Dec. 6, 1966 W. M. MAY ETAL 3,289,661

INTERNAL COMBUSTION ENGINE

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FIG.2.

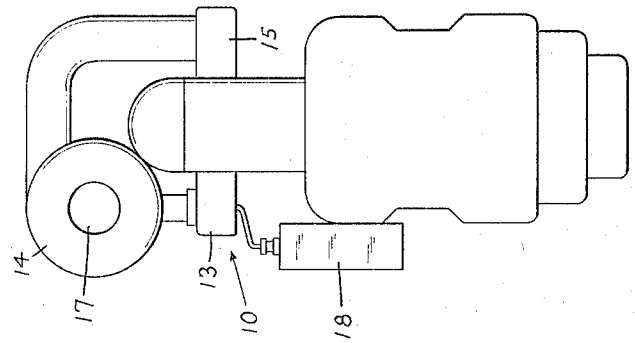
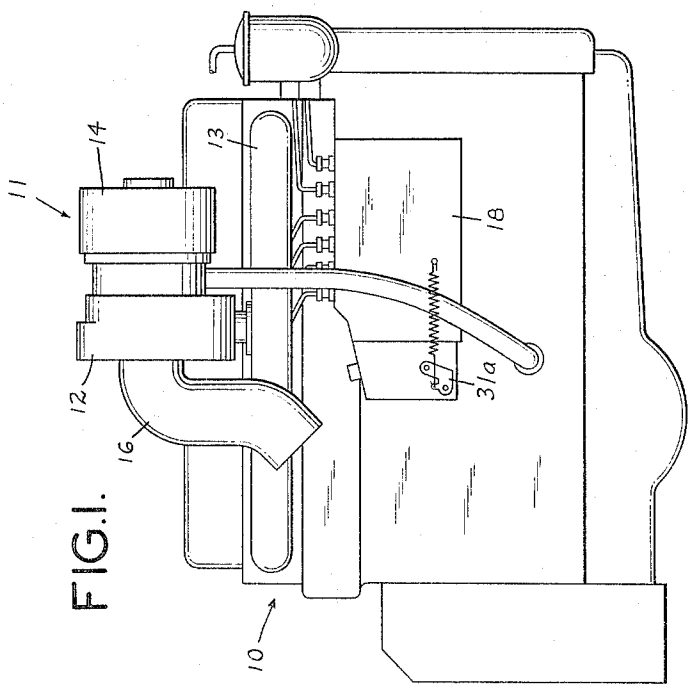


FIG.1.



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FIG.3.

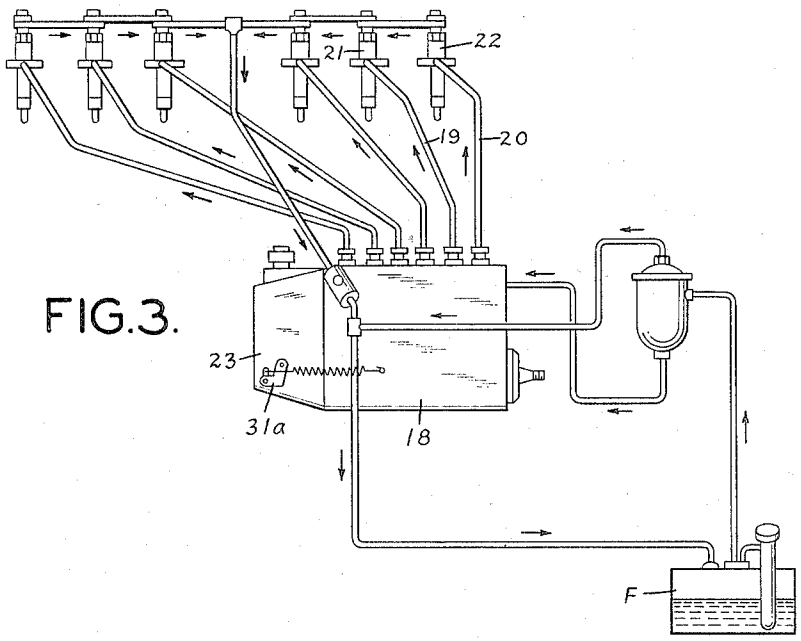


FIG.6.

