frac{\text{delivery pressure}}{\text{inlet pressure}}$$

$$= \frac{P_2}{P_1}$$

$$\text{ideal work} = JwC_p\ T_1 \left(R^{\frac{k-1}{k}} - 1\right)$$

It is now possible to define the efficiency of a compressor by

$$\text{isentropic efficiency } \eta_c = \frac{\text{isentropic work}}{\text{work of compressor}}$$

$$= \frac{JwC_p(T_{is} - T_1)}{JwC_p(T_2 - T_1)}$$

$$= \frac{T_{1s} - T_1}{T_2 - T_1}$$

Then

$$\text{required work of compression} = \frac{JwCpT_1}{\eta_c} \left\{ (R)^{\frac{k-1}{k}} - 1 \right\} \text{ ft lb/sec}$$

If a work coefficient be employed, the ratio of the work of compression to the maximum work is

$$\eta_w = \frac{\text{work of compression}}{\text{maximum work}}$$

$$= \frac{JwC_pT_1 \left[(R)^{\frac{k-1}{k}} - 1 \right] g}{wV^2}$$

$$= \frac{JgC_pT_1 \left[(R)^{\frac{k-1}{k}} - 1 \right]}{V^2}$$

and a relation involving the impeller speed is obtained in terms of temperature rise, etc. It is seen that for any given inlet conditions the pressure ratio is a function of the square of the tip velocity.

The above equations are sufficient for most overall compressor calculations.

SURGE

Another characteristic that must be examined is an instability in flow which occurs as the flow rate is reduced. This effect arises from the pressure gradients produced throughout the machine, which, in normal operation, are sustained by the momentum of the fluid. A boundary layer exists in which the momentum is reduced far more quickly than in the main stream. If this layer thickens sufficiently, the resulting pressure difference may cause the main stream to reverse its direction of flow in the layer. Then, of course, the pressure will disappear and the reversed flow also is eliminated. Pumping starts again, but the boundary layer again builds up and the reversed flow again sets in, producing the phenomenon of "surge," a to and fro air movement at high frequency accompanied by a serious loss of both flow and pressure.

It will be observed that surge results when a reduction of flow leaves space in the diffuser channels for the reversed flow to occur. At the opposite end of the scale, when maximum flow is desired, a point is reached where the flow becomes "choked;" that is, sonic velocity exists at some local section, usually in the diffuser passage. It follows that there is a range of usable flow for each speed of rotation from surge to choke.

Now as the pressure ratio of compression increases, the range of flow between surge and choke gets smaller and the design becomes more critical, with the possibility of little if any change of flow for stable operation. This characteristic can be seen in Figure 1.

In general one can state that where a wide flow range is desired, low pressure ratio is necessary. When both high pressure ratio and wide flow range are necessary, two or more stages of low pressure ratio in series must be used in order to secure the flow range desired. An alternative would be to use a single high-pressure stage designed for the maximum flow desired, with a by-pass arrangement to bleed off the necessary mass flow when the delivered capacity enters the stall region, thus maintaining stable flow from the compressor at all times.

THE RADIAL FLOW TURBINE

In most turbo-charger applications the radial flow or centripetal turbine is used because of its ease of construction, the simplicity of its operating elements and piping arrangements, and its low cost. This turbine can be considered a reversed-flow centrifugal compressor, but fortunately it is not subject to the same limitations as that unit. In the turbine, the combustion gas flow is from a place of high pressure at the entry to the periphery of the wheel, and leaving at the axis of rotation where the pressure is atmospheric.

Hence, the pressure drop is in the direction of flow, so that any boundary layer formations tend to be accelerated in the direction of motion rather than in the opposite direction as with the compressor.

Second, since the gas is at a high temperature, the sonic velocity is high, and therefore there is no choking for the expansion process involved. Because of the increased temperature, the volume of gas has increased and suitable size adjustments must be made for the flow. For example, the turbine nozzle exit area must be relatively large compared with the diffuser inlet of the compressor.

In operation, the mass flow in the turbine is the same as in the compressor, plus a small addition of fuel from the engine. The flow passage of the turbine must efficiently handle the maximum engine flow at full load and speed; at half engine speed, however, approximately half the flow occurs, and so on as speed changes, since the engine displacement is acting as a metering device. As a result, the pressure ahead of the turbine falls rapidly as engine speed decreases and inlet manifold pressure is rapidly lost at low speeds.

