# The General Motors Fuel Injection System

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THE carburetor expert knows the difficulty of properly metering fuel with a carburetor to satisfy all engine conditions: speed, load, temperature, altitude, operating transients, and fuel variables, but with patience he may achieve a fairly good "flow curve" or "calibration curve." Next, the air and fuel have to be mixed as uniformly as possible either within the carburetor or between carburetor and center portion of manifold, commonly called the "riser." The more or less heterogeneous mixture must then be carried through the intake manifold to the individual cylinder ports. Transporting the mixture to the intake ports with a minimum of fuel separation requires (1) maintaining a high flow velocity in the manifold by using small passages and (2) adding heat for further fuel vaporization. At best, this combination leaves some lean cylinders. The overall mixture must then be richened to assure adequate fuel for the leanest cylinder.

The resulting average carburetor and manifold system may then have the following inherent deficiencies:

1. It may use 10% more fuel at wide-open throttle than it should. This not only represents a waste of fuel but also exhausts to the atmosphere annoying amounts of carbon monoxide.

2. Cold starting and warmup may require substantial amounts of fuel to provide the initial wetting of the manifolds, and still have left adequate liquid and vaporized fuel for the cylinders to insure firing. This adversely affects fuel economy and increases air pollution.

3. The hot-spot manifold required by carburetors results in (1) loss of bowl fuel through evaporation because of the increased carburetor temperature, (2) promotion of vapor lock, and (3) some octane depreciation of the fuel because of increased mixture temperature.

4. Manifolds have to be a compromise for volumetric efficiency and good distribution and may not permit full utilization of the power potentiality of the engine.

5. Decreasing manifold vacuum during throttle opening transients increases the manifold wetting. For a smooth engine feel during the transient, additional fuel is required as "make-up" for the increased wetting. This "make-up" fuel supplied by the accelerator pump may, under certain conditions, represent

**THIS paper discusses the fuel injection system which General Motors has been working on. According to the authors it will give greater simplicity, economy, and reliability than the conventional carburetor.**

**Basic development work was done on single-cylinder units. Air and fuel metering methods are discussed for best performance during such operations as warmup and idle.**

**Basic components of the fuel injection system such as the filter and pump are described in detail for their design and performance. Comparison with performance of conventional carburetors is also made.**

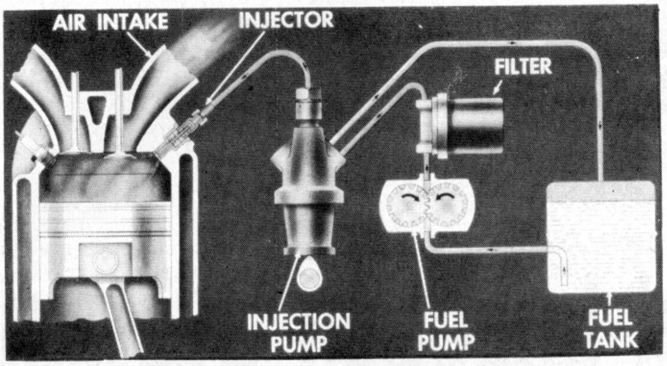

Fig. 1—Aircraft injection system

a loss because this fuel is mainly exhausted as unburned hydrocarbons.

A fuel system capable of correcting even 50% of the average carburetor deficiencies may produce a 10% improvement in fuel economy that would ultimately result in substantial savings to the customer.

The possibility of an exhaust free from hydrocarbons and a fuel system free from evaporation would greatly help in an important phase of reducing air pollution.

It was with these goals and good intentions in mind that a development program was initiated at the General Motors Engineering Staff several years ago toward achieving the best possible fuel system.

*Early Fuel Injection Systems*—Since the aircraft engine had the most exacting carburetion requirements, this industry was first to recognize the carburetor's limitations and to develop new systems.

One of the earliest successful systems consisted of injecting fuel into the combustion chamber (Fig. 1). A diesel-type injection pump was provided with metering controls to obtain the air-fuel mixture required for spark ignition engines. This system did a good job of fuel distribution in the rather narrow cruising and full power range of aircraft engines. This injection system not only overcame the difficult distribution problem on radial aircooled engines, but also eliminated the fire and explosion hazard created by the combustible mixture in the manifold and supercharger of carbureted radial and V-type aircraft engines.

Subsequent development work indicated that:

1. The design of the nozzle could be simplified and its reliability enhanced by injecting fuel into the intake ports rather than into the combustion chamber.

2. No appreciable power difference existed between port and combustion chamber injection.

3. The use of individual plungers for each cylinder was sufficiently accurate for aircraft engines, which operate at fairly substantial bmep, but was very poor for automotive engines at idle and city driving because of poor distribution characteristics inherent to this system at low fuel rates.

4. The system, even with volume production, was too high priced for general use. At the time of our early work it was possibly eight times as expensive as the carburetion system.

Another interesting system that achieved a great deal of attention by Italian manufacturers was the Fuscaldo method (Fig. 2). This system consisted of a source of fuel under pressure at the engine intake ports. Electromagnets opened precision-made valves at each intake port to deliver fuel in relation to engine requirements. While the system offered some simplification over the diesel-type pump and injector, it did not appeal to us as the ultimate in simplicity and reliability.

After a thorough analysis it became evident to us that none of the existing fuel injection systems could compete with carburetors for simplicity, low cost, and low-speed performance and at the same time correct the mentioned deficiencies. Therefore, we decided to work out our own system.

### Fuel Injection System

*Single-Cylinder Tests*—The basic development work was done on single-cylinder units. From these test results obtained at a compression ratio of 8/1 (Fig. 3), we established that:

1. Direct injection into the combustion chamber had no appreciable advantage over injection into the intake ports.

2. Intermittent (timed) injection in the intake ports gave slightly less power and used slightly more fuel than continuous injection.

3. Injection directed toward the intake valve gave the most power, the best economy, the fastest warm-up, and the best acceleration response.

Based on these facts, we chose continuous-flow injection into the intake ports with open orifice nozzles directing the fuel toward the intake valves. Further development work covered extensively the function of every component and represented successive steps of simplification of design, improving metering means, and increasing reliability. I shall not take time to trace this development work. Instead, I believe a short discussion of metering in general would be most helpful before describing the operation of our fuel injection system in detail.

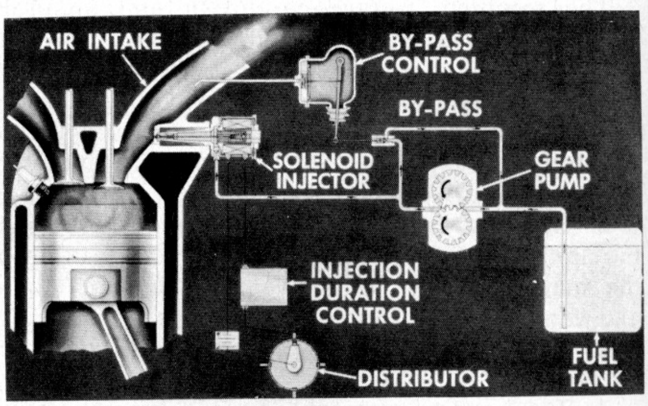

Fig. 2—Fuscaldo injection system

*Metering Methods* (*General*)—Two methods of air and fuel metering were considered for our continuous-flow fuel injection system. These methods can be classified as speed-density metering and mass flow metering.

1. Speed-Density Metering. The fuel injection systems used for aircraft were controlled by means of engine-speed air-density responsive metering systems. This choice may have been dictated by the wide air-density range encountered in flight. The sensor of pressure and of temperature must be correct in proportion to absolute temperature and absolute pressure.

An ambient temperature change from −30 to 100 F or 430 to 560 R, requires a metering change of $560/430 = 1.30$ or 30%. This means that air at −30 F requires 30% more fuel than the same volume of air at 100 F. A similar large change would be shown versus atmospheric pressure (as climbing Pike's Peak).

The speed-density metering system requires a device which will: (1) sense absolute temperature and pressure accurately and (2) correlate these absolute values to the metering means. These requirements make the sensing device costly. Furthermore, this type of metering requires means to measure engine volumetric efficiency versus speed which, on "rammed engines," may prove to be rather complicated and costly.

The speed-density metering system usually requires a supply of fuel in equal amount per cycle. The fuel pump must therefore be a uniform-flow pump which may not depreciate in service.

2. Mass Flow Metering. Let us now examine the method of metering in function of mass of air consumed by the engine, in other words, by using a high efficiency venturi. The temperature and pressure effect have to be corrected only in function of the square root. Consequently, a temperature change from 430 to 560 R causes a metering change of only $\sqrt{560/430} = \sqrt{1.30} = 1.14$ or 14% as compared to 30% in speed-density metering. Actually, fuel-density variations with temperature cancel 4% of this change, and proper utilization of fuel vapor bubbles at high temperatures tends to cancel a large part of the remaining 10%.

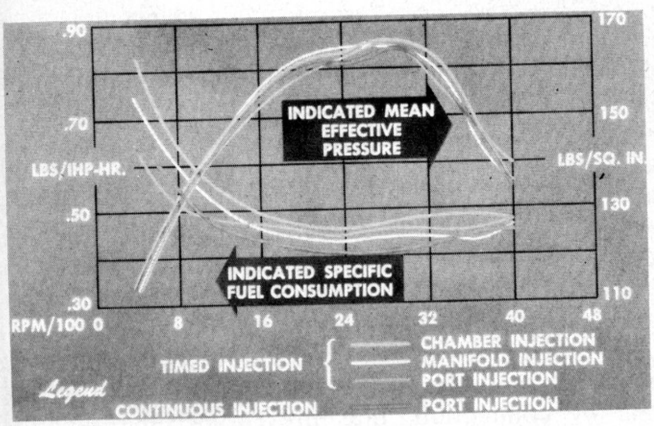


Fig. 3—Comparison of timed and continued injection

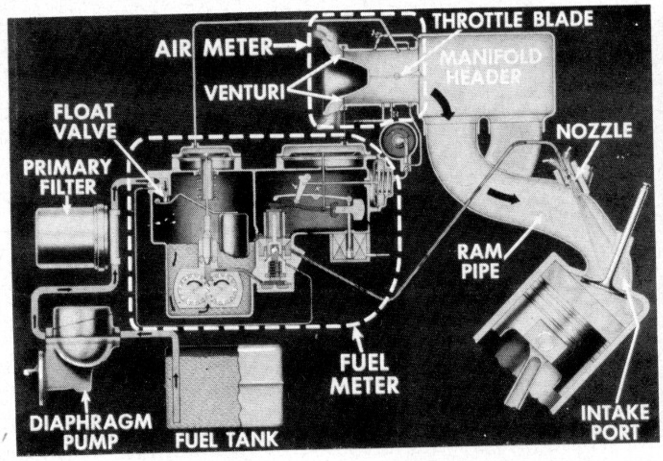


Fig. 4—General Motors injection system

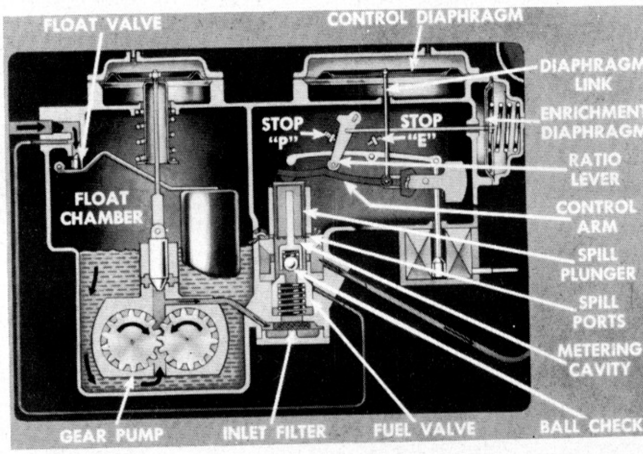


Fig. 5—Fuel meter

The volumetric efficiency of the engine is sensed by the venturi, and no additional correcting signal is required. The pump delivery does not have to be constant (as is the case with the speed-density system) as long as it is in excess of the desired flow at the nozzles.

The control method selected for our injection system is based on mass flow metering in which venturi throat depression is related mechanically to fuel pressure. (See Appendix.)

*Metering System*—Fuel supplied by a conventional 6-psi diaphragm pump (Fig. 4) passes through a 10-micron primary filter to the fuel meter. The fuel is admitted through a float valve—similar to its counterpart in a carburetor—into the float chamber where any vapors which have formed due to temperature rise are vented. A small gear pump located in the bottom of the float chamber delivers the fuel through a second inlet filter and a fuel valve to the metering cavity (Fig. 5). The fuel valve contains an antipercolation ball check to keep the fuel between the valve and the pump at a sufficiently high pressure to elimi-

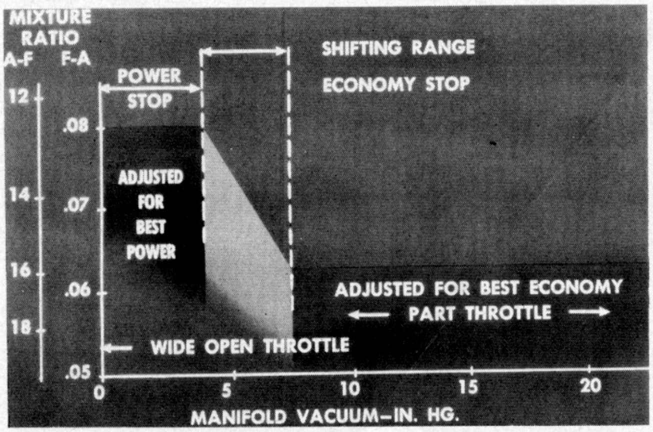


Fig. 6—Operating mixture ratios

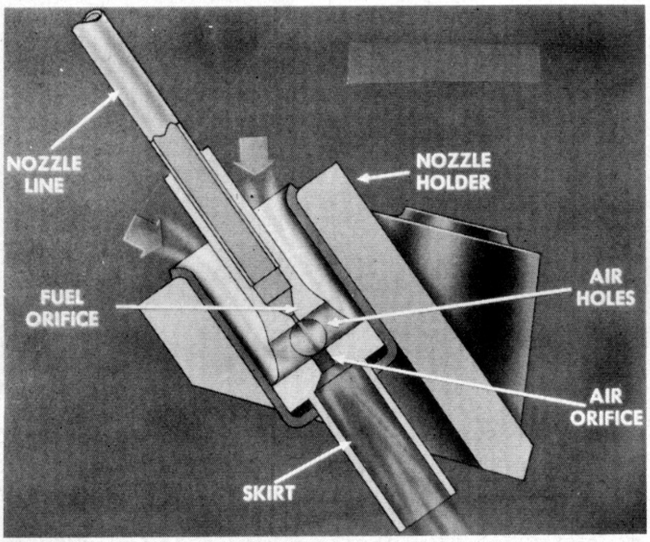


Fig. 7—Injection nozzle

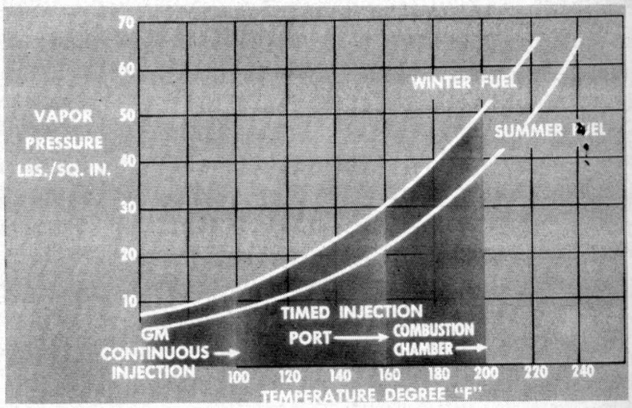


Fig. 8—Average nozzle operating temperatures

nate vapor pockets. Some of the fuel delivered to the metering cavity flows directly to the eight continuous spray nozzles; the remainder of the fuel flows through the spill ports back to the float chamber. The amount of fuel spill is regulated by the spill plunger.