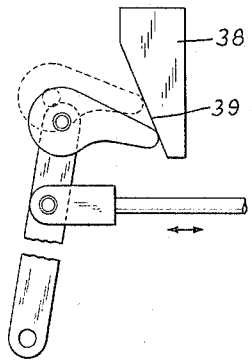
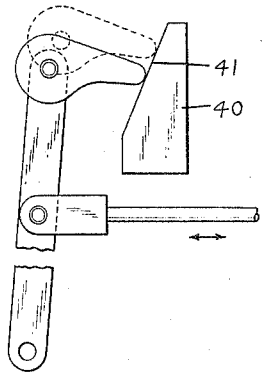


FIG.7.



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FIG. 4.

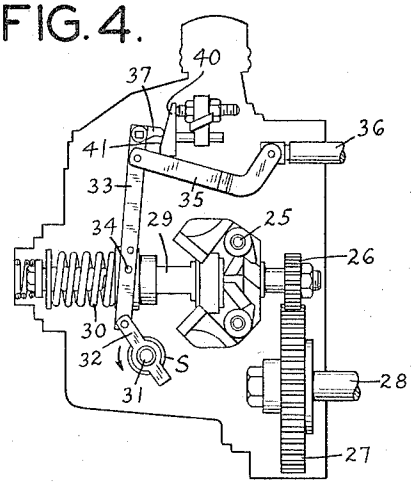


FIG. 5.

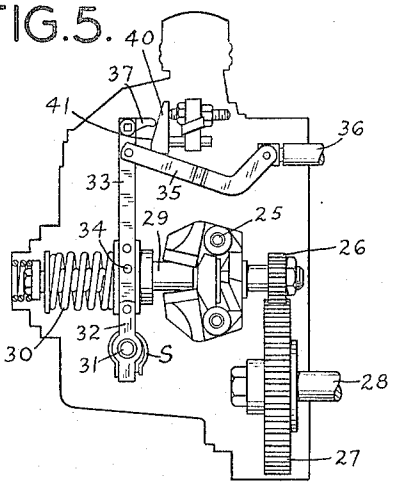
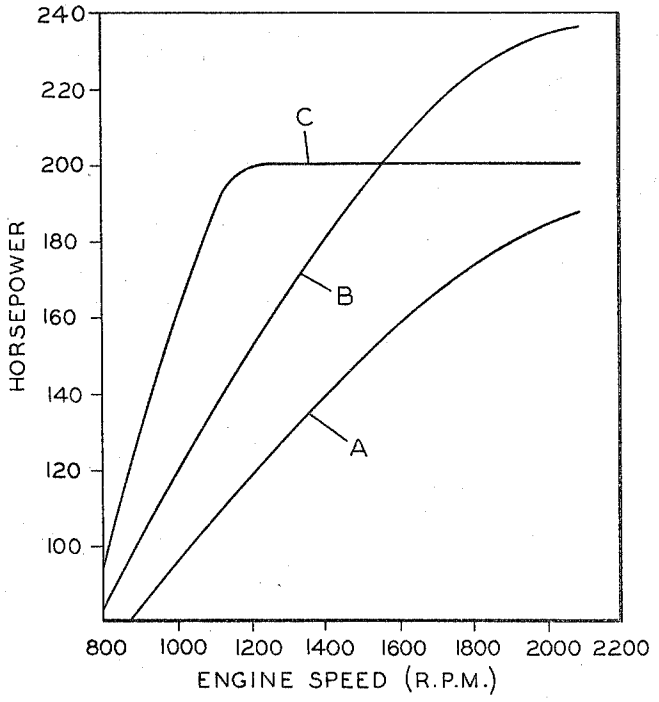


FIG. 8.



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FIG.10.

