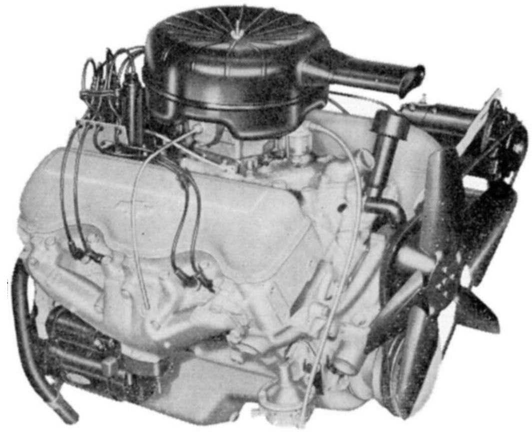
Engineering the "W" Engine Chevrolet's 348-cu-in. V-8



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WITH the 1958 model, Chevrolet introduced a new 348-cu-in. V-8 engine. This engine — commonly referred to as the "W" — broadened the range of engine sizes, so that now Chevrolet makes available a 235-cu-in. in-line 6, a 283-cu-in. V-8, and the 348-cu-in. V-8 in all its passenger cars.

Conception and Design

In order to meet customer demand for the V-8 engine, it became necessary in 1955 to plan for an increase in our production facilities. Because of the rising demand for automatic transmissions in our deluxe line of cars, it was also necessary to look forward to increasing the engine displacement to provide optimum low-speed and mid-range vehicle performance. A larger engine with good low-speed torque, we felt, would make an exceptionally good teammate for the Turboglide transmission, then in the development stage. During the past ten years we all have seen numerous V-8 engines put into production. With very few exceptions, these have been single purpose engines, patterned very much like the V-8 engine first produced by Cadillac in 1949. During this decade we have seen many of these engines redesigned and retooled at great expense, because it became necessary to increase displacement to provide the performance necessary for safe and pleasant motoring.

Primary Objectives — Because of the importance of multiple usage to insure high volume production and to insure maximum usage of automated manufacturing equipment, we recognized the need for establishing a new set of ground rules and conceiving a design which would meet them. We decided that the new design must permit:

1. Adaptability to a broad range of displacement with a minimum number of different parts.
2. Adaptability to broad compression ratio range to match the octane trend of future fuels.
3. Dimensions compatible with the anticipated space limitations of passenger-car design.
4. Provisions for mounting accessories on engines for both passenger cars and trucks.
5. Flexibility of machine tools to accommodate future engine modifications.

Basic Dimensions — Fulfilling the first requirement, adaptability to a broad range of displacement, was a rather straightforward job. After establishing the bore and stroke to provide a displacement in line with passenger-car requirements, we deter-

SPECIFICATIONS of the new Chevrolet Turbo-Thrust V-8 engine described in this paper are:

Type	90-deg V-8
Valve Arrangement	In head
Bore	4.125 in.
Stroke	3.25 in.
Stroke-to-Bore Ratio	0.79/1
Displacement	348 cu in.
Compression Ratio	9.5/1 (passenger car)
Carburetor	Single 4-barrel
Maximum Gross Horsepower	250 at 4400 rpm
Maximum Gross Torque, lb-ft	355 at 2800 rpm
Maximum Bmep	152.3 psi at 2800 rpm