The air enters the air meter through the annular venturi section and flows past the throttle blade to the manifold header and then to the individual intake ports. The air velocity through the venturi causes a depression signal related to the mass airflow. This venturi signal acts on the control diaphragm to create a force which is transmitted to the spill plunger by means of the diaphragm link and control arm, the latter being pivoted on the ratio lever.

An increase in airflow through the venturi causes a relative increase in venturi signal which, acting on the control diaphragm, results in an increase in the force acting on the top of the spill plunger. The spill plunger then moves to a new balanced position to obtain a fuel pressure increase proportional to the venturi signal increase. Since the increased fuel pressure results in a fuel flow proportional to the increase in airflow indicated by the venturi signal, a constant air/fuel ratio will be maintained as long as the linkage ratio is not changed.

The air/fuel ratio can be varied by changing the linkage leverage between the control diaphragm and the spill plunger. This is accomplished by shifting the pivot point on the control arm. When the "ratio lever" rests against stop "P" (Power), the air/fuel ratio obtained is for wide-open throttle operation; (Fig. 6) when resting against stop "E" (Economy), part-throttle fuel requirements are fulfilled. For automatic operation, the ratio lever is controlled by the spring-loaded enrichment diaphragm which is subjected to manifold vacuum; at light load (high vacuum) the lever is held at position "E," at full load (low vacuum) at position "P."

The stops are set to obtain a best economy air/fuel ratio (about 15.5/1) at part throttle and a maximum power air/fuel ratio (about 12.5/1) at wide-open throttle. The enrichment diaphragm spring is set to hold the ratio lever in the economy position above about 7 in. of Hg vacuum.

The nozzle (Fig. 7) has an accurately calibrated open-fuel orifice somewhat less than 1/64-in. diameter. The nozzles are mounted, at the entrance of each intake port, in plastic nozzle holders. Just below the fuel orifice, there is a small air chamber which receives filtered air through four 0.100-in. diameter air holes from the air cleaner. This air supply assures that the nozzle discharge is at all times near atmospheric pressure regardless of manifold vacuum fluctuations, and consequently the amount of fuel injected depends solely on the metering system pressure. In line with the fuel orifice across the air chamber is the opening to the intake port. This is a 0.040-in. diameter air orifice. It should be noted that under all manifold vacuum conditions the atomization of fuel is enhanced by the mixing of air and fuel through this 0.040 orifice. The air volume passing through the air orifice represents about one-fourth of the air used at closed throttle for idling.

The tubular skirt below the air orifice is cooled by evaporation of the fuel which in turn cools the fuel line by conduction, thus preventing metering disturbances caused by fuel vapor bubbles. In fact, the fuel at the nozzle and in most of the line is at a lower temperature than when it entered the system from the diaphragm pump.

By using this scheme of refrigerating the fuel by means of its latent heat, this open-orifice type of nozzle becomes suitable for handling high vapor pressure fuel even under summer conditions. The nozzle operating temperature range of this nozzle, due to its location and insulation, is much lower than that of nozzles used in other fuel injection systems (Fig. 8). This lower temperature range greatly reduces vapor problems.

The fuel orifice size was selected to give a minimum pressure at idle and cranking of about 8-in. fuel head and a maximum pressure of the order of 200 psi at engine top speed and wide-open throttle. Under normal driving conditions, the pump pressure seldom exceeds 20 psi (Fig. 9). These moderate pressures can be achieved with a plain gear pump.

The fuel pressure at idle and when starting has been found sufficiently high to prevent adverse distribution effects caused by normal engine attitude changes.

*Idle Operation*—During idle operation the throttle is closed and about one-fourth of the air requirement is supplied through the individual nozzle air chambers. The rest is provided through a passage around the throttle blade and is regulated by the idle-air by-pass screw (Fig. 10) within the throttle body. The correct idle mixture is set by means of the idle mixture screw. This screw controls the amount of venturi signal boost caused by bleeding in manifold vacuum from the idle orifice.

*Off-Idle Enrichment*—To give best economy for steady-state operation at idle and off-idle, the engine requires a mixture which is richer than for higher speeds (Fig. 11). The basic venturi system results in a leaner mixture at low airflows but, because of the plunger weight and the nozzle placement below the spill level in the fuel meter, there is some inherent enrichment near idle. The remainder of the required idle mixture enrichment is supplied through the idle orifice, and the remainder of the enrichment required in the off-idle range is supplied through the off-idle orifice (Fig. 12).

As the throttle is opened from the idle position, the off-idle orifice is subjected to manifold vacuum, thereby intensifying the venturi signal. The amount

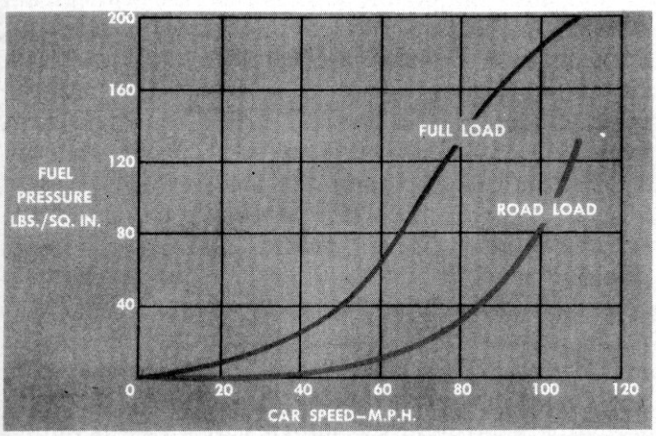


Fig. 9—Fuel pressure versus car speed

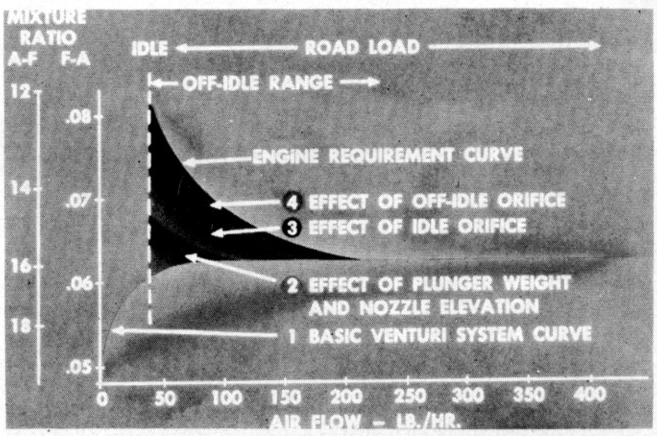


Fig. 11—Idle and off-idle mixture ratios

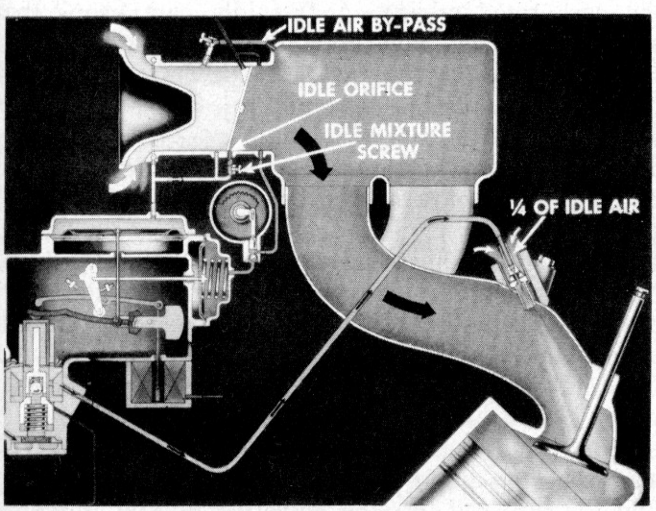


Fig. 10—Idle operation

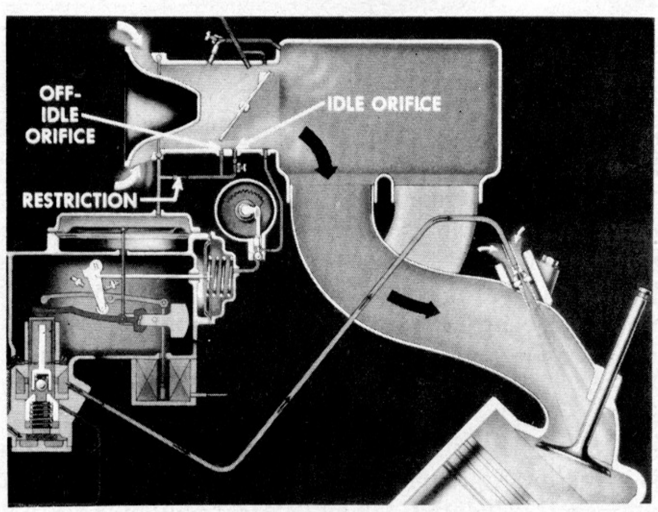


Fig. 12—Off-idle enrichment

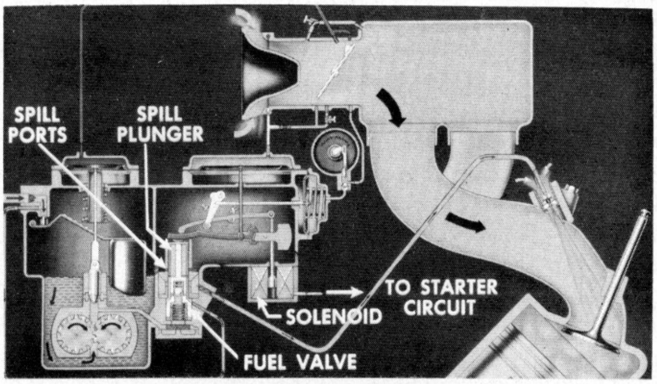


Fig. 13—Cold starting

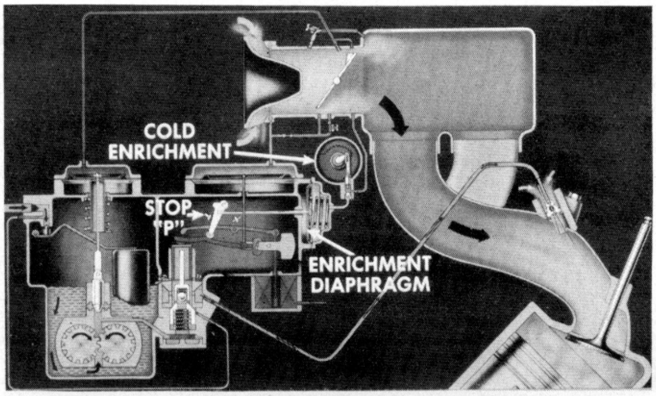


Fig. 14—Warmup enrichment

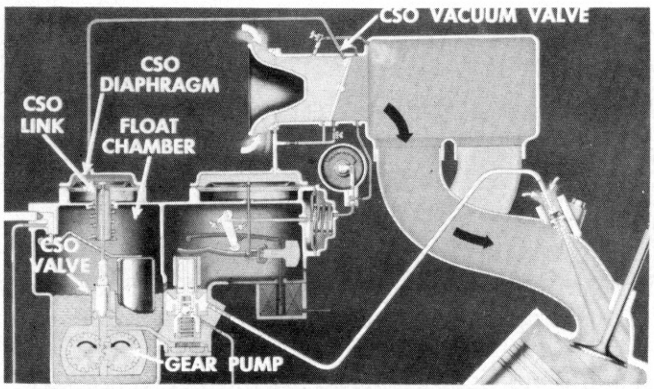


Fig. 15—Coasting shut-off

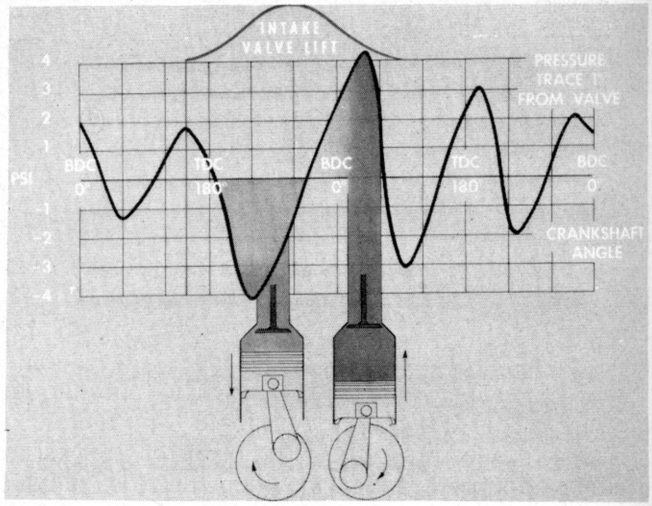


Fig. 16—Dynamic supercharging

of off-idle bleed is regulated by the restriction in the bleed passage.

*Transients*—Our fuel injection system does not require an accelerator pump to supply extra fuel during throttle opening transients. Two factors are responsible for the elimination of the accelerator pump: (1) manifold wetting is kept to a minimum by port injection and (2) instantaneous fuel meter response. Upon throttle opening, the manifold is filled by an immediate inrush of air greater than the stabilized air requirement, causing an equivalently larger venturi signal. Since the fuel meter linkage parts are very light and the travel small, their inertia is negligible so that the inrush of air will instantly cause a rise in fuel pressure. The result is a shot of fuel proportional to the inrush of air.

*Cold Start and Warmup*—Because of the direct distribution of the fuel to the intake ports, it is not necessary to provide a substantial fuel enrichment to wet and flush the manifold as is the case with a carburetor. The starting enrichment required is achieved by means of a solenoid connected to the starter circuit. This solenoid (Fig. 13) forces the plunger down far enough to unseat the fuel valve. This action results in: (1) completely cutting off the spill by covering the spill ports and (2) uncovering a passage from the float chamber inlet to supply extra fuel directly from the diaphragm pump.