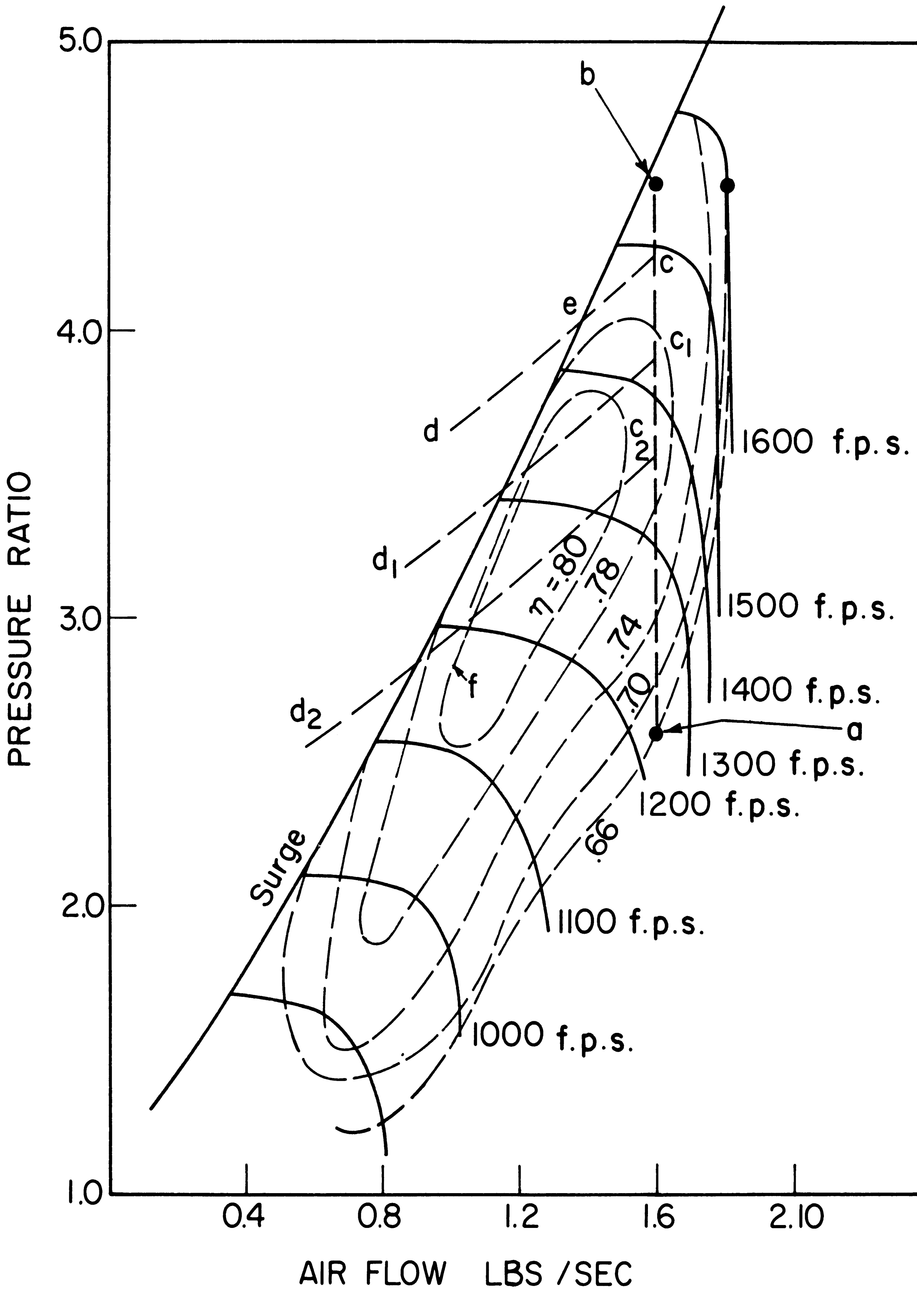

Figure 1. Typical compressor map.

As is generally admitted, the pressure ratio should be maintained at low engine speed in order for the engine to meet army requirements for satisfactory torque, smokeless exhaust, etc. In the turbine this is readily accomplished by using movable nozzle blades, with which the angle of flow of gas can be adjusted to the wheel speed, and by reducing the nozzle area as flow decreases and thus preventing reduction of the inlet pressure in the turbine with flow; this increases available energy, and then produces a higher manifold pressure at low engine speed.