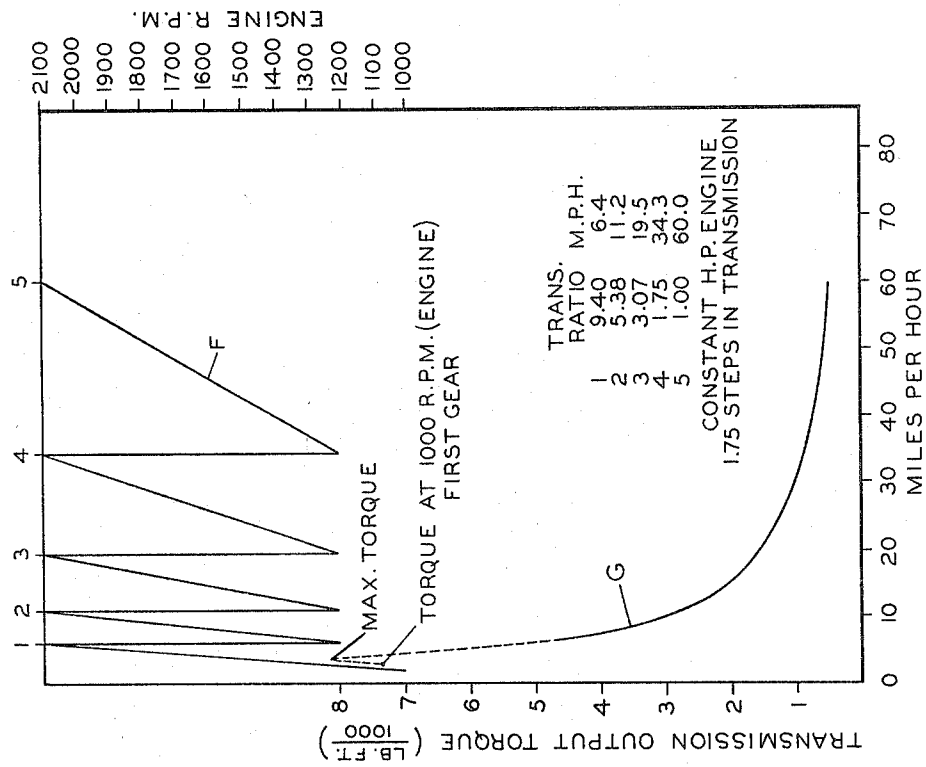
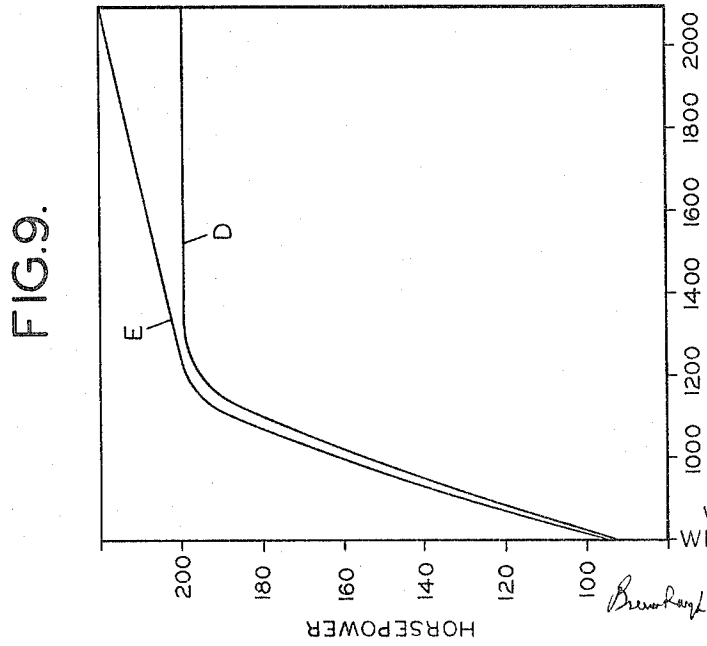


FIG.9.



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INTERNAL COMBUSTION ENGINE

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Filed Sept. 16, 1964, Ser. No. 396,946
5 Claims. (Cl. 123—140)

This invention relates to improvements in internal combustion engines and relates more particularly to fuel and air supply systems for internal combustion engines whereby a substantially constant power output is obtained over a wide range of engine speeds thereby making possible the use of simplified transmissions for supplying power to the driving wheels of a vehicle or other devices.