During warmup the cold enrichment mechanism keeps the ratio lever at the power stop position (Fig. 14) by cutting off manifold vacuum to the enrichment diaphragm. This enrichment assures solid steady-state engine operation during warmup. By directing the fuel stream upon the intake valve, the incoming mixture is warmed up considerably faster than by means of the "hot spot" of the intake manifold of the carbureted engine. A fast idle cam keeps the engine idle speed high enough to eliminate stalling by slightly opening the throttle blade during warmup.

*Coasting Shut-Off*—The fuel injection system operates with dry manifold, and consequently, the amount of fuel vaporization in the manifold during deceleration at closed throttle is negligible. A further step to eleminate exhaust fumes containing unburned fuel during coasting is provided by a "coasting shut-off" (CSO, Fig. 15) located directly above the gear pump outlet, and consisting of a valve, which under normal operating conditions is closed by a spring. A CSO link connects the valve to the CSO diaphragm which is subjected to manifold vacuum at the throttle body. During coasting, high manifold vacuum exerts a force on the diaphragm in excess of the spring load, and the pull rod lifts the valve off its seat, thereby allowing the pump to discharge directly back into the float chamber, no fuel flowing to the nozzles. The vacuum valve is so arranged that vacuum can be applied to the diaphragm only when the throttle is closed to eliminate CSO action as the engine is accelerated at no load.

### Dynamic Supercharging

Fuel injection makes possible the use of any type of supercharging, the most simple one being a dynamic supercharging method by means of ram pipes. This phenomenon is basically similar to the hydraulic ram or "water hammer" occurring when rapidly closing a water faucet at the end of a long pipe. The kinetic energy of the fluid is momentarily spent to produce a pressure rise. A typical pressure record obtained with a pressure pickup located close to the intake valve (Fig. 16), shows the amount of ram that can be realized. The effect upon the engine torque characteristics of this type of supercharging is well known. Fig. 17 shows imep versus rpm curves for a V-8 engine using various lengths of ram pipes. Their effective use is one of the several indirect gains made possible by fuel injection.

The basic components of the General Motors fuel injection systems are shown in a schematic cross-section (Fig. 18). This illustrates the system's basic simplicity and inherent self-compensation for wear and manufacturing variation.

The spill plunger always controls spill rate to accomplish basic fuel metering as it seeks to balance:

1. The force exerted on the control diaphragm by the static pressure drop at the venturi.
2. The force exerted on the spill plunger by the nozzle metering-pressure.

Airflow establishes a diaphragm force which is transmitted through the control arm to the spill plunger. The plunger is depressed to reduce the flow of spill fuel and raise the pump outlet pressure. The plunger seeks a spilling position where the fuel and air forces balance. The spill-plunger fuel pressure is applied to the nozzles.

The "gear" pressure pump is purely an energy source and cannot affect calibration providing it delivers a supply of high-pressure fuel in excess of engine requirement. Normal wear or manufacturing variation of the spill plunger, the sleeve, the gear pump, or the mechanical components will produce changes in operating positions and fuel leakage. However, these changes will not upset the air-fuel force-balance or the metering calibration, hence the system is self-compensating.

A primary objective of fuel injection is improved distribution. With continuous injection the nozzles must be matched (Fig. 19). Present practice is to guarantee that on any engine the richest nozzle will not be more than 5% richer than the leanest. Unfortunately distribution may be acceptable at low pressure but out of limits at a higher pressure. Semi-automatic equipment establishes the flow calibration to eliminate operator error. Each nozzle, at both a high and a low flow, is classified by the time required to pass a fixed volume of fluid. All nozzles on the same unit are then within 5% at either calibration point. Actually the production variation in nozzle flow is about 15% at high and low calibration. If

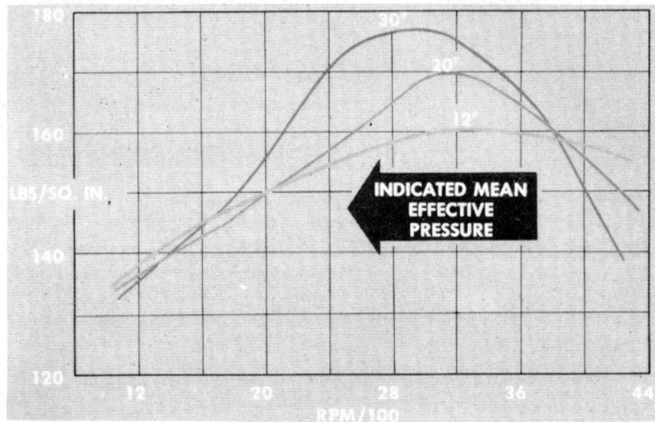


Fig. 17—Comparison of ram pipe lengths

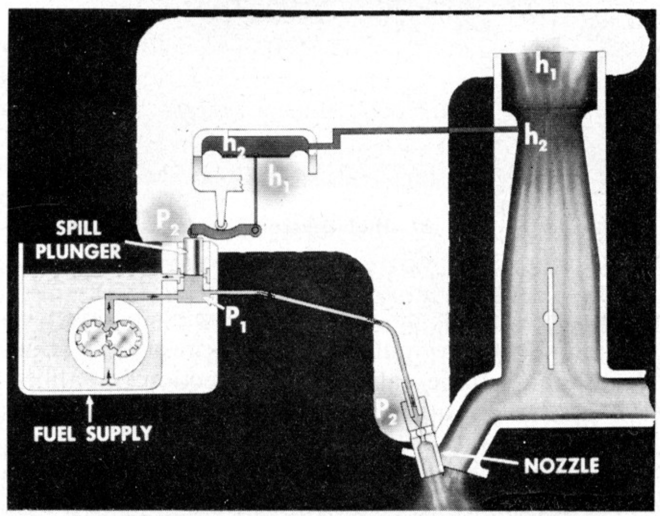


Fig. 18—Fuel injection system of General Motors

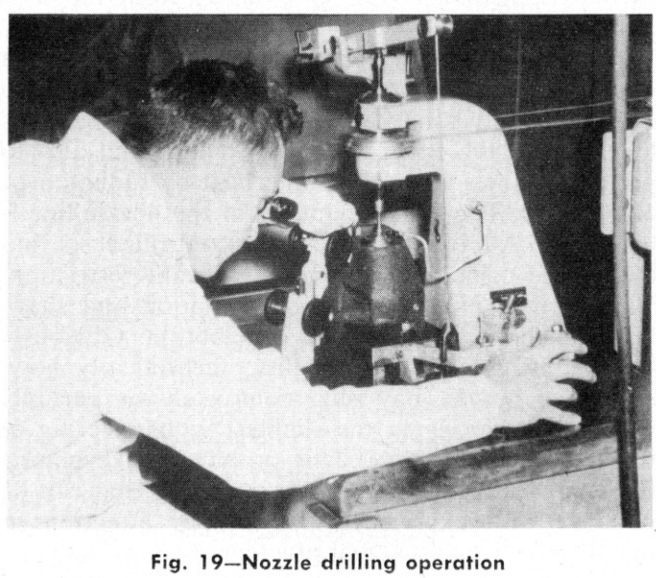

Fig. 19—Nozzle drilling operation

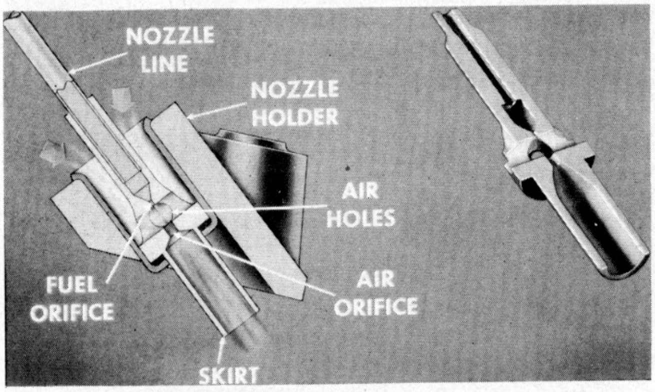


Fig. 20—Injection nozzle

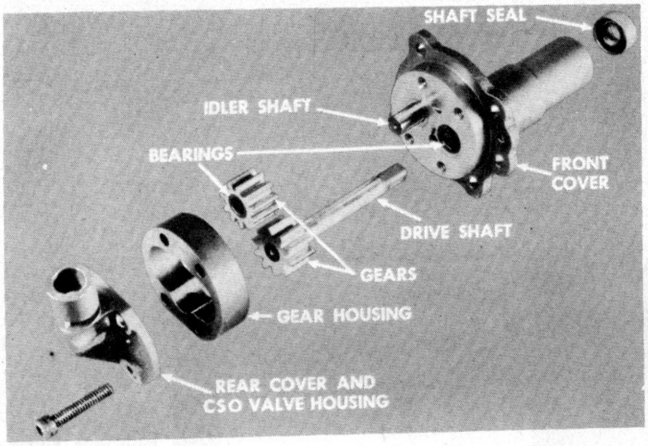


Fig. 21—High-pressure pump

all nozzles are to be used, overlapping bands with varying slopes are required and this results in about 20 different nozzle calibrations. Selection of calibration assists in satisfying a specific engine model's fuel requirement.

The nozzles (Fig. 20) are fed by individual tubes which are either soldered or attached by a tube fitting. If threads are added to the nozzle they are external so that tightening chases chips away from the tube joint.

Fuel flows between the two lands of the spill sleeve to either one or two distributors and then to each nozzle. It is important to make all connections to eliminate the entry of foreign matter and also to protect all lines from absorbing heat by radiation or conduction. The inside diameter of the nozzle line is 0.049 in. All fuel passages are controlled so that the nozzle orifice is the only significant restriction. This permits minor line-length variation and sharp bends without affecting fuel distribution. Obviously nozzle plugging is a possibility; theoretically however, one nozzle may plug completely or partially without interfering in the slightest with metering of the remaining nozzles. This problem, (like getting three square meals a day), is something that can be handled with experience and proper attention to details as proven by actual practice.

Some logical steps toward the solution of this problem are:

1. Cleanliness of parts and of the assembly and test areas.
2. Provision of reliable filters in the fuel system.
3. Protection in storage or shipment from entry of foreign matter.
4. Prevention of part deterioration in production or service which may result from corrosion, wear, heat-treat scale, loose burrs, or inadequate plating.

### High-Pressure Pump

For reasons of economy and reliability, it is desirable to use a pump of basic simplicity. The gear pump offers many advantages in this system (Fig. 21).

1. There are no valves in the pump.
2. No inlet piping is required.
3. Wear on gear-tooth form does not affect efficiency.
4. A pressure relief valve is not required—the spill plunger performs this function.
5. Only one shaft seal is required, and this is subjected to low pressure. Other shaft leakage can easily be dumped to the float bowl.
6. The pump is immersed in fuel which keeps it at bowl fuel temperature.

The pump is flange-mounted to the side of the bowl with a $\frac{1}{64}$-in. Buna N rubber-on-nylon gasket. Delivery to the fuel meter is routed through this flange. A second delivery path is opened to prevent fuel delivery to the nozzles when the car is coasting. The pump runs at one-half the engine speed. At any engine operating speed, the fuel delivery averages twice the wide-open throttle requirement. This allows compensation for normal wear and is a decided benefit to instantaneous acceleration. Power is taken from the ignition distributor driveshaft through a shielded flexible shaft.

The poor lubrication of gasoline and the 200-psi maximum pressure necessitate special manufacturing

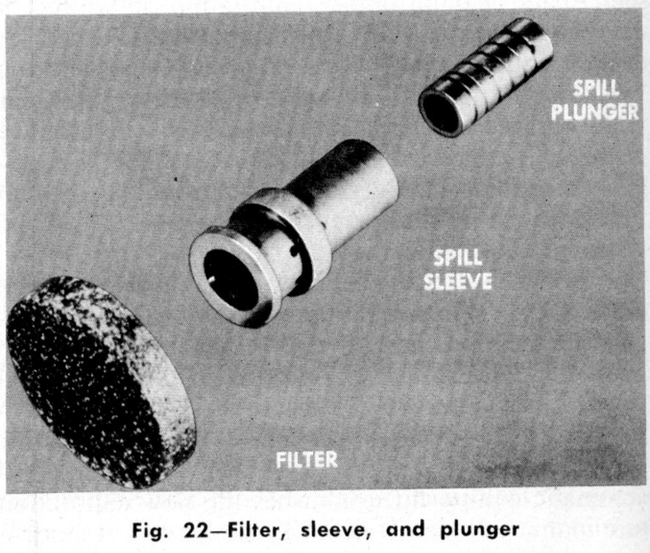


Fig. 22—Filter, sleeve, and plunger

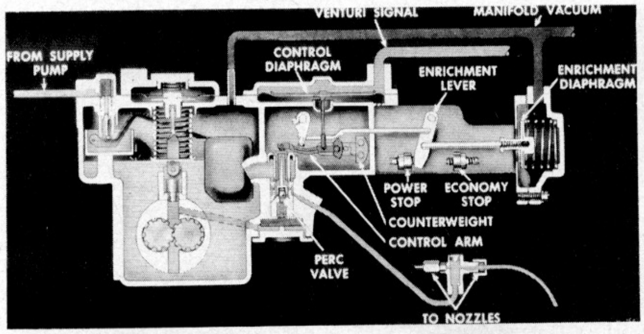


Fig. 23—Fuel meter operating at wide-open throttle

and production considerations to prevent wear and leakage.

1. Gears, covers, and housing require high precision in squareness, roundness, flatness, and concentricity. Surface finishes are of high quality and critical dimensions are held to less than 0.0005-in. total tolerance. The simplicity in shape, where these requirements apply, is of great benefit.

2. Gears and shafts are hardened steel; shaft bearings are a carbon-graphite compound; covers and housing are of cast iron with a rust-preventive coating.

3. Rather than use a highly precise set of dowels, the parts are spaced at assembly by a special oil film applied to the gears. This assures running clearances equal to the oil film between the gears, covers, and gear housing.

### Filters

A supply pump is required to transfer fuel from the gasoline tank to the float valve. This fuel is supplied at about 5 psi and passes through a 10-micron (0.0004-in. particle size) filter. This filter is intended primarily to protect the gear pump and to accumulate foreign matter from the fuel.

A second filter (Fig. 22) is placed between the gear pump and the spill plunger to trap any rare particles which might plug a nozzle. Present design calls for a $^3/_{16} \times 1$-in. diameter capsule of compressed Monel wire, which has the following ideal qualities:

1. It passes no particle above 0.004 in. (100 micron).

2. It passes all smaller particles so as to prevent unnecessary pressure loss with dirt accumulation.

3. It has an infinite number of fuel passages to minimize pressure loss.

4. It will discourage the flow of thread-like materials.

5. It is made of an inert material and is strucurally strong to eliminate deterioration or disintegration of any type.

6. It is sealed in assembly with a diametral and end-compressive load.

### Spill Plunger and Sleeve

These parts perform the dual purpose of controlling the spill-fuel volume and transmitting a force to the control arm proportional to the metering pressure. The parts are made of stainless steel with a Rockwell C hardness in excess of 45. The two parts are lapped to about 0.0006-in. clearance to produce unbinding motion and minimum leakage. The lapped clearance is checked by the time required to leak a fixed air volume under controlled pressure. A sharp-edged spiral groove is cut in the plunger to facilitate the removal of foreign particles trapped between the plunger and the sleeve.