An inferior method of achieving this same result is to use a waste gate. In this design, the turbine nozzles are designed for part load and speed operation and are too small for full load and speed. This means that, if no corrective steps are taken excessive back pressure will be applied to the engine under full speed conditions which would result in an overspeeding of the turbine. To control this, some exhaust gas is by-passed around the turbine directly to the atmosphere when the exhaust conditions exceed the desired maximum values. It follows of course that part load and speed conditions are improved at some sacrifice of full load performance.

EFFICIENCIES

Also important is the efficiency attained by both compressor and turbine. The efficiency of the compressor should be as high as possible at all times, since it determines the temperature rise of the air for any given pressure ratio. The lower the delivery temperature, the greater the power output of the engine and the lower the heat losses.

Since the turbo must not develop more power than the compressor can absorb, the best turbine design for the temperature and pressure available is not necessarily the correct one for a given engine. In all practical cases (certainly with four-cycle engines) the exhaust gas contains more available energy than is required to drive the compressor. The result is that the turbine efficiency, flow path, etc., need not be at maximum values; one can make sacrifices to reduce cost of manufacture, increase ease of maintenance, etc., and still obtain enough energy for the work of compression. If the overall efficiency of compressor and turbine amounts to about 50-60%, a satisfactory turbo-charger unit can be achieved for most conventional engines.

Where abnormal supercharge or other conditions warrant, the development of highly efficient units for modern requirements can undoubtedly be justified; but as manifold pressure requirements go higher and higher the main emphasis must be placed upon compressor efficiency and range of operation between surge and choke. The turbine can readily be designed to meet the compressor requirements for some time to come.

THE RESPONSIVE TURBO-CHARGED COMPRESSION IGNITION ENGINE

Now let us look at the application of the responsive turbo-charger to military needs. The first subject of investigation will be the measures to be taken to obtain some responsive performance, if possible, from an engine using the turbo-charger as at present conceived.

It has been shown that if the engine is to be responsive, its BMEP must increase inversely in relation to the speed ratio change; then

$$\frac{BMEP_1}{BMEP_2} = \frac{rpm_2}{rpm_1}$$

Reference 3 has indicated that for present engines a responsiveness of about 1.7:1 is necessary if present performance with torque converters is to be achieved. Let it be assumed that a 2:1 responsiveness could be achieved today, i.e., that constant horsepower can be developed over a 2:1 speed range. In addition, assume that full power is obtained with a BMEP of 250 psi; then

$$\text{BMEP at half speed} = 250 \times \frac{\text{rpm max}}{\text{rpm min}}$$

$$= 500 \text{ BMEP}$$

Now it can be argued that since this 500 psi mean pressure is about the maximum possible value at present, this should be the mean pressure at full speed. If this were considered necessary, then it is believed any additional responsiveness would be almost impossible at present. It will be assumed that 250 to 500 psi is acceptable in a present-day engine with responsive action; this assumption will make it possible to investigate the turbine conditions and examine their possibilities.

In order to achieve a BMEP of 250 a supercharger ratio of 2.5:1 is required; this represents a manifold pressure of about 75" Hg abs. To reach the desired 500 MEP, this ratio must be increased to approximately 5:1 or 150" Hg. These values are stated on the assumption that aftercoolers are used in each case. The above ratios can be secured from centrifugal compressors, provided that the tip speeds below are employed.

Pressure Ratio 2.5:1

$$\eta_w = \frac{JgC_pT_1 \left(R^{\frac{k-1}{k}} - 1\right)}{V^2}$$

Assume $\eta_w = 0.73$

$$V^2 = \frac{778 \times 32.2 \times 0.241 \times 580 \left(2.5^{0.286} - 1\right)}{0.73}$$

$$= 1{,}434{,}000$$

$$V = 1{,}200 \text{ fps approximately}$$

Pressure Ratio 5:1

$$V^2 = \frac{778 \times 32.2 \times 0.241 \times 580 \left(5^{0.286} - 1\right)}{0.73}$$