The reciprocating internal combustion engine is highly developed and is the principal prime mover used in ground vehicles. Nevertheless, it is recognized as being somewhat less than satisfactory for the reason that its maximum power output cannot be utilized unless it is combined with a multiple speed transmission which enables the engine to be operated at or near its full load maximum speed to deliver power to the driving wheels of a vehicle or other device which must operate over a wide range of speeds. The ultimate utility of the engine actually can only be achieved by use of an infinitely variable transmission system. However, the hydraulic, electric or friction drive transmissions of this type are devices which have relatively low efficiency for power transmission and usually are expensive, heavy, big and of questionable durability. The most advanced types of automatic transmissions have not proved suitable for use in heavy duty trucks and similar vehicles because of their size, weight, cost and lack of durability. For that reason, the transmissions which are most generally used in heavy duty trucks and the like are manually shifted, multiple speed, stepped ratio gear boxes having, in many instances, as many as 15 or 20 speed ratios to enable the maximum power output of a reciprocating engine to be used to the fullest practical degree. A transmission of this type is complicated, difficult to shift, heavy, more expensive and larger than is desired.

Because of the deficiencies pointed out above, attempts have been made to develop reciprocating engines that would develop a reasonably constant output over a wide range of speeds. These attempts have not been successful because of a combination of excessive weight, size, complexity, inefficiency and cost of the system. These difficulties have caused the development of internal combustion engines to go in the direction of the development of higher horse power increasing the piston displacement of the engine, increase of engine speed, the number of cylinders in the engine and by the use of superchargers or turbochargers, i.e., exhaust gas driven superchargers.

When a turbocharger is used, the amount of air supplied to the cylinders of the engine is increased as the amount of fuel is increased, with the maximum amount of air being supplied under full load conditions and at the maximum speed of the engine. Thus, for example, in an engine operating under full load conditions or full load governed speed conditions, the turbocharger rotor rotates at the maximum speed permissible with an excess factor of safety to allow for speed up under normally expected high altitude conditions. The turbochargers commonly used have a relatively low efficiency and output in the lower speed ranges and a progressively increasing efficiency and output in the higher speed ranges. The design of such turbochargers can be modified to supply larger amounts of air in the lower speed ranges but the danger of over speeding of such turbochargers in the high engine speed, high output ranges and at high altitudes ordinarily prevents their use.

In accordance with the present invention, reciprocating internal combustion engines are provided with fuel supply means which supplies increased amounts of fuel and proportionate amounts of combustion supporting air in the lower speed ranges of engine operation and lesser amounts of fuel and air than normal in the higher speed ranges of the engine so that a substantially uniform or only slightly increasing power output is obtained throughout a wide range of engine speeds. In this way little change occurs in horsepower output throughout the effective operating range of the engine thereby enabling transmissions of less complex type and having greatly decreased numbers of gear ratios to be used even in heavy duty vehicles and other devices. With engines of the type embodying the present invention, it has been found that the specific fuel consumption of the engine in the normal operating range can be reduced with improved miles per gallon, the engine can be operated at lower speeds with consequent lower piston speeds and less wear on the engine. Moreover, in turbocharged engines, smaller and less expensive turbochargers can be used than with conventional engines and further turbocharger speed compensation for change in altitude is not required because of the reduction in fuel delivery per stroke at the higher engine speeds. Other advantages are that smaller air cleaners and mufflers can be used with engines and the air intake and exhaust systems can also be reduced in size.

The transmission of a vehicle equipped with the new engine can be simplified to the extent that a simple five or six speed gear box can be substituted for a fifteen to twenty speed transmission with full availability of the developed engine horse power over full range of road speed. The simple transmission does away with the need for two stick transmissions or two speed rear axles and can be lighter in weight, smaller and less expensive. Simplified shifting of gears enables an operator to drive the vehicle equipped with a new engine with less effort, over longer periods of time without strain and with greater safety.