The spill ports in the 0.281-in. ID sleeve consist of 10 radial 0.060-in. diameter holes. With a gear pump of 200% of WOT engine consumption, the spill flow and area at WOT are nominally equal to the nozzle flow and area. This implies that the plunger would uncover enough of 10 (0.060-in. diameter) holes to equal the area of eight nozzle orifices. This is equivalent to less than a 0.005-in. opening, hence 0.005-in. closing travel will double fuel flow. A low travel for the spill plunger and control diaphragm is an obvious advantage because of the minimized inertia and delay in metering response to accelerations.

The spill plunger is hollow to reduce its mass. Actually, a pressure head of 4-in. fuel on the plunger will support its weight. The lowest operating pressure is about 8-in. fuel at idle.

### Control Diaphragm

The control diaphragm (Fig. 23) must have an accurate response to air pressure differentials of 0.01-in. $H_2O$ while having structural strength to withstand safely a 13-lb load at the highest conceivable signal of 55-in. $H_2O$. Both the spill plunger and the diaphragm work over a range where the highest pressure differential with normal operation is in excess of 3000 times the lowest. For acceptable accuracy no fewer than three precision manometers are required merely to measure this range of vacuum.

The same low travel requirement which applies to the spill plunger is essential for the diaphragm, in order to minimize both friction and changes in the diaphragm's 3-in. diameter effective area. This diaphragm is a nearly flat, 0.008-in. thick, Buna-N-coated-nylon material bonded to a single, ribbed, magnesium plate to reduce mass and to increase section modulus.

### Internal Linkage

The design and fabrication of these parts stress low friction, low mass, strength, and static balance. The control arm is an aluminum die casting with a stainless steel roller to minimize fretting on the plunger. It is counterbalanced about its fulcrum with a matched lead ballast so that its mass will not disturb the delicate balance between air and fuel metering forces. For the same reason a second pivot and counterweight supports the control arm, the diaphragm

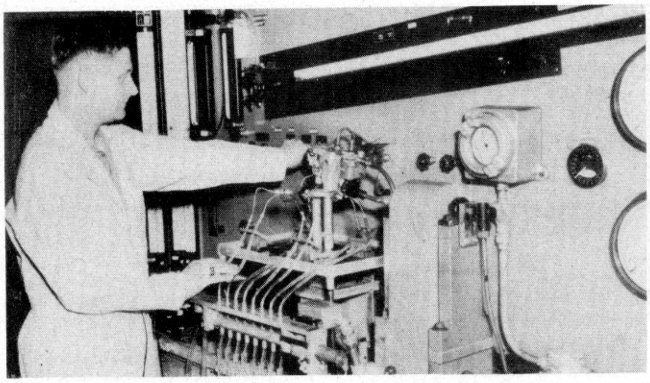

Fig. 24—Fuel meter test setup

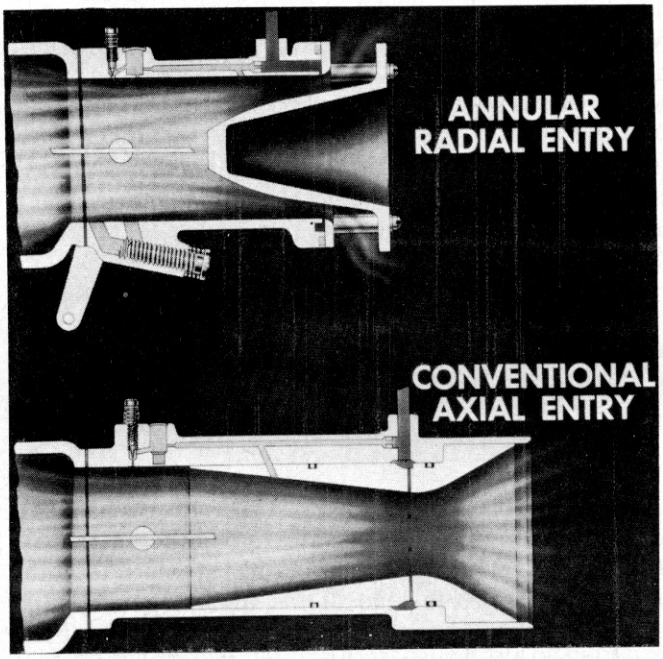


Fig. 25—Air meter design comparison

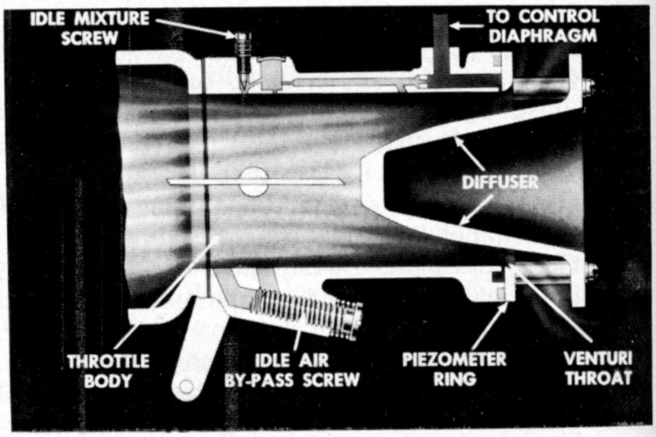


Fig. 26—Air meter at wide-open throttle

link, and the unsupported weight of the diaphragm.

A second reason for counterbalancing is to prevent any motion or vibration from producing an unbalanced inertia force which may disturb the calibrated forces.

### Power and Economy

The mixture ratio is proportional to the mechanical advantage between the air and fuel metering forces. To change from power to economy, the ratio lever and roller move, thus changing the fulcrum point between the two forces. Contrary to the schematic, the enrichment and ratio levers are on a common shaft and are positioned by the spring-biased enrichment diaphragm.

As shown, the manifold vaccum on the enrichment diaphragm is below 5-in. of Hg. indicating a heavy load condition, and the ratio lever is positioned to give power mixtures. If manifold vacuum rises above 7 in. of Hg., the diaphragm spring is overcome to move the enrichment lever to the opposite stop. The ratio lever position now gives the control diaphragm less mechanical advantage, and a leaner economy mixture results from the same venturi signal.

### Spill-Fuel Control

This problem is deceiving although the eventual solution in retrospect seems embarrassingly obvious. The spill must be to bowl pressure, and care must be exercised to baffle the spill to remove velocity energy prior to discharge to the bowl; otherwise, there is "fuel injection" in the float bowl, and loss of splashed fuel to the vents becomes a serious problem. The stamped cap is pressed into the casting and seals at the top of the spill sleeve thus forcing fuel through the schematically-oversimplified baffling-passage. Its cross-section must be large enough to avoid any significant back pressure on the 10 spill holes.

### Vapor Handling

Below the spill sleeve is a spring-load ball valve which acts to keep pump discharge pressure 4 psi higher than the plunger fuel pressure. At idle, vapor formation is prevented upstream of this "perc (anti-percolation) valve." Downstream, vapor may be released and is free to rise and go out as spill fuel. Liquid gasoline moves radially through eight holes to the annulus around the spill sleeve and then to the nozzles.

Directly below the float valve is an open heat-insulated plastic box which catches hot fuel at hot-idle condition thereby permitting vapor escape prior to its overflow to the main bowl where temperatures are lower. The overall design handles vapor quite effectively. Due to the low control diaphragm idle signal (0.01-in. $H_2O$), atmospherically venting the float bowl is not adequate to keep vapor pressure from affecting the metering forces. A restricted manifold vacuum vent connection to the bowl is used to prevent metering disturbance. This also simplifies vapor disposition during normal or hot operation.

An effort has been made to reduce the mass of all components of the system. The heat retention of any part is proportional to its weight; therefore, thin sections are a benefit. This requirement necessitates some compromises with other considerations, but at present all body and cover castings on the air and fuel meters have been made of sand-cast aluminum. Intake manifolds are of fabricated sheet steel or cast aluminum. With more experience and production, changes may be made to die castings of zinc or aluminum.

#### Fuel Meter Calibration

After checking each subassembly as thoroughly as possible, the nozzles and fuel meter are calibrated as a unit. As pointed out, the ratio lever position controls the mixture ratio. At an artificial control diaphragm signal, and an arbitrary pump speed, the enrichment lever stop is adjusted to produce a required fuel flow for maximum power. The enrichment lever is then moved to the economy stop where a second screw is adjusted to give maximum economy. By adjusting the two stops as required, compensation is made for:

1. The matched set of nozzles used (a set of nozzles with a different flow calibration would require readjustment).
2. Variations in effective area of spill plunger and diaphragm.
3. The production variation in unintended fuel passage restrictions and the relative positions of mechanical forces.

Fig. 24 shows the production test stand. After proper adjustment of pump speed and the artificial diaphragm signal, both a high and low fuel flow are checked for total flow and nozzle distribution. Delivery from each nozzle is collected in a burette to verify that the allowable 5% tolerance is met and to prove that flushing out the complete fuel system has not washed dirt into any orifice. No nozzle or meter should thereafter be disassembled unless a subsequent distribution check is made. Routine testing is also performed to test or adjust:

1. All valve seals and opening calibration.
2. Spring calibration or adjustment.
3. Fuel or diaphragm leakage.
4. Fuel metering—hysteresis and repetition.
5. Pump drive torque, and so on.

#### Venturi Principles

The venturi differs from a carburetor in that the maximum suction at the throat relative to the inlet pressure is lower (30-in. $H_2O$ at 24-lb per min airflow). Air expansion with this relatively low pressure drop is negligible, and the incompressible fluid flow equation can be used for both air and fuel flows to make the metering system theoretically sound. The resulting larger venturi also helps volumetric efficiency.

Compactness, low frictional loss, and high venturi metering signal were stressed in the air meter design (Fig. 25). A high signal over loss ratio is a direct function of the venturi L/D ratio (exit taper length/throat diameter). A long exit taper forms an efficient diffuser to reconvert air velocity to static pressure without frictional loss. Originally, a conventional axial flow venturi with 1¾-in. diameter throat was used. It had an L/D of only 2.7 in., while the overall length was 7 in.

The unique radial-entry annular venturi presently used proved better in many respects. D is equal to the annular throat width or 0.300 in., hence the L/D ratio increased to 8.3/1. The length required for just the original venturi was as great as now required by a complete air meter. At 24 lb per min, the loss across the venturi is only 2-in. $H_2O$, while the signal or velocity pressure is 30-in. $H_2O$.

The inlet to the venturi throat is so arranged that no component of air velocity can act upon the piezometer slot (Fig. 26). Vertical location and concentricity of the diffuser fortunately are not too critical; therefore, controlling dimensions have from 0.002–0.004 in. total tolerance. An unexpected critical part proved to be the blending of the inlet radius on the piezometer ring. It appears that streamline variation in the vicinity of the slot cannot be tolerated if the signal is to be held constant.

#### Throttle Valve and Shaft

As with a carburetor, the driver controls the engine through throttle valve opening. A single $\frac{3}{32}$-in. brass throttle valve is used in a 3-in. diameter bore. Because of a 75-lb vacuum load, the shaft is ½-in. diameter stainless steel and runs in bronze bearings. The single bore gives the maximum projected valve opening with minimum circumferential leak area. The 3-in. diameter bore was selected to reduce total air meter frictional loss to about 6-in. $H_2O$ at maximum airflow (assumed at 24 lb per min). For the area of bore, the air meter presents much less restriction and better volumetric efficiency than a conventional carburetor.

#### Idle Control

About 25% of the idle air passes through the nozzle or leaks around the closed throttle valve. The remainder bypasses the valve through an adjustment so as to give warm-engine idle-speed control.

Because idle mixture needs to be richer than road-load economy and because a high-capacity venturi is partially ineffective at low airflow, the venturi signal is artificially strengthened by the application of above-critical vacuum on idle drillings as a function of throttle-valve position. The same critical vacuum across the throttle valve results in airflow varying directly with valve opening and independently of vacuum variation. The resulting fuel flow is directly

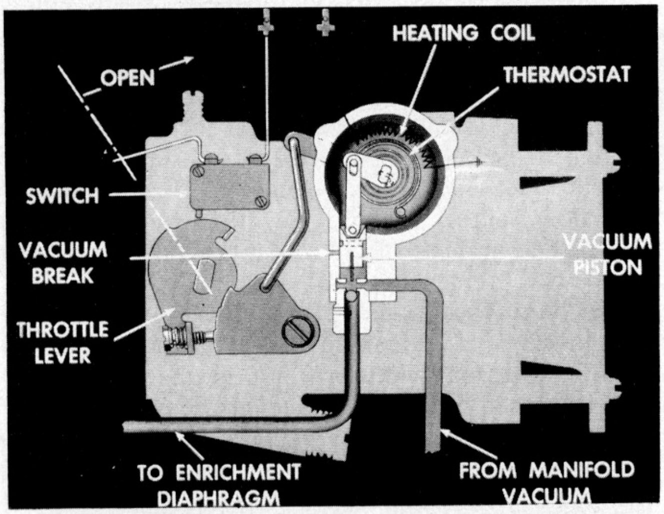


Fig. 27—Air meter calibration for cold operation

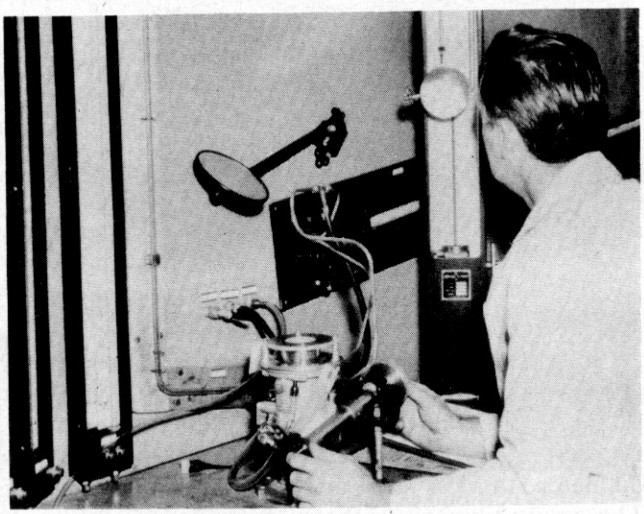

Fig. 28—Air meter test setup

proportional to airflow until the pressure drop is below critical when the venturi is effective.

The lowest idle-speed mixture may be tailored to a specific engine's requirement with an idle mixture adjustment screw. This adjustment has a diminishing effect as the valve is opened. The entire idle mixture tailoring becomes ineffective at about 3 to 4 lb of air per minute.