$$= 2{,}804{,}000$$

$$V = 1{,}700 \text{ fps approximately}$$

The 1,200 fps for the 2.5:1 is easily attainable; most present chargers have perhaps a somewhat higher velocity for the same ratio since the excess energy available in the exhaust gases has rendered high efficiency unnecessary. For 5:1 ratio, however, such a compressor, designed in a single stage, would have an extremely small flow range without surge, so small that operating it with an engine would be almost impossible. Second, because of the high tip velocity, supersonic velocities in diffusers would be more than possible, and the bursting stresses in the impeller would be hard to contain; hence the 5:1 ratio would be difficult to produce in a single-stage machine. It follows that a two-stage machine would probably be contemplated for this case; therefore if the range of ratios proposed is required, a two-stage machine must be contemplated for the 2.5:1 ratio also. Though this two-stage design is feasible, it is to be avoided if at all possible. If manifold pressure ratios are increased in the future so that the power plants can provide engine mean pressures of more than 500 psi, then a two-stage machine giving a range of ratios from 4:1 to 7:0 or 8.0:1 ratio for responsiveness would be required.

Let it be assumed for the present, however, that a single-stage machine of 2.5 to 5.0:1 could be designed to operate at the desired velocities; that 2.5:1 is achieved with the engine at full speed, say 2,800 rpm; and that the ratio gradually increases to 5.0:1 as the engine speed drops to 1,400 rpm.

Since the machine is to be a turbo-charger, it is necessary to evaluate the capabilities of the exhaust gases under each condition of operation, as follows.

BALANCE OF POWER BETWEEN COMPRESSOR AND TURBINE

The work of compression for each of the ratios can be calculated as follows:

$$\text{isentropic efficiency} = 75\% = \eta_c$$

Pressure Ratio 2.5:1

$$\text{work} = \frac{wCpT_1\left(R^{\frac{k-1}{k}} - 1\right)}{\eta_c} \text{ Btu/w lb}$$

$$= \frac{1.0 \times 0.241 \times 580 \left(2.5^{0.286} - 1\right)}{0.75}$$

$$= 55.8 \text{ Btu/lb of air}$$

$$= 79.0 \text{ H.P./lb of air/sec}$$

Pressure Ratio 5:1

$$\text{work} = \frac{1.0 \times 0.241 \times 580 \left(5.0^{0.286} - 1\right)}{0.75}$$

$$= 109.0 \text{ Btu/lb of air}$$

$$= 154.2 \text{ H.P./lb of air/sec}$$

In the above calculations the compressor efficiency has been assumed constant at 0.75 for each ratio. This would not be exactly true in practice; a difference of some 6 to 8% could be expected. However, ignoring this change in efficiency will probably not affect the accuracy of the general principles under examination.

For constant horsepower an approximately constant air flow is to be expected. Examination of Figure 1 shows that for a constant air flow of 1.6 lb/sec a 4.5:1 ratio is achieved by the compressor at an efficiency of about 76%, and a 2.5:1 is secured at 66%. It must be remembered that the compressor represented by this diagram was not designed for the purpose involved in this investigation, but in order to reach these values a maximum efficiency of 80% or better had to be achieved.

The work of the turbine will be estimated by assuming that the engine back pressure is 0.87 of the inlet manifold pressure (this provides for adequate scavenging of the exhaust products), and that the F/A ratio at maximum load at the two speeds will be the same at 0.043 pounds of fuel per pound of air. In these conditions, the exhaust gas temperature can be expected to be about 1,250° F at 2800 rpm and possibly as high as 1,500° F at 1,400 rpm. Admittedly, this is an educated guess, since no engines of the type being considered are operating at pressure ratios of 5:1 so far as is known. The conditions to be investigated are as follows:

Pressure Ratio 2.5:1

expansion pressure ratio = 0.87 x 2.5 = 2.175

gas temperature = 1,710° abs

From figure 8 of Ref. 2,

maximum work/lb of air = 87.4 Btu/lb of air

for F/A = 0.043

= 123.6 H.P./lb of air/sec

Pressure Ratio 5:1

expansion pressure ratio = 0.87 x 5.0 = 4.35:1

gas temperature = 1,960° abs

From figure 8 of Ref. 2,

$$\text{maximum work/lb of air} = 173.8 \text{ Btu/lb of air}$$

$$\text{for } F/A = 0.043$$

$$= 246 \text{ H.P./lb of air/sec}$$

It follows that the turbine would have to be at least 64% efficient for the 2.5:1 engine and 62.3% for the 5.0:1. Since such efficiencies are readily obtainable, the efficiency of the compressor need not be as high as 75%; however, the lower the efficiency the greater must be the capacity and size of the aftercooler, and these factors could greatly affect the size of the overall package.

It can be concluded that for the conditions investigated, there would be no problem in driving the compressor from the exhaust gases for the conditions investigated, and that unrealistically high efficiencies need not be achieved.