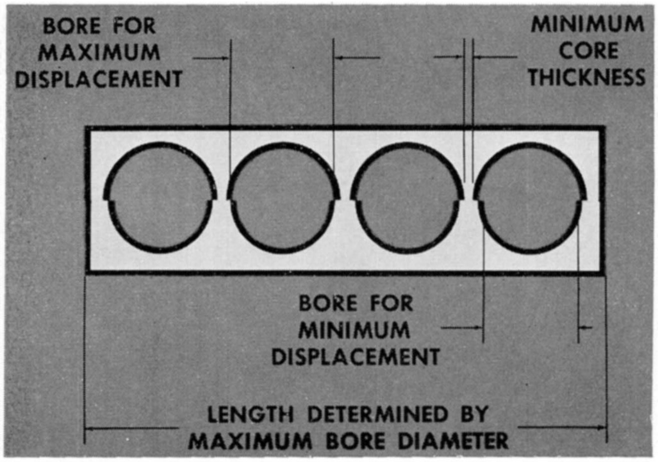

Fig. 1 — Determining basic dimensions of engine from cylinder bore size

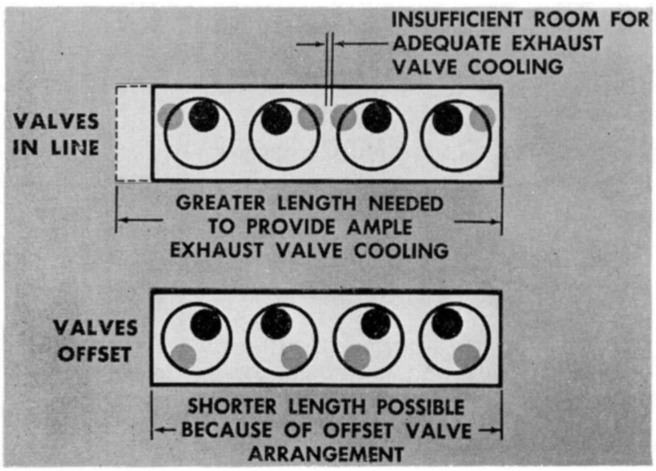

Fig. 2 — Determining basic dimensions of engine from valve arrangement

mined the bore size that would satisfy our maximum displacement truck engine requirements. The largest bore size then established the bore centers that would provide full circumference cooling and minimum core thickness between bores which would be acceptable to the foundry (Fig. 1).

Preliminary Studies — Starting in 1955, we made numerous design studies of engines having greater displacement than our then new 265-cu-in. V-8 engine. We built and tested two of these engines of approximately 300-cu-in. displacement, with the same basic external dimensions and configuration as the 265-cu-in. V-8. In one engine, increased displacement was obtained by increasing the bore to 4 in. while retaining the 3-in. stroke. This necessitated joining the bores, creating a difficult casting problem and preventing complete coolant circulation around the cylinder.

In the other engine, the stroke was increased from 3 to 3.3 in. and the bore from 3¾ to 3 13/16 in. In addition to necessitating completely new tools and equipment for crankshaft machining, both these designs would have severely limited further displacement and compression ratio increases that might be required in the future.

In 1956 the ratio of V-8 to in-line 6 engines at Chevrolet was increasing rapidly due to customer acceptance of the V-8. Management was confronted with the necessity of buying more lines for V-8 engine production by model year 1958.

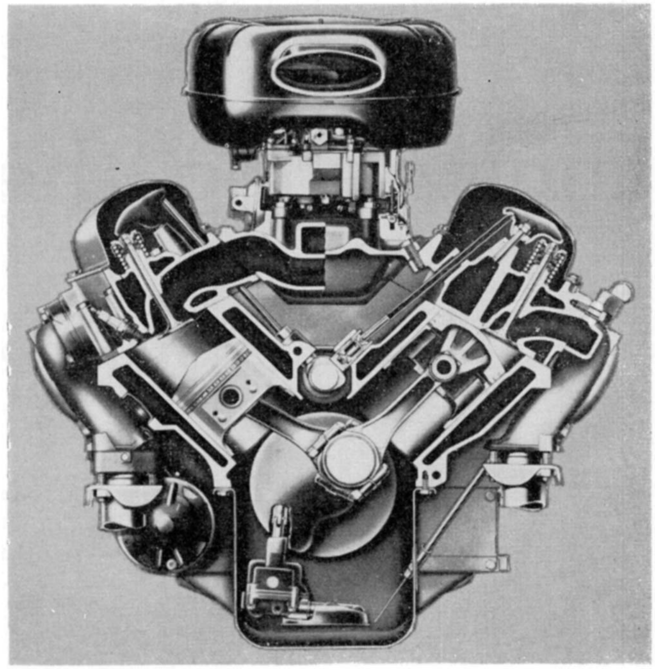

Fig. 3 — Cross-section of engine

The cost of new capital equipment would not be greatly increased by providing the added lines with a design which could add materially to the displacement range for passenger cars. The same basic engine could be used for the higher gvw truck models which Chevrolet is now building. As a result of these considerations, the proposal to develop an entirely new engine was approved.