### Fast-Idle Control

As in a carburetor, (Fig. 27) a thermostat opposed by a vacuum piston senses ambient engine temperature so as to raise a fast-idle cam when the engine is cold. The high step of the cam gives optimum throttle opening for cranking. After the engine first fires, manifold vacuum acts on the piston and overcomes the thermostat. The piston moves to the vacuum break-point thus lowering the cam to the second step. Here, piston vacuum is bled by a port

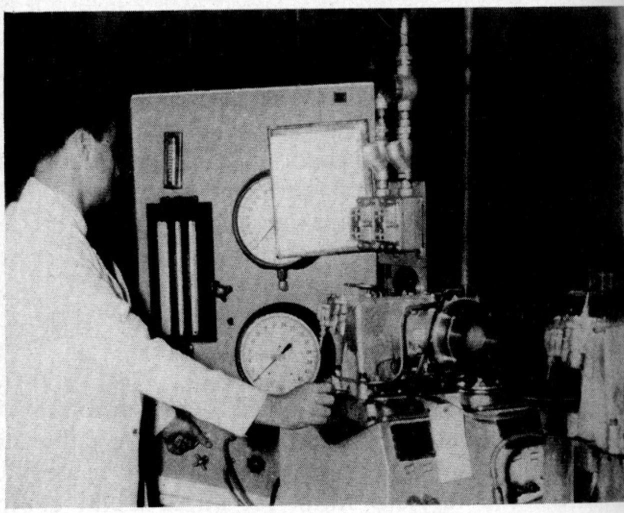

Fig. 29—Final test setup

in the housing, and the throttle now may close to the optimum throttle opening for warmup.

To stimulate warmup, an electric heating coil adjacent to the thermostat is connected to the generator armature terminal and is energized whenever the engine is running. The thermostat progressively relaxes to drop the cam to reduce the idle speed.

### Cold Enrichment

The enrichment required for engine warmup has been defined. The manifold vacuum applied to the power enrichment diaphragm is taken through the cold enrichment housing. The vacuum, normally applied to the thermostat piston, also acts to seat the ball valve below, which blocks diaphragm passage. This produces a richer mixture for cold light-load operation. As ambient plus thermostat-heater temperatures rise, the piston is permitted to move, and the protruding pin unseats the ball. Finally, as cold enrichment becomes unnecessary, full manifold vacuum acts on the enrichment diaphragm to regulate the ratio lever position, and the piston seals its own vacuum supply passage.

### Hot Starting

If the throttle is always partially opened on a hot engine, the solenoid control of cranking mixture produces quick starts. A hot engine cranked at closed throttle may become so overrich that subsequent starting with the throttle open will be delayed. To be positive, a normally-closed throttle switch on the air meter is opened whenever the throttle is more than three-fourths open. This eliminates the solenoid as a fuel enrichment force and leans the overrich engine so it may be started with the same maneuver used on a carbureted engine.

### Air Meter Test

Testing is done basically by passing various measured airflows through the air meter and testing the various output signals. (See Fig. 28.) The characteristics of the venturi are very consistent so that signal test at one airflow is indicative of the overall performance above 5-lb air per min. In the idle range, a more thorough test is needed to measure the air meter signal since it is created by manifold vacuum acting on calibrated idle orifices. As previously stated, readings taken are proportional to the square of ultimate fuel flow so the test is proportionally more accurate than if fuel flows are measured. The mean limits of the air meter are used as adjusted diaphragm signals in the fuel meter calibration. Other adjustments and tests are made to verify the calibration of idle and cold enrichment devices.

### Final Test

After having verified the output signals of the air meter, and fuel-flow output of the fuel meter with adjusted signal inputs, the complete manifold, fuel and air meters, nozzles, and so forth, are built up as they will be shipped to the customer. At this point, a final test as shown in Fig. 29 is run with adjusted airflows through the air system to permit measuring total fuel flows. Nozzle targeting is checked, and all functions are spot-checked to verify the final assembly work and initial calibration. No readjustment should be required at final test.

It is desirable that the manifold become a component part of the fuel injection system chiefly to permit the final test of the fuel meter and nozzles after all assembly work is complete. This may lead to an integration of all the components and to eventual design simplification.

The need for fuel injection has been established. The customer is primarily interested in the advantages of today's fuel injection system but to the engineer, the door is open to a new potential of advancements not possible with the conventional carburetor. In the present and future fuel injection designs, the keynote should be simplicity. This will demand a greater measure of ingenuity to achieve the function but it cannot help but attain the two major goals of every engineer, economy and reliability.

Chevrolet became actively interested in the GM fuel injection system primarily for the Corvette sports car, in order to realize the full potential of the engine for high performance. We had been aware of the theoretical advantages of fuel injection, but were also aware of the relatively high cost. The GM system promised maximum advantage with a minimum of disadvantage, such as cost. As we progressed in our application development of this system to the Chevrolet engine, it became obvious that we should extend its use to our regular passenger cars as well.

Our immediate task was to determine how much of the advantages are attainable, how much of the drawbacks we have to contend with, separate inherent advantages from incidentals, and to evaluate the possibility of attaining comparable results with improvements in conventional carburetion. The areas of evaluation were to cover dynamometer and vehicle operation and embrace all aspects on which carburetion is judged.

The first step was to equip a conventional 4-barrel manifold with an air meter and nozzles, as shown in Fig. 30. Care was taken to avoid a pressure drop

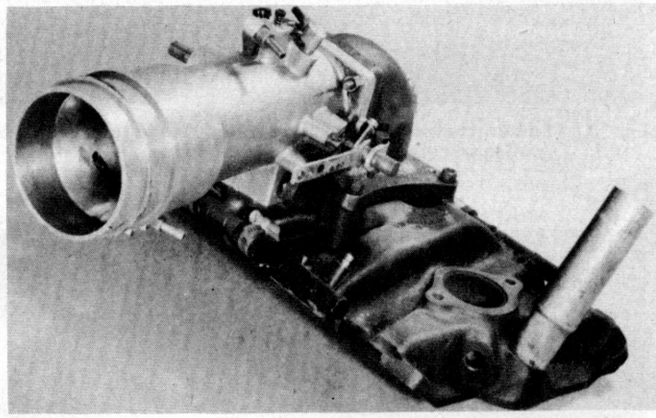

**Fig. 30—Conventional 4-barrel manifold with air meter and nozzles**

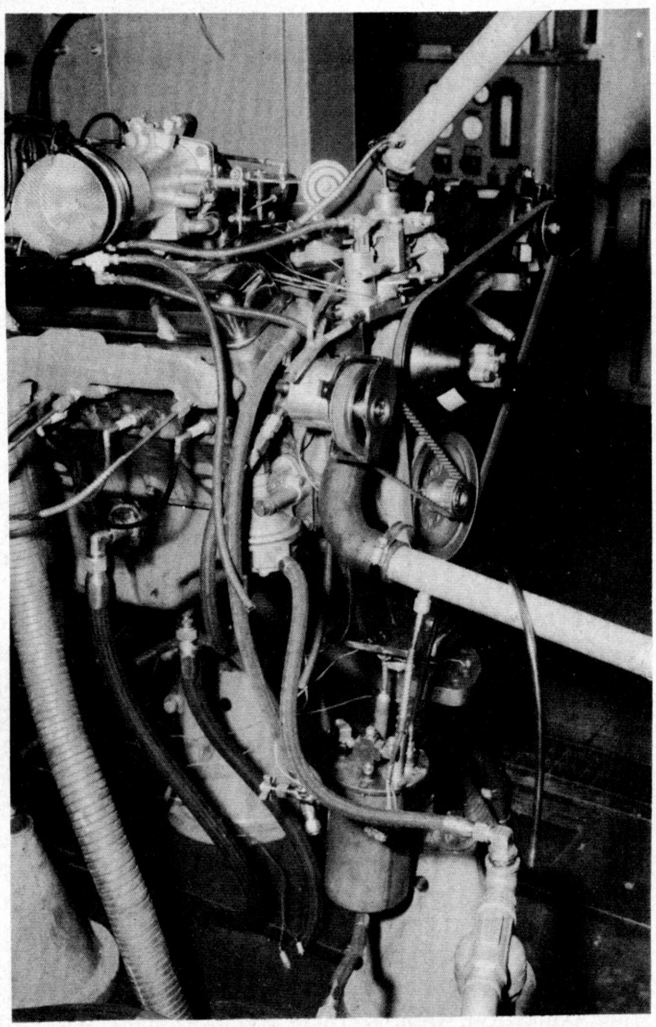

**Fig. 31—Complete installation of fuel injection system**

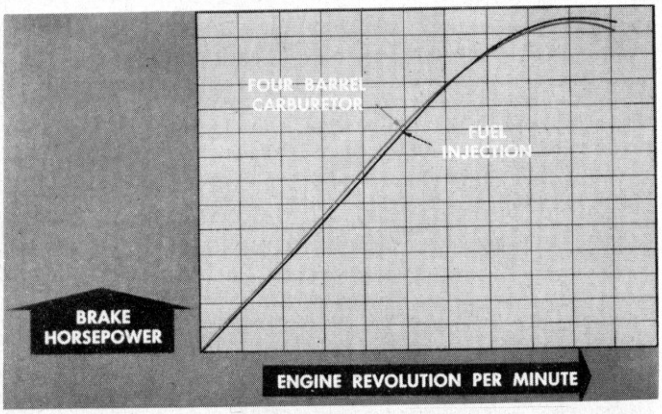


Fig. 32—Power comparison of fuel injection versus 4-barrel carburetor using 4-barrel carburetor manifold

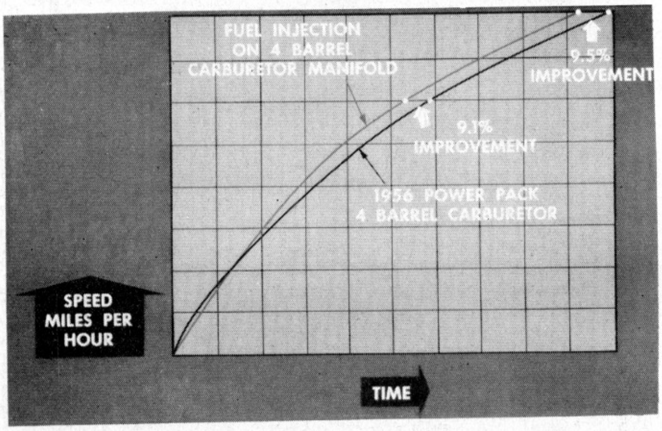


Fig. 33—Acceleration of car equipped with engines of identical power

in the transition from the air meter to manifold, and the nozzle location was carefully selected.

Fig. 31 shows the complete installation with a cobbled up and, at that time, a very experimental injection system.

A power comparison between the 4-barrel carburetor and fuel injection on the 4-barrel manifold was run on the same engine, and the results of comparative tests are shown in Fig. 32. For all intents and purposes they can be termed identical.

The results were not surprising since, within the manufacturing precision at that time, flow variations of the order of 5% were accepted on the nozzles, and the mixture distribution of the well-designed carburetor manifold was well within that range. Also, it should be noted that in this test the carburetor runs were made with heat blocked off, which, in this comparison, nullifies the advantage of the cold manifold inherent to fuel injection. True, the compression ratio increase with fuel injection was not explored in that test, but the spark requirement curves suggested very little possible gain on that score.

The conclusion of the test was that in a steady-state condition, that is, on a dynamometer with air and fuel flow established, fuel injection, as such, did

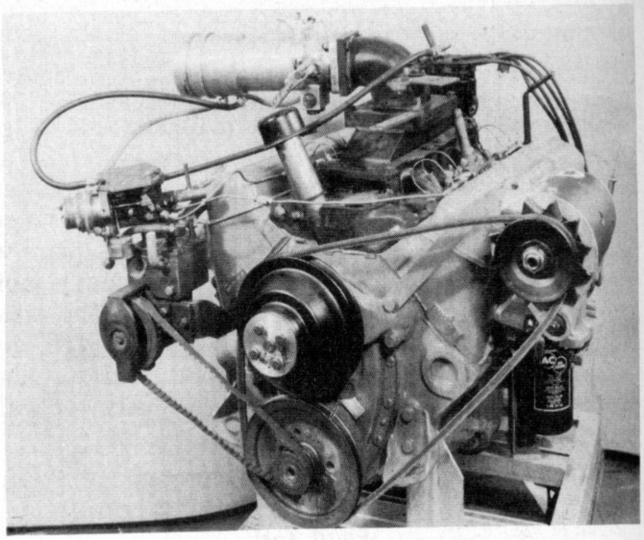

Fig. 34—Two 4 barrel manifolds equipped for fuel injection

not show appreciable superiority over the conventional carburetor.

Having injected and carbureted engines of identical power enabled us to isolate the behavior under transient conditions. The acceleration of a 1956 car was tested alternately with fuel injection and carburetor. The results are shown in Fig. 33. It can be seen that the dynamic condition tells an entirely different story.

The spectacular increase in vehicle performance with otherwise identical dynamometer power can be attributed to the ability of fuel injection to maintain the best power mixture throughout the transition. The mixture composition which the carburetor provides during the transition, relying largely on the acceleration pump, at best could be termed erratic, and the results of the acceleration test brought out the inherent advantage of fuel injection which a carburetor operating with a wet manifold cannot attain. The difference in performance indicated that fuel injection brought out the work potential of an engine which otherwise existed on the test graphs only. We valued this quality since it meant more work per engine pound and dollar.

Our base line high performance engine was equipped with two 4-barrel carburetors. It is a well known fact that engines of that character do anything but justice to their name in the low and medium rpm range and come into their own toward the upper end of the operating speed. By necessity a high-performance engine has a low-velocity manifold, and the problem of mixture formation during the transition is thus aggravated further.

In a manner similar to that used on the 4-barrel manifold, a two 4-barrel manifold was converted for fuel injection as illustrated in Fig. 34. It can also be seen that the fuel injection unit, without reaching a pinnacle of compactness, progressed in that direction, and the maze of injection lines was replaced by two main lines and eight short pipes, as shown in Fig. 35.

The dynamometer tests yielded roughly the same results—that is, similar output was obtained from both carbureted and injected engines.

The injection equipment was transferred to an automobile and the behavior on acceleration was startlingly superior. Not only was acceleration superior throughout the speed range, but the responsiveness even at lower engine speeds was instantaneous and solid.

The most important lesson resulting from these tests was the establishment of the fact that fuel injection offered freedom from compromise in one aspect of induction system design. The eternal compromise between "high end" and "low end" in terms which affect mixture formation does not exist with fuel injection. This is a very valuable quality, not attainable with single or multiple carburetors, meaning, again, more work per engine pound and dollar.

With the inherent advantage of fuel injection as a work producer established, the incidental advantages, such as the possibility of dynamic supercharging by means of appropriate manifold design, could be explored. Length, cross-section, and changes in cross-section of the intake pipes have a marked effect on volumetric efficiency of the cylinder.