Flow Requirements

One further limitation is the quantity of gas flow under the two conditions and its effect upon such problems as surge. Since the engine displacement, of course, is constant per rev. at all speeds, the air flow will be proportional to the engine speed, if we neglect such factors as variable volumetric efficiency and volume of scavenge air during valve overlap; this assumes a constant air supply temperature after the cooler. Hence mass flow will vary directly with the pressure ratio if the inlet temperature is constant. An aftercooler could not be expected to maintain an exactly equal air temperature over such wide ranges as those contemplated, if designed for the most severe condition; however, it would tend to a constant outlet temperature (inlet temperature to engine) from the cooler.

Air flow at constant speed = displacement x rpm x density

$$= \frac{\pi d^2}{4} \times \ell \times n \times \frac{\text{rpm}}{1{,}728} \times \frac{P_m}{RT_m} \text{ lb/min}$$

$$= \text{constant} \times P_m \times \text{rpm lb/min}$$

d = diameter of cylinder ins

ℓ = stroke

n = number of cylinders

$T_m = 200° \text{ F}$

Air flow at 2,800 rpm and 2.5:1 ratio

Air flow = constant x 14.7 x 2.5 x 2,800

= 103,000 x constant lb/min

Air flow at 1,400 rpm and 5:1 ratio

Air flow = constant x 14.7 x 5.0 x 1,400

= 103,000 x constant lb/min

Thus, near equality of air flow at the two speeds is required. Since, with responsiveness, the same horsepower is to be developed at each speed, and horsepower depends upon the pounds of air consumed by the engine, this equality is expected in any case. Thus, the problem is one of delivering a constant mass of air over the engine speed range from 2,800 to 1,400 rpm while the pressure ratio gradually increases from 2.5 to 5.0:1.

Figure 1 can be used as a typical compressor map for a centrifugal compressor having a range of flows and ratios close to that involved in these assumptions. Design can change the shape and efficiencies to a considerable extent, and a machine specifically designed for the problem being considered could have somewhat better characteristics than those shown. The two tip speeds of interest here are 1,200 and 1,700 fps approximately; the latter speed has been recommended for a two-stage design when the characteristics shown in Figure 1 can be improved upon. If we neglect this two-stage requirement and examine the present diagram for a constant air flow, it appears that the machine represented by Figure 1 could provide constant flow of 1.6 lb/sec at 1,200 fps for a ratio of 2.6:1 with an efficiency of about 66%, while for a velocity of 1,550 fps a pressure ratio 4.5:1 at an efficiency of 76% is secured. These ranges of both pressure and efficiency are within a reasonable limit of the desired value. Also, it has been shown that there is excess energy in the exhaust gases; therefore the reduction of efficiency of the compressor of Figure 1 is not of great moment, since the turbine can provide ample power.

Note that the exact specified ratios of 2.5 to 5.0:1 are not completely met by the compressor of Figure 1. However, this machine was not designed for such a method of operation. The diagram does show, however, that the above range of values is not impossible; improvements must be made, or two stages used, in order to meet the exact conditions.

Examination of Figure 1 reveals that the main problem of constant flow comes from the narrow range of flow at the high speed condition. Taking the 1,600-fps flow line as an example, a shift of the surge line to the left by a small amount would improve both the usable pressure ratio and flow without surge.

It can be concluded that the conditions established for responsiveness from 2,800 to 1,400 rpm are almost within the reach of a practical single-stage turbo-charger and could be met completely with little difficulty in a two-stage machine; that there is sufficient exhaust gas energy for turbo-charging; and that abnormal efficiencies are not required.

The additional requirement for a constant mass flow at variable pressure ratio raises the question of turbine nozzle area. Since the pressure ratios are assumed to be 2.5:1 and 5.0:1, if a single-stage turbine is employed the probable expansion nozzle of the 2.5:1 machine will be approaching the choked condition, and the 5:1 will amost certainly be choked. It can then be written, to a first approximation under choked conditions, that:

$$m \propto \frac{A \times P}{\sqrt{T}}$$

where m = mass flow

a = nozzle throat area

p = nozzle throat pressure

It has been shown that mass flow is constant; for the two ratios, therefore,

$$\left(\frac{A \times P}{\sqrt{T}}\right)_{R=2.5} = \left(\frac{A \times P}{\sqrt{T}}\right)_{R=5.0}$$

Now the temperature variation at the throat will not be great, since $T_{2.5} = 1{,}710°$ abs and $T_{5.0} = 1{,}960°$ abs are the gas temperatures at nozzle entry. It follows that for the above equality to exist, since $P_{5.0} \approx 2P_{2.5}$, a variable throat area will be required despite the equality of mass flow.