Valve Arrangement — The basic length could then be maintained, provided that room was available for valves of the size needed for the maximum displacement engine, with spacing adequate for good exhaust valve cooling and freedom from valve seat distortion. Placing the valves in line would increase the length of the engine in order to provide sufficient space between valves. Staggering the valves, on the other hand, would allow for the necessary space while still maintaining the minimum overall engine length (Fig. 2). Advantageous location of the valves was relatively easy to achieve by means of individually mounted rocker arm mechanism that has been used successfully by Chevrolet and Pontiac since 1955. Flexibility of this design also permitted the use of a common rocker arm for all the valves.

The required piston proportions, counterweight radius, and connecting rod length established the basic height of the cylinder block (Fig. 3).

The established engine length permitted excellent proportions for bearing length and cheek thickness (Fig. 4). These features were combined with large overlapping journals to produce a stiff crankshaft.

Combustion Chamber — In order to meet the second objective, provision for a range of compression ratios, considerable design effort was expended Many manufacturing problems had to be investigated. It was obvious that with the combustion chamber placed in the cylinder head, the foundry must retool every time a compression ratio change is in order. The necessity of making special heads to provide a range of compression ratios and to per-

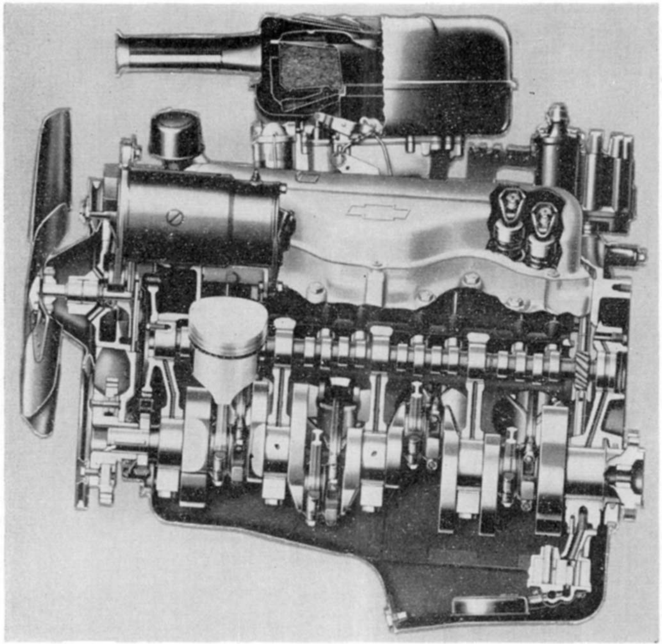

Fig. 4 — Lengthwise view of engine

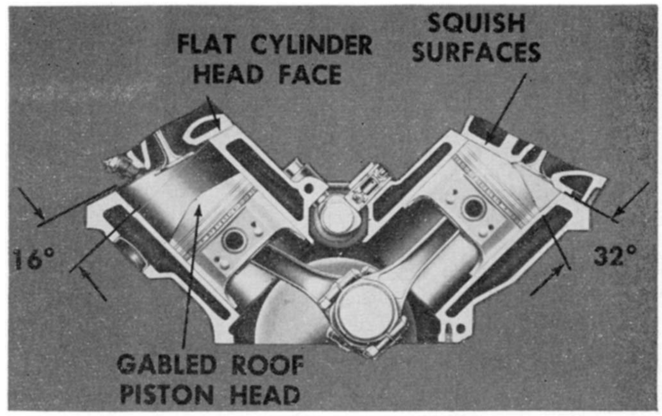

Fig. 5 — Engine block geometry for combustion chamber

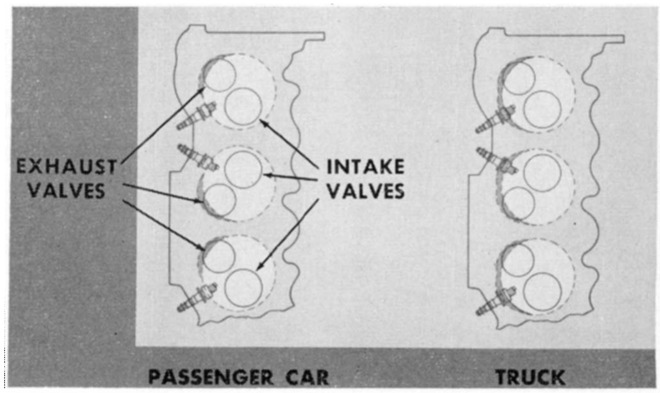

Fig. 6 — Combustion-chamber arrangement

mit attachment of accessory mountings for the various model applications is of serious concern to the manufacturing and service departments. Chevrolet manufactures and services eight different heads for the 283-cu-in. engine because of the requirements imposed by multiple model usage. It would be possible to reduce this number of cylinder heads. However, when the necessary tools and equipment are added to a previously established automated machine line, the cost becomes prohibitive because generally the volume of production for special requirements is low.