Furthermore, the end result is very closely related to the effective flow areas of the intake valves and their timing. Therefore, basic information had to be obtained on the engine itself using telescoping pipes of different cross-sections. Fig. 36 shows the installation with telescoping pipes on cylinders two and seven. The photograph was taken at the end of the run so the pipes look somewhat stubby.

Some of the characteristic curves obtained from these tests are represented in Fig. 37. It is evident that while fuel injection did away with compromises peculiar to the carburetor, it introduced the necessity for compromises all its own.

We can state broadly that long pipes of small cross-section produce high and narrow peaks, and short pipes of large cross-section produce low and broad peaks, and that large cross-sections show lower flow losses at high engine speed. The choice of pipe, apart from space consideration, is largely governed by the desired engine characteristics. Broadly, it can be stated that engines with a narrow range of operation will benefit by a high but narrow ram peak and engines with a wide operating range will avoid such pipes since they impose a compromise on the compression ratio and violently restrict the upper end performance.

It is evident from the compression pressure curve of the long and narrow pipe that, if the compression ratio is chosen to match the peak, a loss of thermal efficiency must be accepted at other operating speeds. If the compression ratio is matched to some median value, then the ignition advance curve must execute an about face near the ram peak only to leap forward again as the density of the charge rapidly thins out after the peak is past.

Fig. 38 shows the comparison between two pipes of convenient length with small and large cross-section which came under consideration for the manifold design.

The engine configuration at that time, that is, the

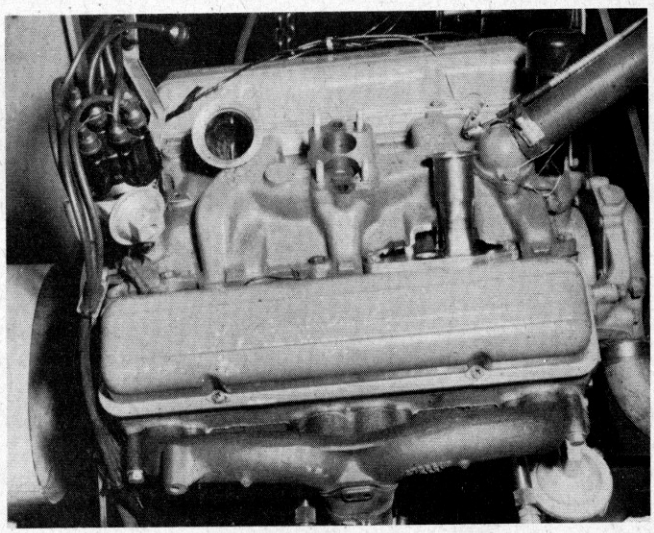

Fig. 36—Telescoping pipe installation on cylinders

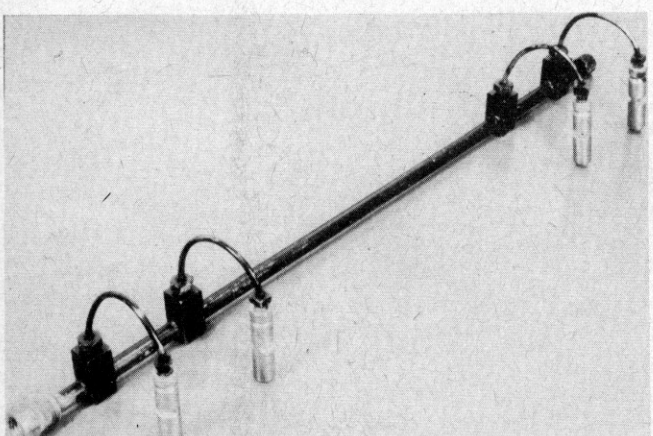

Fig. 35—Compact fuel injection unit

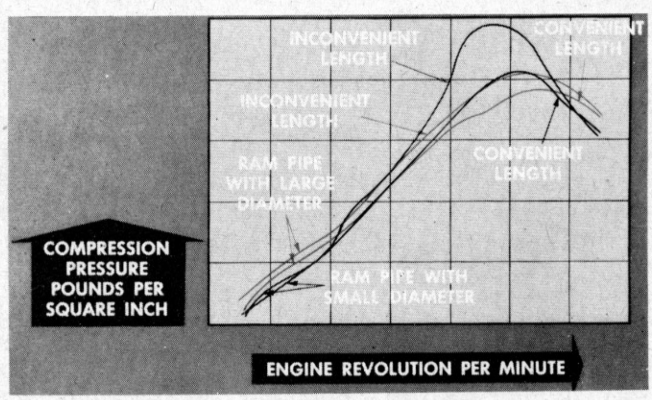


Fig. 37—Motoring compression test on 1955 Corvette engine

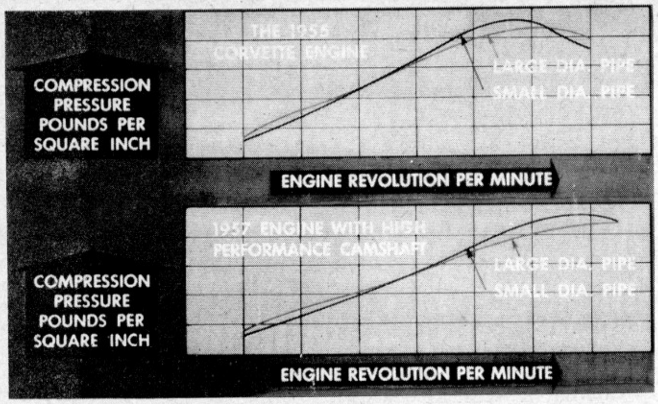


**Fig. 38—Gains and losses of small-diameter pipe versus large-diameter pipe where both are of convenient length**

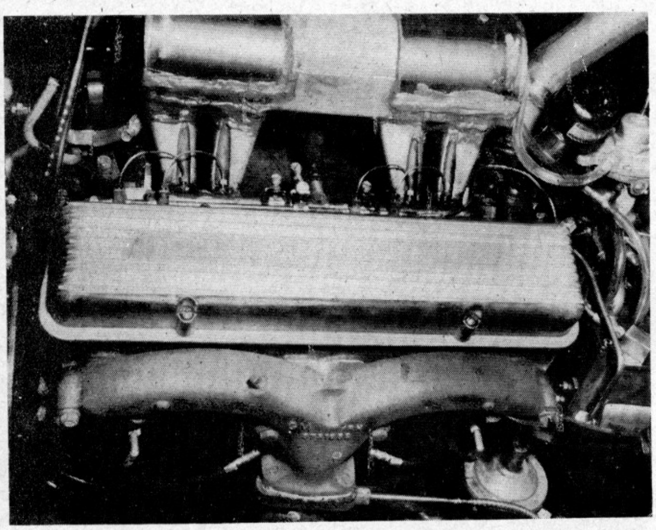

**Fig. 39—Welded manifold**

cylinder head and camshaft and our concern with the upper end of the curve indicated the pipe of large cross-section as the basis of the most desirable manifold design.

Other primary considerations of the injection manifold design are air distribution and air supply. In respect to air distribution, the necessity for a 2-branch manifold, common to 90-deg V-8 engines, does not exist.

All cylinders can aspirate from a common plenum, provided care is taken that the volume from which overlapping cylinders aspirate is adequately supplied. Furthermore, manifolds with a pronounced ram effect demand either a large venturi cross-section so as not to throttle the flow during ram peaks or a considerable plenum volume to provide an air accumulation for individual pipe pulsations.

Fig. 39 illustrates the welded manifold which incorporated the above thinking, and its tapered "sturdy legs" show evidence of the desire to reduce flow losses at some sacrifice of ram. The results were very satisfactory, and a number of cast manifolds of the type in Fig. 40 were produced to satisfy the need for dynamometer and vehicle development units. The early experimental injection system is very well shown in this illustration including an appendix (acceleration pump) which was later amputated as superfluous.

While experiments with hot and cold starts and the behavior of the vehicles went on with the above described manifolds, further experimentation with the ram pipes was carried out.

An improved head design and a new camshaft imposed a different ram pipe requirement, and Fig. 38 shows that, with new configuration, the straight pipe of low cross-section was markedly superior in the intended operating range. Therefore this pipe became the basis of our production manifold design.

Fig. 41 shows the evolution of the manifold from early experimental designs to the present production parts. As can be seen, the first and last designs are similar, being of 2-piece construction. However, while the reason for the two piece construction of the first manifold was convenience, in the case of the production manifold it was a recognition of desirability. The intermediate design taught us the necessity of insulating the nozzles against heat to achieve desired idle characteristics when the engine was hot. The water crossover passage in our 1-piece manifold, and exposure of the bottom of the manifold to hot oil raised the nozzle temperature to the extent that hot idle was limited to a few minutes. The 2-piece design of our production manifold allows us to insulate the upper part from the lower part with its hot water passages and other hot surfaces.

Furthermore, Fig. 42 shows that the metal-to-metal contact of the nozzles with engine parts is prevented by the insulating nozzle holder and by the air space surrounding the nozzle skirt. The cumulative effect of this change is the ability to maintain idle operation even after the engine coolant may have reached the boiling point.

This functional consideration which led to a clean

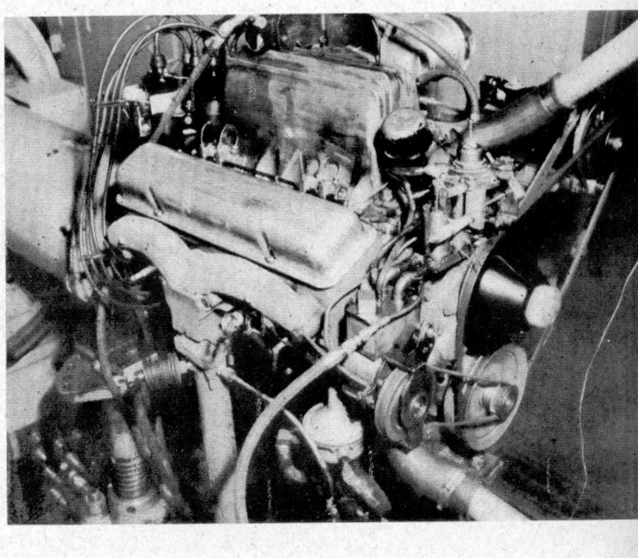

**Fig. 40—Cast manifold**

separation between the engine and fuel injection system has the advantage of assembly and handling convenience. The engine is "sewn up" with the lower part in the same manner as it would be with our conventional manifold, and the upper part, consisting of the induction and injection system, is bolted on as a package—very much like a conventional carburetor.

The nozzle location, in terms of its distance from the cylinder-head intake port, direction of the spray, and its relation to the passing airflow, has significant bearing on dynamic performance, and on the ease of cold starting and warmup. It is evident that the farther the nozzle is removed from the port the more it will resemble the carburetor in dynamic operation.

It appeared, therefore, that nearness to the intake valve is desirable, and the nearest convenient location for the nozzle outlet is at the juncture between the cylinder head and manifold.

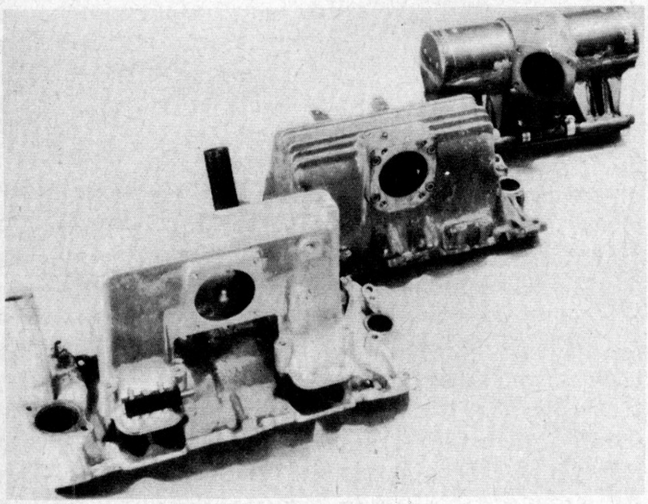

Fig. 41—Evolution of manifold

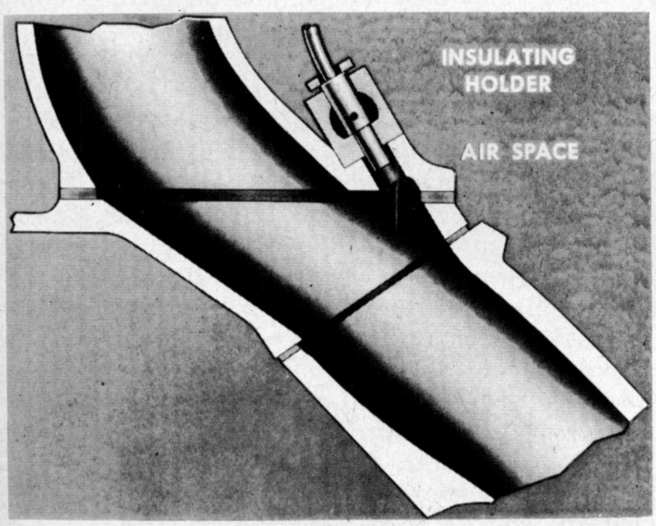


Fig. 42—Chevrolet nozzle installation: nozzle holder and heat insulator

In regard to the aim of the nozzle, the above considerations suggest an aim which avoids the walls and is directed at the intake valve itself. Furthermore it seems desirable to direct the fuel toward the periphery of the valve rather than on the valve stem, particularly in the case of the Chevrolet port layout which imparts moderate rotary motion to the incoming air. The aim and location of the nozzle in respect to the airstream is shown on Fig. 43. The nozzle is placed in a local indentation to combine the necessary intrusion with a minimum protrusion and expose the nozzle skirt to tangential airstreams.