The turbo requirements can thus be listed as follows:

(1) Pressure ratio requirements of 2.5:1 at high and 5.0:1 at low engine speed.

(2) Variable charger speed from 1,200 to 1,700 fps for the single-stage variety (A two-stage machine is preferred in practice).

(3) Constant mass flow over the speed range for constant horsepower.

(4) Variable turbine nozzle area.

(5) Good but not high efficiencies of the compressors and turbines.

Since all of these properties are now attainable, it can be concluded that a responsive turbo-charged compression ignition engine operating within an engine speed range of about 1.8 to 2.0:1 can be built, provided that the necessary development work on engines and turbos is carried out.

Problems to Be Considered

As with most proposals, there is an accompanying set of disadvantages and difficulties to offset some of the gains. In the case under consideration, the following points must be kept in mind, and evaluated in connection with the overall proposal.

Selected rated load

It has been assumed that the full speed rated load of the engine is kept at a value approximately that of present production engines for combat vehicles, viz., a BMEP of 250 psi involving a supercharge pressure ratio of about 2.5:1. But at the same time no hesitation was made in doubling the MEP at half engine speed, viz., 500 psi at 5:1 ratio charger.

This at once raises the question whether the full speed rating should not also be raised to 500 psi. Should this be done, three alternatives could be employed.

(a) Keep engine displacement as at present and have about twice the horsepower available at maximum speed, resulting in improved acceleration and obstacle crossing. This would lose the responsive character under consideration and would result in a roughly constant torque being developed at all speeds; this is still a great improvement over existing power plants.

(b) Reduce engine displacement to half that used at present, to give the same maximum power as at present in a more compact engine, which could be housed in smaller space, could probably be better protected by armor, and would consume less fuel and weigh less. This engine would, of course, also lose its responsiveness, since if the MEP was held at 500 psi down to half speed the resulting power output at 1,400 rpm would be one half that of case a. There would still be a constant torque at the output shaft; however, this torque would still be superior to that at half speed in a present engine where manifold pressure has been reduced considerably by reduced turbo speed.

(c) Cases a and b could be combined. That is, some semblance of responsiveness could be retained by designing an engine for, say 350 MEP at high speed and 500 psi at low speed. The resulting engine would be somewhat smaller and lighter than present engines, and its performance would approach responsive operation over a smaller speed ratio.

The above appear not exactly as disadvantages but as different ways of effecting vehicle performance without exceeding a specified maximum BMEP, one which seems within the realm of possibility. At present, the resulting variations of the speed range of responsiveness vary from zero to 2:1, but the zero responsive torque at slow speed would be higher than with present models.

What has been termed zero responsiveness here does not mean that torque does not rise as speed is reduced. In case (c) the MEP change from 350 to 500 means that the torque increases in that proportion as the speed slows down. The term responsiveness has been reserved here for that portion of the operating speed range over which the H.P. is constant.

Firing Loads and Stresses

The use of a 5:1 charging ratio will, of course, increase compression and firing pressures, so that improved bearings, lubrication, stress distribution, etc., will be required. The inclusion of some form of variable compression ratio control would lessen these requirements and prevent engine failure.

Acceleration Conditions

Present engines are criticized for the amount of smoke they emit, and the engine being described here would emit even more during a sudden acceleration from idle, because of the high MEP it reaches at slow speed. The reason is that during such changes the speed of the turbo-charger lags behind that of the engine. The proposed responsive charger would reach higher charging pressures faster than a fixed-nozzle charger, because a variable-nozzle design must be employed in the proposed charger. But at the same time a greater fuel supply—equivalent to 500 MEP rather than 250—is being injected into the engine. In view of the improved turbo conditions at both idle and slow speed—i.e., the higher pressure ratio available across the turbine under both conditions—this smoke problem may be no worse than at present except at the very instant of a sudden openings of the throttle; it would not last as long as with present engines.