For a better understanding of the present invention reference may be had to the accompanying drawing in which:

FIG. 1 is a side elevational view of a typical internal combustion engine equipped with the system embodying the present invention and with the engine accessories omitted therefrom;

FIG. 2 is a schematic front elevational view of the engine showing the manifold connections with the turbocharger;

FIG. 3 is a schematic illustration of a typical fuel injection system for the engine;

FIG. 4 is a schematic illustration of the governor system for the fuel injection pump;

FIG. 5 is a schematic illustration of the governor under another operating condition;

FIG. 6 is a schematic illustration of a portion of conventional governor for a fuel injection pump;

FIG. 7 is a schematic illustration of a portion of the governor for the fuel injection pump modified in accordance with the present invention;

FIG. 8 is a chart of the power output curves of a naturally aspirated engine, a conventional turbocharged engine and the power output curve of the new engine embodying the present invention;

FIG. 9 is a chart showing the power output curve of a bare engine embodying the present invention and the power output curve of an engine embodying the invention constructed and arranged to deliver a substantially uniform horse power to the rear wheels of the vehicle, and;

FIG. 10 is a chart having curves thereon showing the performance of a vehicle equipped with a five speed trans-

mission and shift ranges in the various speeds of the engine.

Referring now to FIGS. 1 and 2, a typical engine embodying the present invention is a fuel injection compression-ignition or diesel engine **10** having a turbocharger **11** thereon including a turbine **12** driven by the exhaust gas discharged through the exhaust manifold **13** of the engine and a centrifugal compressor **14** driven by the turbine for supplying air to the air intake manifold **15** of the engine. The exhaust gases passing through the turbine **12** are exhausted through the centrally disposed exhaust duct **16** to the muffler of the vehicle (not shown). Air is supplied to the compressor **11** through the centrally located air intake **17** which may be connected to an air filter, not shown.

As indicated above, the engine is a diesel engine supplied with fuel by means of a fuel injection pump **18** which, as shown in FIG. 3, receives fuel from a fuel tank F and supplies it through high pressure fuel conduits **19**, **20** etc. to the fuel injection nozzles **21**, **22**, etc. and to the several cylinders of the engine. For purposes of illustration, the fuel injection pump **18** is of conventional type utilizing a constant plunger stroke for each fuel injection nozzle. Such a fuel injection pump is manufactured by American Bosch Arma Corporation and identified as the APE 6 BB fuel injection pump. It is provided with a fuel pump governor **23** generally of the type manufactured by American Bosch Arma Corporation and identified as the GVB/C type of governor which is conventionally supplied with the pump but, as explained hereinafter is modified in accordance with the present invention.

The turbocharger **11** is designed to supply a greater amount of air in the lower speed ranges of the engine than the turbochargers conventionally used on turbocharged engines and the fuel injection pump also is modified to supply larger than normal maximum amounts of fuel in the lower speed ranges of the engine and lower than normal maximum amounts of fuel per engine stroke in the higher speed ranges than the conventional fuel injection system.

Referring now to FIG. 8, curve A illustrates the horse power output of a conventional engine of the non-supercharged or non-turbocharged type, that is, a naturally aspirated engine. It will be apparent that the horse power of this engine increases substantially directly in proportion to the engine speed.

Curve B illustrates the power output of a turbocharged engine of the same kind showing an increase in the horsepower also substantially proportionate to the increase in speed of the engine. The increased horsepower output is obtained for the reason that the engine is able to burn larger amount of fuel per combustion stroke due to the proportionally greater amount of air supplied to the cylinders by the turbocharger.

Curve C illustrates the power output of the engine embodying the present invention. In this engine, a turbocharger having a higher efficiency in the low speed range of the engine supplies larger amounts of air to the cylinders so that a greater amount of fuel can be burned at lower speeds and the horsepower output is higher in the low speed ranges than with the conventional non-aspirated or naturally aspirated engine. The horsepower output is leveled off and maintained substantially uniform throughout a speed range of, for example, about 1200–2100 r.p.m. by decreasing the maximum amount of fuel supplied per piston stroke to the engine with increased speed of the engine. Inasmuch as the amount of fuel is decreased, the exhaust gases are correspondingly reduced in energy and the turbocharger speed does not increase in proportion to engine speed and the turbocharger supplies relatively less air but in the proper proportion for combustion of the fuel. In other words, the amount of air supplied by the turbocharger is maintained in the proper ratio for combustion of the fuel introduced into the system so that the turbocharger is substantially self-regulating and maintains the power output of the engine substantially uniform throughout the predetermined or desired speed range of the engine.