The views expressed in regard to air distribution are fully confirmed by our manifold design. Fig. 44

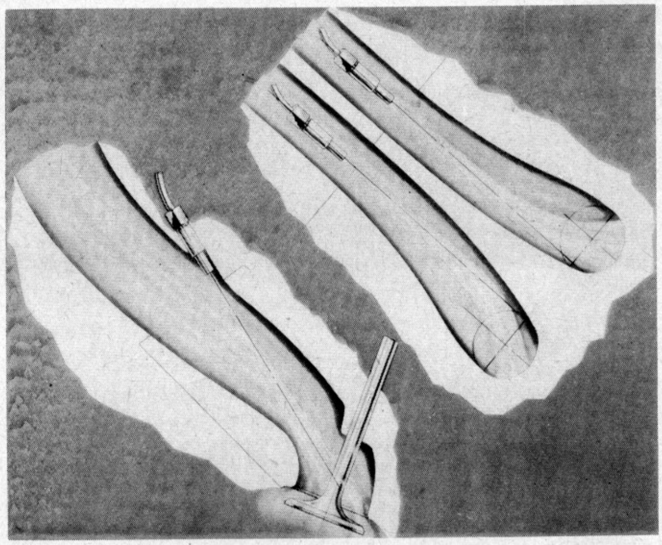

Fig. 43—Nozzle aiming

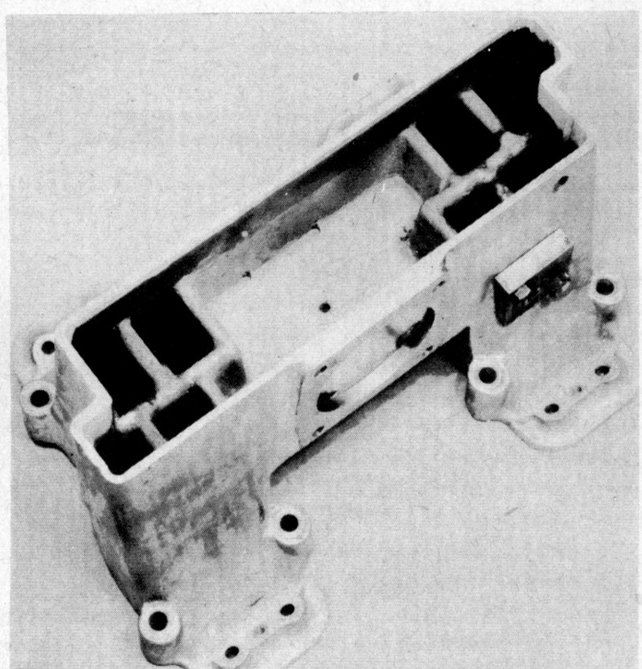

Fig. 44—Cross-section of manifold

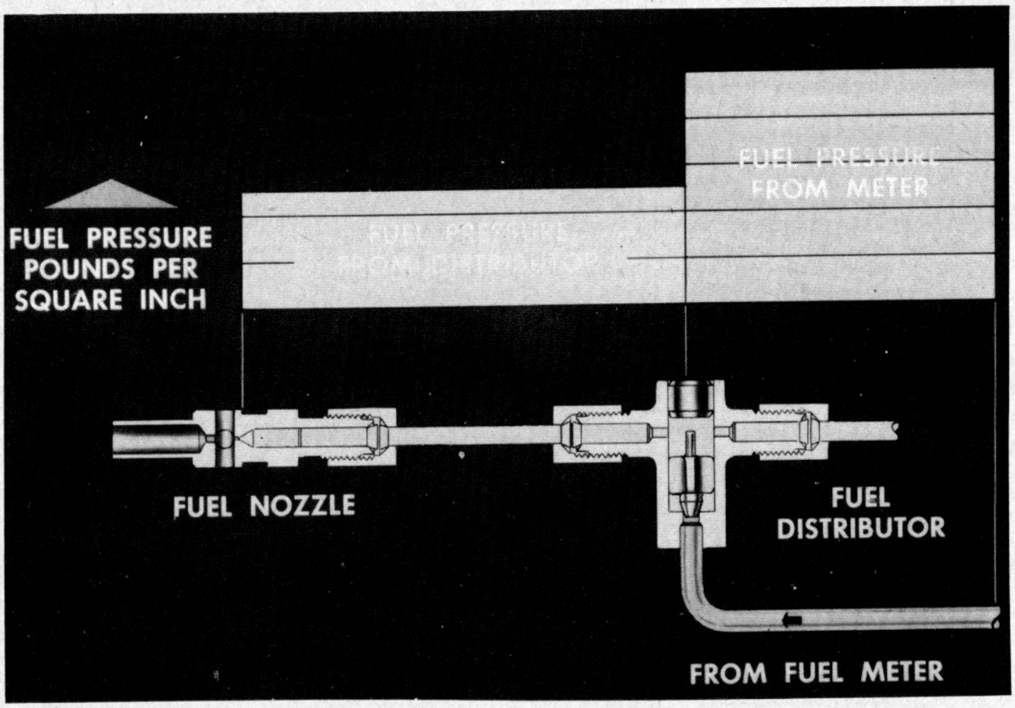


Fig. 45—Reduction of idle fuel rate

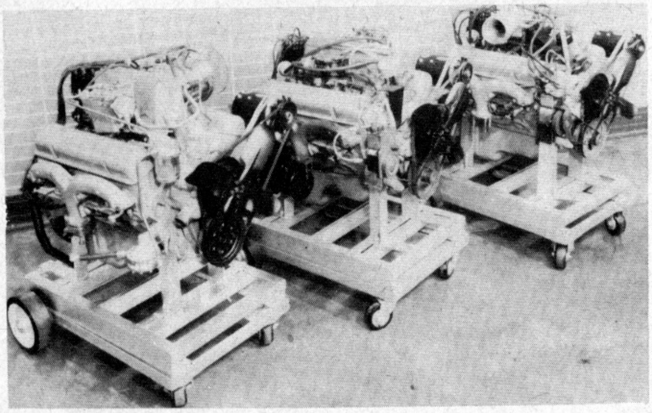

Fig. 46—Evolution of engine with fuel injection

shows the manifold with its top sawed off, and, as can be seen, the individual rampipes aspirate from a common plenum and are grouped without regard to the firing order but according to their geographical location. The showing of an air distribution graph is dispensed with since the identicalness of the values plotted in different colors resulted in an undefinable single blur.

Contemplating the entrails of the manifold, it may occur to the onlooker that the sizes of cross-sectional areas of the air meter and individual pipes are somewhat disproportionate. There are two reasons for this apparent discrepancy.

During the ram peak the pressure waves of the air pulsations in the individual pipe travel at the velocity of sound, whereas the steady flow through the air meter is far from being sonic. Furthermore, the only way to design a Chevrolet is like buying children's clothing—one has to allow for growth.

To increase the range of idle adjustment for all atmospheric conditions and all altitudes, and particularly at the height of Pike's Peak which is so dear to Chevrolet's heart, a fuel rate reduction device was incorporated. Fig. 45 illustrates the method by which a pressure drop is induced at the low fuel pressures. The reduction of rate is a function of balance between the check needle weight in the fuel distributor and its exposed area on the pressure side of the line. At pressure higher than idle fuel pressure the check valve is lifted off its seat and does not interfere with normal metering operations.

Fig. 46 shows some major steps in our development. It can be seen that the metering system hesitantly, but unerringly, moved from the front of the engine to the manifold to form a compact, unified package.

Installed in the vehicle, either Corvette or conventional passenger car, the fuel injection engine provides superior performance, reduced fuel consumption, and instantaneous response to the throttle at all speeds and loads and in all attitudes and states of motion. It enables us to provide a high-performance vehicle with a low speed and low load docility heretofore unattainable. Above all, fuel injection makes usable the work capacity of the engine previously hidden, and with this provides a further step in progress toward efficiency.

This section of the paper has discussed the essential phases of the application development. A great deal of work went into the development of hot and cold running, and above all into the transient responses for all speeds and loads, which do not lend themselves readily to factual and graphic representations.

#### Acknowledgment

The fuel injection system described in this paper represents the coordinated efforts of many people. Mr. Dolza would like to acknowledge the outstanding contribution of members of this group, especially Stephen Kalmar, his assistant, George Ransom, William Kolbe, Stanley Mick, Max Homfeld and John Zimmerman of his staff, and Raymond Heafner now of Rochester Products Division.

Mr. Arkus-Duntov would like to acknowledge the contributions to this project by J. Burnell, D. Bedford, Jr. and F. Frincke, all of Chevrolet, and would like to state that all of this work was carried out in close cooperation and with the unreserved help of the Power Development Group of Mr. Dolza. I would also like to acknowledge the contributions made by E. N. Cole and H. F. Barr, who were instrumental in making fuel injection possible for Chevrolet.

## APPENDIX

#### Airflow

The pressure drop $dp$ at the venturi throat is expressed by

$$dp = K_1 \rho V^2 \tag{1}$$

where:

$K_1$ = Venturi coefficient

$\rho$ = Air density

$V$ = Air velocity

The weight flow rate of air $a$ is given by

$$a = \rho A V \tag{2}$$

where:

$A$ = Throat area

Combining Equations (1) and (2)

$$a = \rho A \sqrt{\frac{dp}{K_1 \rho}} = A \sqrt{\frac{\rho}{K_1}} \sqrt{dp} \tag{3}$$

and if

$$C_1 = A \sqrt{\frac{\rho}{K_1}}$$

$$a = C_1 \sqrt{dp} \tag{4}$$

#### Fuel Flow

The pressure drop $\Delta p$ across the nozzle orifice is expressed by

$$\Delta p = K_2 \delta v^2 \tag{5}$$

where:

$K_2$ = Nozzle orifice coefficient

$\delta$ = Fuel density

$v$ = Fuel velocity

The weight flow rate of fuel $f$ is given by

$$f = \delta W v \tag{6}$$

where:

$W$ = Nozzle orifice area

Combining Equations (5) and (6)

$$f = \delta W \sqrt{\frac{\Delta p}{K_2 \delta}} = W \sqrt{\frac{\delta}{K_2}} \sqrt{\Delta p} \tag{7}$$

and if

$$C_2 = W \sqrt{\frac{\delta}{K_2}}$$

$$f = C_2 \sqrt{\Delta p} \tag{8}$$

The air/fuel ratio is then

$$\frac{a}{f} = \frac{C_1}{C_2} \sqrt{\frac{dp}{\Delta p}} \tag{9}$$

that is, the air/fuel ratio (by weight) is proportional to the square root of the ratio of air pressure drop at the venturi throat (venturi pressure) to fuel pressure drop at the nozzle orifice (fuel pressure).

The metering force $F_a$ on the main control diaphragm is

$$F_a = M\ (dp)$$

where:

$M$ = Diaphragm area

The opposing force $F_f$ on the plunger can be expressed as

$$F_f = N\ (\Delta p)$$

where:

$N$ = Plunger area

For any given air/fuel ratio $F_a = F_f$ and hence for a lever ratio of 1,

$$\frac{M}{N} = \frac{\Delta p}{dp} \tag{10}$$

that is, the ratio of control diaphragm diameter to spill-plunger diameter is equal to the ratio of fuel pressure to venturi pressure.

For the nozzle orifice size selected, the value of the fuel pressure $\Delta p$ at full throttle and top speed may be of the order of 200 psi, while the signal pressure $dp$ under the same conditions may reach about 2 psi.

According to equation (10) the ratio of the area is

$$\frac{M}{N} = \frac{200}{2} = 100$$

#### Discussion . . .

of this paper as well as the following Winkler-Sutton paper is found on pp. 765–768.

fuel injection for 1957. Not much more than a name is viewed here so let's look under the hood. Yes, this is the extent of what you'll see of this year's fuel injection for Pontiac. What is actually viewed here is the inlet air duct, the air cleaner, and a shroud which completely encloses the fuel induction assembly. The shroud provides the air inlet box for filtered air to the venturi and nozzles and at the same time provides cooling for the entire system. This feature, in addition to nozzle block and intake manifold insulators, provided the necessary cooling to enable us to still be idling after 30 min at 100 F. A minimum of 30 min at idle at steady state will result in satisfactory city driving at high ambient temperatures. The shroud has also proven itself to be an excellent shield against dust and dirt from collecting on any part of the injection system.

Inside the shroud, Pontiac's basic system is the same as those described in the foregoing papers; however, our development efforts have been directed toward adapting the GM fuel injection system for conventional passenger-car use. Designing and testing intake manifolds of many configurations having various sizes and length ram tubes, led to our released model which provides substantial torque gains above 2000 engine rpm. This design combined with the GM system has resulted in substantial increases in car performance. A gain in acceleration time from 0–60 mph of more than 10% over our conventional 4-barrel carburetor car was achieved.

While our main goal for this year has been to produce a power-plant having all the advantages that fuel injection can offer for superior performance and smoothness under all car driving conditions, fuel economy has not suffered. We have shown level road fuel economy to be about equal to that obtained on our carburetor cars, and during a cross-country trip, the fuel injection car averaged more than 5% better overall economy than a comparison car equipped with a conventional 4-barrel carburetor.

Our performance development has been carried on quite extensively at both the hot and cold extremes of weather conditions. Running in the hot climates has proven our fuel injection system to be resistant to vapor locking tendencies even with high vapor pressure fuels. High altitude running also presented no particular problems. Cold weather development, however, particularly in the range of 0 F has entailed considerably more development.

It can be said that fuel injection offers the engineer a tool with great possibilities for refinement. Our future goal for fuel injection will be to maintain our high standard of performance while simplifying the system with a resultant reduction in cost.

## Discusses General Motors and Bendix Fuel Injection Systems

**C. G. Nystrom**
American Bosch Arma Corp.

THE excellent papers presented by the authors on the subject of gasoline injection is of interest to a great number of people today and of particular interest to the American Bosch Arma Corp. because we, too, have a gasoline injection system of which we are quite proud.

#### Basic Development

The basic development of the ramjet constant-flow system as presented by Mr. Dolza of GMC, has some very interesting features of note, and both he and his associates should be complimented for their success in this development.

During the development of the American Bosch metered gasoline injection system, we studied various carburetion systems, including conventional carburetors, pressure carburetors, and multinozzle constant-flow arrangements. Among the latter was one basically similar to the system described by Mr. Dolza which was both dynamometer and road tested. In our analysis of all the carburetor and injection systems considered, we concluded that the timed-metered injection system offered the greatest potenial and the most advtantages, not only for the application on existing engines, but in the application to engines of both two and four cycles designed specifically to take full advantage of the gasoline injection.

Messrs. Kehoe and Stoltman, along with their associates should be congratulated on this presentation of the production development of ramjet constant-flow system. It is interesting to note the great detail used in describing the various component parts and the accuracy needed during manufacture, as well as the numerous tests of the component parts of the system, and the final calibration test of the entire system including the intake manifold. The only conclusion I could arrive at was that this type of fuel system has inherent features indicating a high manufacture cost.

#### Application Development

I will not comment on this phase of the development because we are accessory manufacturers and have to rely on the various engine manufacturers to design the intake manifolds and cylinder heads to be able to take the full advantage of our injection system.

#### Electrojector

Messrs. Winkler and Sutton, Fuel Systems Engineering Department of the Eclipse Machine Division, Bendix Corp., have made an excellent presentation of this paper on the Electrojector fuel injection system. This system, incidentally, is a timed system as you know, and basically we are in agreement with this idea. The system, which has been described in the authors' paper, has a number of very interesting features. The electronic modulator or "brain box" for the control is a new and attractive approach which is said to have solved the many complex control problems such as optimum air/fuel ratio over the entire speed and load range as well as starting, idling, load, and acceleration enrichment required during warmup and the smooth transition to the requirements of the warm engine. In addition to these requirements, there is the automatic altitude and temperature compensation and the all-important fuel cutoff during deceleration.