Specific Heat Flow

Increase in the mean pressure at slow speed will increase the specific rate of heat flow to the coolant, which may result in new demands on lubricating oils, radiators, etc. However, if the engine is responsive, i.e., if it develops about half the maximum MEP at full speed, the heat flow problem at slow speed and maximum BMEP should be little different from that at full speed, since the same H.P. will be developed and the heat to be disposed of is an approximately constant function of the H.P. produced. Some adjustment of cooling fan requirements may be needed.

Controls

Since the fuel supply and the turbine nozzle area must operate in coordination, a control problem arises. At full speed and load the area control will need to give the turbine maximum throat area. As the vehicle encounters increased resistance and the engine slows down and behaves responsively, increased turbine power and speed is required per pound of gas to produce the necessary increase in charging pressure. Unless the fuel throttle position is changed, the control system must operate to reduce the turbine nozzle area, increasing the available energy of the exhaust gases. This must continue to occur as engine speed falls, under maximum resistance conditions, to the half-speed assumed in this report.

If resistance is reduced at any point the fuel throttle will be closed; this will limit the fuel flow and thus the horsepower of the engine. Meanwhile the charger control could maintain the same exhaust pressure setting as before or, perhaps with some advantage, it could permit the intake and exhaust manifold pressures to fall also since there would be more than sufficient air for the combustion process. Development of the most advantageous system of control would need to be based on a part load analysis of the overall problem.

If the engine is idling and a large load is applied, the necessary large increase in fuel supply can be provided almost instantaneously by the fuel pump throttle. In such a case the air required for combustion should also be provided as rapidly as possible. This could be done by having the turbine nozzles set for the high pressure condition at idle so that supercharger speed increases as rapidly as possible.

Continued operation at any intermediate speed and H.P. less than the responsive one could permit a lower manifold pressure in order to reduce pressure loads on the engine parts and to gain in fuel economy if found desirable as indicated earlier.

The control requirement can probably be determined only by a complete analysis. In any case some complication in the control devices seems inevitable.

Surge

Another factor that must be considered is surge. As Figure 1 shows, it appears possible that responsive operation will occur along a line such as a - b of Figure 1. If so, surge will be encountered only at slow engine speed, indicated by b; any further increase in pressure ratio above that at b from any transient condition would produce surge. This type of surge would probably not be very troublesome to overcome, as in normally operated chargers all that is necessary is to have b situated some distance below the surge line.

If the part load analysis should reveal that advantages are to be gained by operating along a series of lines such as c - d at c_1d_1, c_2d_2, etc., at part load conditions, surge might well be encountered at a large number of points as shown. In this case further control mechanisms seem required. A surge indicator or its equivalent could be incorporated in the control mechanism to permit the use of only that portion of, say, c - e which lies within the stable portion of the compressor map. The surge control device would operate on the turbine nozzle area to increase the flow and thus reduce the pressure along a line such as c - e - f in place of c - d.

This type of control seems to be simple since increase of nozzle area automatically increases the range of flow without surge. Adjustment of conditions for satisfactory operation under maximum responsive conditions would result in satisfactory conditions at all other speeds.

CONCLUSIONS

As a result of this analysis, it can be stated that:

(1) A responsive turbo-charged compression ignition engine is a possibility.

(2) The development of a high-pressure-ratio charger (5.0:1 or better) is a necessity if present engine size and weight are to reamin unchanged for full load and speed.

(3) The charger must have a variable turbine nozzle for satisfactory operation.

(4) During responsive operation, the air flow for the engine must remain constant and equal to that at full load and speed as engine speed decreases.

(5) With a 2:1 responsiveness a 2:1 change in exhaust back pressure must be achieved by nozzle area variation, if aftercooler temperature remains constant.

(6) The turbo speed must increase by about 45% as the engine goes from full to half speed.

(7) Control of compressor surge appears to be simpler with a responsive turbo-charger than with a fixed one.

(8) This analysis has not assumed any engine operating conditions that have not been approached in some of the new, advanced single-cylinder engines.

(9) The control problem will need some careful study for the production of a suitable design, but no impossible conditions seem to exist.

(10) A complete study of part load performance seems warranted.

FUTURE WORK

It is considered advisable to conduct a complete performance prediction in order to ascertain what advantages or disadvantages are likely to be encountered in any set of circumstances, to evaluate the control problem, and to gain some knowledge about the most efficient way of operating such a system of engine and turbo-charger. Such an analysis is planned for a future report. In addition, the action of a centrifugal supercharger geared to the turbine via a differential drive will be examined to see if any practical application can be made in this manner to achieve some responsiveness.

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