If the combustion chamber is placed in the head, the designer is also faced with a dilemma. For good volumetric efficiency at high speed, space for large valves must be provided. At the same time, the new engine needs to have the highest permissible compression ratio, and latitude to go still higher in the future. Unfortunately, these requirements are not compatible; eventually, it would become necessary to compromise at the expense of major cylinder head machine equipment changes.

Making the cylinder head with a flat bottom and placing the combustion chamber in the upper cylinder bore appeared to have possibilities of meeting our objective. Regardless of what changes might be made in piston shape, stroke, or bore size, the flat bottom cylinder head would remain the same. It would also lend itself to freedom from valve shrouding, promoting efficient flow characteristics. The actual combustion-chamber shape could be achieved by contouring the piston head, by angling the top of the block, or by a combination of both. How it was to be done depended on what we believed necessary to establish a sound combustion-chamber design:

1. Compactness, for fast burn rate.
2. Adequate quench and squish area, for turbulence.
3. Central spark-plug position, for minimum flame travel.
4. Latitude to obtain different combustion volumes for broad compression ratio range, without affecting piston shape or basic machining equipment.

Inclining the top of the block to 16 deg and shaping the top of the piston like a gabled roof with a 16-deg angle resulted in a 32-deg wedge-shaped combustion space. Approximately one-half of the piston top surface and the underside of the cylinder head, which are parallel, provided the desired quench area (Fig. 5).

The addition of two milled cutouts to extend the volume of the combustion wedge can provide a compression ratio of 7.5/1; one milled cutout produces a 9.5/1 compression ratio (Fig. 6).

Compression Ratio Flexibility — The difference between the volume of these cutouts provides a wide compression ratio range without making any changes in the piston or cylinder head. The number or size of cutouts is varied simply by adding or removing cutters.

Because of the position of the exhaust valve in relation to the cylinder bore, a definite minimum cutout is required to provide clearance for exhaust valve lift.

Future increases in compression ratio beyond 10/1 can be accomplished by modifying the top of the piston. The manufacturing equipment has been designed for this eventuality and changes can be made at reasonable cost.

The staggered valve positions were also advantageous in establishing the spark-plug position. The plug is in a favorable position for good scavenging, fast burn rate, and freedom from oil fouling.