The electronic spray valve for metering the fuel is a rather old idea and was developed by Mr. Kennedy of the Atlas Imperial Diesel Engine Co. in 1932. An installation was made on a 6-cyl low-compression spark-ignition oil engine for marine service. This engine was exhibited at the New York Motor Boat Show in January, 1933. In 1934, a smaller oil engine was installed in a truck and was driven from Los Angeles to New York and returned and proved very successful. The description of the Atlas system is published in *Diesel Power*, February, 1933, and *Automotive Industries,* March 4, 1933. The Atlas system lacked proper automatic control. Of course, the knowledge of electronics was very limited 24 years ago, and I don't believe that the transistor was even conceived. It is in this field that Bendix has made the greatest strides, and we should give credit to the author and his associates for the unique adaption of a basically sound injection principle.

## Comments on Both Fuel Injection Systems

**—M. J. Kittler**
Holley Carburetor Co.

BOTH the General Motors paper and the Bendix Aviation paper on their respective fuel injection systems were of great interest. General Motors is to be congratulated on being the first American manufacturer to offer a fuel injection system to the general public on their cars. Bendix Aviation is to be commended for introducing an electronic system as a part of the control mechanism for an automotive fuel injection system. The conception of the electronic control of metering introduces a whole new dimension to the handling of the basic problem of fuel metering control. The future development of electronic controls for fuel metering systems should prove to be very interesting.

As a general comment on both papers, I was rather pleased to note that the claimed improvements with respect to power and economy were held to fairly reasonable limits. It is my opinion that a great deal of early publicity in both the technical press and the general press was highly tinged with optimism, which cannot help but result in disappointment when the actual results are appraised. Gains of the order of 5 to 10% in power and economy appear attainable with a good fuel injection system. Gains in excess of this would arouse some suspicion in my mind at least. It may be well to mention here that it is only fair that fuel injection power data be compared to carburetor data attained with the use of a dual 4-barrel carburetor installation.

I was also pleased to note that fuel/air ratio indicated in the papers, and also distribution variations indicated were in a region not unfamiliar to those acquainted with good carburetion practice. Distribution accuracy between cylinders of 5% total spread, or better, is not easy to obtain, regardless of what kind of a fuel-feed system is used. In the case of fuel/air ratios there is no readily apparent reason why fuel/air ratios required with fuel injection should be appreciably different than those required with carburetors, assuming that a similar range of mixture distribution variations is attained in both installations.

There was a comment relative to speed-density metering versus mass airflow metering. Carburetors operate on the principle of mass airflow metering and the General Motors system, following this same principle, is subject to similar limitations. If the basic venturi size is adequate for maximum airflow at minmum pressure drop, then the metering forces available at low speeds will be exceedingly minute. For this reason we feel that speed-density methods offer better control of metering force than mass airflow methods.

The construction indicated for the fuel nozzles in the General Motors system raises the suspicion that these nozzles may be subject to icing under adverse conditions. These nozzles are carefully insulated to prevent heat absorption, and yet they are, in fact, expansion nozzles and will therefore produce refrigeration. I would like to ask if this condition has been investigated.

The emphasis on the use of rampipes at the intake ports carries overtones of racing engine thinking. As has been so ably pointed out, rampipes are tuned to certain speeds and are only of benefit under wide-open throttle conditions. Whether or not rampipes are used is of no consequence throughout the broad range of ordinary traffic and highway driving and ordinary passenger-car operation. Another obvious disadvantage of rampipes is their very length itself, which adds directly to the overall height of the engine. To be effective, the rampipes must have length, and when they have length they add unwanted dimensions to the engine envelope.

Referring to the Bendix Electrojector system, there are one or two theoretical questions that come to mind.

The first question has to do with the matter of reliability. It has been generally accepted that a hydromechanical mechanism possesses a greater degree of reliability than a parallel electronic mechanism. Undoubtedly great strides have been made in improving the reliability of electronic mechanisms, but I would like to ask what Bendix's experience has been in this area.

The second question has to do with the inertia—mechanical, electrical, and hydraulic—of the fuel and the solenoid valves at high engine speeds. It would appear that inertia effects could become quite pronounced at engine speeds upwards of 4000 rpm.

Since the concept of using an electric brain box for control of fuel metering is so new, it is difficult to envision how easy it would be to change the calibration curves to suit the requirements of different engines. Engine requirements vary widely, and I would be interested in hearing a discussion of the method used to calibrate the "brain box" to meet the engine needs.

It may well be that these two very interesting papers will mark the start of an extended period of evolutionary development of fuel injection systems for motor vehicle engines.

## Questions Aspects of Fuel Injection Systems

**—E. R. Mason**
Chrysler Corp.

THESE comments refer to General Motors fuel injection system. We agree with Mr. Dolza's conclusions that continuous-flow port injection is the most logical and economical system for volume passenger-car production. We also agree that by utilizing a system of this type engine performance is in no way compromised when compared with timed injection of either the port or direct cylinder type.

Before discussing the mass flow system I would like to call attention to the statement that speed-density metering places exacting requirements upon the fuel supply pump. It is stated that the speed-density system requires a pump with constant delivery characteristics. With this we must disagree. The end result of speed-density metering is a controlled fuel delivery nozzle pressure. The speed-density fuel control may be regarded as a pump pressure regulator. So long as the pump is capable of delivering quantities in excess of engine requirements and at pressures in excess of nozzle requirements no other specification need be imposed.

### Mass Flow Control

The General Motors arrangement is interesting and obviously is a carefully planned endeavor to develop a practical automotive system. However, the schematic of this system leads to a comment regarding the fundamental approach which was employed. The basic attraction of a mass flow system for automotive use has been that it required no engine-driven components. This feature has been weighed by many against the obvious drawback of mass flow metering. That drawback is lack of adequate control signal at idle and low engine outputs. This performance region is of extreme importance in passenger-car service. In the case of the General Motors arrangement it appears that benefit has not been taken of the most attractive feature of the mass flow system, namely, by incorporation of an engine driven pump.

Fig. 11 shows mixture ratios and their means of attainment in the low output region. It is interesting to note that control of starting, idle, and road load power requirements is not covered by the mass flow system through 50 mph. Throttle ports and differential vacuum signal manipulations provide the basis for fuel metering up to about 220-lb of airflow per hour. A third comment regarding the mass flow system is its well-known tendency to be influenced by pulsating airflow. Can eight individual ram tubes be gathered into a single volume without a resulting pulsation which interferes with desired metering characteristics of the venturi?

### Vapor Handling

Throughout, much attention has been given to vapor handling characteristics. This attention we believe is well placed. Vapor handling and control of vapor formation is one of the most difficult and important features of an automotive fuel injection system. I believe many people will agree with me that the most difficult obstacle to development of a satisfactory passenger-car injection system is proper control of vapor. In Mr. Dolza's discussion of air density compensation, he states that proper utilization of fuel vapor bubbles at high temperature is used to cancel adverse air density effects. If this be true, we have met the master; for Mr. Dolza has trained his bubbles better than we have. When studying the vapor handling characteristics of the schematics, I looked in vain for a fuel bowl vent. Fig. 23, however, shows a manifold vacuum vent. The discussion indicates that this vent is restricted to avoid metering disturbances. Inasmuch as it has been indicated that a variation of 0.01 in. of water will affect metering, I do not understand how a controlled vent can accurately maintain bowl pressure anywhere near such limits. If it can, is it possible to handle the large vapor quantities which may result from coasting bypass or normal recirculation at high ambient temperature in so confined a volume as a fuel float bowl? It would seem that during periods of high vapor generation, this vapor augments the normal idle flow and under such conditions can an acceptable idle be maintained.

Vapor lock is mainly a problem of getting fuel from the tank to the fuel bowl. By utilizing an engine-driven diaphragm supply pump it would appear that this system has not solved the generally recognized vapor lock problem.

### Nozzles

With regard to the injection nozzle construction and installation on the manifold, considerable importance is placed upon means of using evaporative cooling to control temperature of the fuel delivery line and nozzle as a means of controlling vapor formation. Our experience with low-pressure continuous flow systems has never given any indication that fuel-line temperature was of any importance. We have found that fuel temperature and vapor formation must be meticulously controlled until the metered fuel is delivered to the nozzle line. In the case of this system we wonder if an adverse feature has not been incorporated. It would appear that the refrigeration effect would be highly detrimental to engine operation under atmospheric conditions conducive to ice formation in conventional carburetors.

We wonder if the cold start, warmup, and hot-cranking fuel control has been found to be satisfactory. Our experience indicates that the hot-cranking control and the cold start and warmup device must be interrelated to cover all temperature requirements. Normal hot cranking requires a fuel flow of about 10 lb per hr. Cold cranking may require up to 35 lb per hr. To cover this range of fuel flow with temperature we believe the two devices must act together. In connection with this subject, we wonder if an electrically heated thermal warmup unit can ever be related to any significant engine temperature during either warmup or cooldown. We might also question whether elevation of the cruise fuel/air ratio to power mixture is sufficient to cover warmup requirements.

A last comment on operation of the system would be to question the conclusion that no accelerator pump is necessary. We have found that when the leanest possible road load mixtures have been reached, enrichment is required for solid engine operation during transient accelerating conditions.

## Authors' Closure To Discussion

MR. KITTLER has pointed to the fact that gains in power and economy are shown only in the order of 5 to 10%. Perhaps we have been too humble in this respect. Most of the improvements shown are on a steady-state basis with carburetor heat blocked off, while actually we find our largest gains on a transient basis. Gains on the order of 15% are not uncommon for both power and economy. Mr. Kittler further suggests that we make all fuel injection comparisons to the dual, 4-barrel carburetor arrangement. This we would do happily because here is where fuel injection really shines. Our fuel injection system not only shows performance gains in the high-speed range but far surpasses the dual, 4-barrel carburetors in performance at low speeds from the standpoints of power, smoothness, and economy.

Mr. Kittler's remarks concerning mass flow metering limitations are correct as applied to carburetors where it is necessary to use the venturi depression directly as the pressure drop to obtain fuel flow. However, in our fuel injection system the venturi depression is accurately multiplied mechanically to obtain a much greater fuel pressure range. We have found mass flow metering to be the most satisfactory system.

Nozzle icing is not as serious a problem as it first appears to be. The refrigeration from the fuel, fortunately, is very effective at high ambient and fuel temperatures and is very small at low temperatures since it is directly related to the vapor pressure of the fuel. Our nozzle is designed to produce about 10-deg cooling at high temperatures and the small amount of refrigeration at 30 to 40 F ambient causes only a small amount of ice under certain conditions. This ice consists of a thin coating in the large air holes and causes a slight amount of fuel enrichment during warmup but disappears as the underhood temperature increases.

Regarding Mr. Kittler's remarks on rampipes, they are very effective in the 40- to 65-mph range of driving especially with automatic transmissions that shift down and permit engine speeds in the peak ram range.

Mr. Nystrom reports that timed injection offers the most advantages on both 2- and 4-cycle engines. We have found that continuous injection can be made for less cost without sacrifice of power and economy. It is true that timed injection will be best for 2-cycle engines, but we believe that the 4-cycle engines should not be forced to use timed injection and suffer cost-wise because of the 2-cycle engines.

Mr. Nystrom raises the question of nozzle clogging. Our experience during development is that dirt can be removed before assembly and the system will stay clean thereafter.

We are happy to learn that Mr. Mason has come to the same conclusions as we regarding types of injection; that continuous-flow port injection is the most economical system with no sacrifice in engine performance.

Our statement regarding the speed-density metering fuel pump was that it "usually requires a supply of fuel in equal amounts per cycle." We realize that there are some exceptions. For example, the engine speed can be sensed electrically, in which case the pump need not be uniform. Perhaps Mr. Mason is referring to a density metering system with no speed sensor.

It is possible with the GM fuel injection system to have the pump driven separately from the engine. However, we believe the direct engine drive is the lowest in cost and the most reliable.

In our mass flow metering system, we use manifold vacuum bleeds to obtain the enrichment required at idle and off-idle. Fig. 11 is an artist's attempt to illustrate the method rather than give exact data. Since the scales do not start at zero, the effect at first appearance seems larger than actually shown; for example at idle, where the bleeds make the largest signal modifications, the effect shown is less than 20%. Actually during our experimental work, we have had some units which required no idle bleed and some required no off-idle bleed.

We have found no disturbance to metering caused by the rampipe pulsations. In fact, we had one experimental installation with only four cylinders and a much smaller plenum volume between the air meter and rampipes, and metering was satisfactory at all engine speeds.

The control diaphragm in the GM fuel injection system is placed between two chambers: the upper chamber is subjected to the metering depression signal, and lower chamber is vented to the static pressure at the venturi inlet. In this manner, the diaphragm is affected by metering signal only and is not disturbed by vapor pressure or manifold vacuum in the float chamber. The lower diaphragm chamber is formed by using a very small hole around the diaphragm link and a large vent to the venturi inlet.

The fuel vapors are then vented from the float chamber to the manifold. Using "educated bubbles," we can alter the effect of the venturi signal so as to lean the mixture at high temperatures.

We have found that most of the vapor bubbles are formed when the fuel is admitted through the float valve; that is, when the pressure is reduced from about 6 psi to atmospheric pressure. During coasting, when the fuel is shut off and no new fuel admitted, there is very little vapor formation.

On hot-idle tests the GM fuel injection system has continued to idle indefinitely where otherwise identical carbureted cars have boiled and were stopped. In these tests, it was interesting ot note that carbureted cars boiled much more readily than fuel injected cars indicating lower heat transfer to the cooling system. As a result, we have found vapor lock much less severe with fuel injection even though we use the same diaphragm pump.

Engines equipped with GM fuel injection system have been found to be much less sensitive to flooding during cranking than with carburetors. The hot and cold starting methods described have been found to be satisfactory on a great number of such tests.

The warmup thermostat is primarily heated electrically by a heating coil energized through the generator armature. At idle when the engine warmup is slow the generator voltage is low, and there is less heat supplied to the thermostat. Also because of its location, the thermostat is somewhat sensitive to ambient air temperaure and engine temperature. This system has proved satisfactory on hundreds of warmup tests.

For warmup enrichment, we have found the air/fuel ratio obtained at the power mixture stop to be sufficient for baseline temperatures as low as $-10$ F. At lower temperature, a few seconds engine warmup may be required before complete flexibility of the engine can be attained, and this should not be objectionable for the sake of engine wear.

Concerning Mr. Mason's skepticism about our omission of the accelerator pump, we would like to comment that we do not set the control for "the leanest possible road load mixtures." We set our economy stop for the mixture which requires the least amount of fuel for road load. This best economy mixture is considerably richer than the leanest possible mixture at which the engine will run. At this best economy mixture and with the proper attention to details as explained in the paper, we do not need the accelerator pump.

Mr. Kittler has questioned the reliability of electronics. We have had no operational failures during our tests. We have full confidence in electronic components. Also, Mr. Kittler questioned the inertia effects on fuel lines. We have had some trouble, and this was our basic reason for changing to the common rail system.