Referring now to FIG. 9, curve D corresponds to curve C in FIG. 8 and shows the constant horsepower output of a bare engine, that is, an engine not driving accessories, transmission and so forth which absorb power that otherwise would be supplied to the rear wheels. Due to the load on the engine of these accessories, a more suitable power output curve E is designed into the fuel and air supply system of the engine so that the delivered horsepower at the rear wheels of the vehicle remains substantially uniform throughout a speed range of about 1200 and 2100 r.p.m.

With an engine having a power output of the type described, a five or six speed transmission can utilize the power output of the engine to essentially the same extent as a fifteen speed transmission with a conventional turbocharged engine. Thus, curve E shows the operation of the engine in r.p.m.'s in the five speed ranges of the transmission and throughout a range between 0 and 60 miles an hour. The variation in engine r.p.m. indicated in curve E when the vehicle or engine is under load is substantially within the constant horsepower speed range of the engine, e.g., 1200 to 2100 r.p.m. and the lower speeds in each transmission ratio approach or include the engine speed at which maximum torque is developed.

The curve G in FIG. 10 illustrates the performance of the new engine with a five speed transmission having the ratios indicated, performing in the range between about 10 and 60 miles an hour.

As to the manner in which fuel supply and turbocharger operation is accomplished, reference is made to FIGS. 4 to 7 of the drawings. Fuel injection pumps are provided with governors by means of which the maximum fuel delivery of the pump is controlled. Referring to FIG. 4, a centrifugal weight governor **25** is driven by means of gears **26** and **27** from the cam shaft **28** which reciprocates the plungers of the fuel pump and is driven by the engine. The governor weights move outwardly in response to increased speed of the engine and inwardly in response to reduced speed of the engine. A collar **29** is moved to the left as the speed increases and to the right by means of a spring **30** as the speed decreases. A shaft **31** carries a lever **31***a* (FIGS. 1 and 3) which is connected to the accelerator pedal linkage to enable the operator to increase and decrease the amount of fuel supplied to the engine and the speed of the engine. Movably mounted on the shaft **31** is a fulcrum lever bracket **32** which is pivotally connected to a fulcrum lever **33**. A torsion spring S resists but permits relative rotation of the shaft **31** and the bracket **32**. The fulcrum lever **33** carries a pivot pin **34** which pivotally engages the collar **29** and is moved thereby to the left or right depending upon engine speed. The fulcrum lever **33** is also connected by means of a link **35** to a rack **36** which controls the amount of fuel supplied by the fuel injection pumps to the various cylinders of the engine. As the rack **36** is moved to the left, the amount of fuel supplied to the cylinders is decreased and as the rack is moved to the right, the amount of fuel supplied is increased. The governor is also provided with a maximum fuel delivery stop plate engageable by a cam follower **37** on the upper end of the fulcrum lever **33**. As shown in FIG. 6, the governor included with the Bosch fuel injection pump has a maximum fuel delivery stop plate **38** of the configuration disclosed in FIG. 6. The cam follower **37** does not engage the plate **38** except when the fuel supply lever and/or the governor combine to rock the fulcrum lever **33** clockwise to provide maximum power output at a predetermined engine speed. As the fulcrum lever bracket **32** is moved counterclockwise by the accelerator pedal of the vehicle the inclined surface **39** on the cam limits the displacement of the rack **36** to the right and the maximum amount of fuel supplied to the

engine provided that the load on the engine is greater than the throttle position demands. As shown in dotted lines in FIG. 6 and in the low speed position of the bracket **32** shown in FIG. 5, the fuel delivery possible with the cam plate **38** is less than with the fulcrum lever bracket **32** and lever **33** in the high speed position as shown in full lines in FIG. 7 and in FIG. 4. The normal operation of the governor, therefore, is to allow the fuel pump rack to be shifted farther to the right with increased fuel delivery in the high speed position as compared with the low speed position of the governor at full throttle. In a conventional engine, the fuel delivery, per piston stroke, remains substantially constant over the speed range under full load.

In accordance with the present invention and as shown in FIGS. 4, 5 and 7, the stop plate **38** is replaced with a new stop plate **40** in which the inclined cam surface **41** is slanted in a direction to decrease the fuel delivery per piston stroke to the engine as the cam follower **37** is moved downwardly between the low or medium speed positions shown in dotted lines and the high speed position shown in full lines. Consequently, in the higher speed ranges sensed by the governor **25**, the maximum fuel delivery per stroke supplied to the cylinders is decreased as the engine speed increases with the result that the horsepower output remains substantially constant, as explained above, throughout the range of normal driving speeds of the engine. Also as indicated before, the decreased amount of fuel delivered to the engine at high engine speeds reduces the volume of exhaust gases so that the turbocharger is driven at a relatively lower speed and the amount of air supplied to the cylinders is decreased and maintained in a proper ratio for efficient combustion of fuel in the engine. The combined effects of increased fuel and air supply in the lower engine speed range and decreased fuel and air supply in the higher engine speed range provide an engine output of the type indicated in curves C and D of FIGS. 8 and 9 with the resulting benefits.

While the maximum horsepower output of the engine may appear to be reduced as compared to a conventional turbocharged engine, the fast that a horsepower output remains constant in the operating range of the engine and in parts of the operating range is higher than with a conventional engine, the overall effect is increased engine efficiency and a high power output at the low end of the operating range at or near to the engine speed at which maximum torque is developed. Therefore, the apparent loss in horsepower output at the high engine speeds is more than compensated for by more uniform power delivery throughout the operating speed range. More uniform horsepower output allows the engine to be operated with a lower average fuel input in the normal operating speed range with resulting fuel economy. Also inasmuch as the turbocharger is not required to supply as much air for combustion of the fuel in the maximum speed range of the engine, the air filter and exhaust system can be reduced in size as can the turbocharger itself with substantial savings in weight, cost and complexity.

While the invention has been illustrated as utilizing a conventional fuel injection system in which the fuel injection pump is controlled by a modified mechanical governor, it will be understood that hydraulic, vacuum, velocity and electrical governors also can be modified in similar manner to control the amount of fuel injected in the operating speed range to achieve the desired results. Also, the system can be applied to naturally aspirated compression-ignition engines and to internal combustion engines of the spark-ignition type with or without a turbocharger through suitable governor controls to the carburetor linkage to regulate the amount of fuel supplied to the cylinders. Accordingly, the form of invention described herein should be considered as illustrative and the invention is limited only as defined by the following claims.

We claim:

1. A turbocharged internal combustion engine having a substantially uniform horsepower output over the range of useful operating speeds, comprising a reciprocating internal combustion engine, means for supplying fuel to said engine for combustion therein, operator-controlled means for regulating the supply of fuel charge to said engine, engine-speed controlled means for regulating the maximum supply of fuel to said engine, means in said engine-speed controlled means for progressively increasing the maximum fuel charge supplied to the engine as the engine speed decreases, and a turbocharger efficient in the lower useful operating speeds of said engine and driven by combustion gases from said engine for supplying air to said engine for combustion of fuel therein, said engine and turbocharger combination having a volumetric efficiency which is substantially constant throughout said range of useful operating speeds, producing an increasing positive scavenging pressure differential with decreasing engine speed and an inlet manifold charge pressure substantially constant throughout said range of useful operating speeds thereby matching the increasing fuel charge with decreasing engine speed to produce substantially constant horsepower output throughout said range of useful operating speeds.

2. The engine set forth in claim **1** in which said engine speed-controlled means comprises a governor, and said means for progressively increasing the fuel supply comprises a cam cooperating with said governor to increase the amount of fuel supplied to the engine as the engine speed decreases.

3. The engine set forth in claim **1** in which said means for supplying fuel to said engine is a fuel injection system.

4. The engine set forth in claim **3** in which said engine is a compression-ignition engine.

5. The engine set forth in claim **1** in which said operator controlled means comprises a pivotally movable lever having a cam follower thereon, and means connecting said lever to said means to supply fuel to increase and decrease the amount of fuel supplied in accordance with the movement of said lever, and said engine speed-controlled means comprises a governor having a displaceable collar, means connecting said lever pivotally to said collar for moving said lever in response to engine speeds, a fixed cam having a surface engageable by said cam follower for limiting movement of said lever at maximum fuel supply and progressively increasing the movement of said lever as the engine speed decreases to increase the amount of fuel supplied to said engine.

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