Placing the plugs above the exhaust manifold permits shorter wires between the distributor and

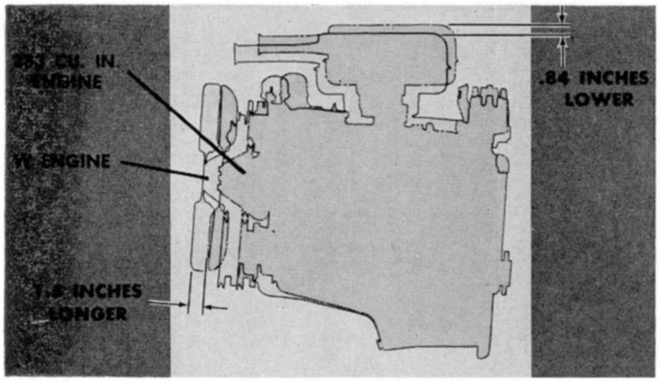

Fig. 7 — Overall size, "W" engine versus 283-cu-in. engine

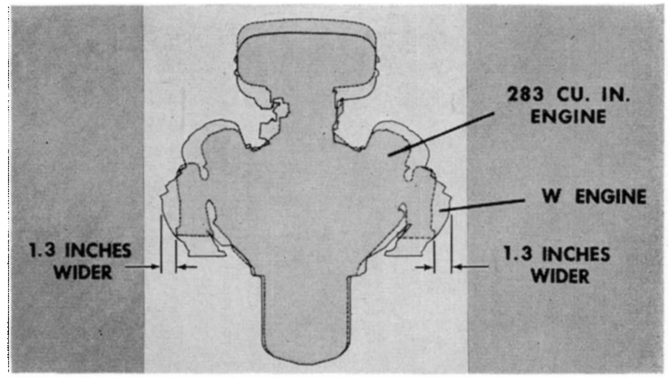

Fig. 8 — Overall size (width), "W" engine versus 283-cu-in. engine

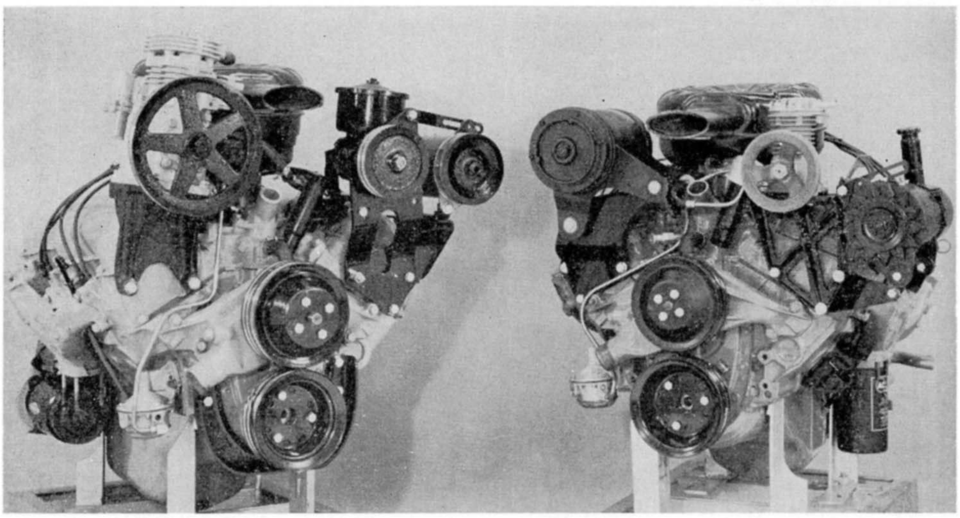

Fig. 9 — Mounting of compressors; left — truck engine, right — passenger-car engine

plugs and better arrangement of wires. This location eliminates the possibility of burned wires and makes the plugs easily accessible.

External Dimensions — During the entire design program, we kept in mind the space limitations imposed by passenger-car design, which had determined our third objective for the new engine. That we succeeded in meeting this limitation is indicated by the fact that the "W" engine assembly with a piston displacement of 348 cu in. is only 1.5 in. longer and 2.6 in. wider than the 283-cu-in. engine assembly. In height, we were able to effect a decrease of about 0.8 in. (See Figs. 7 and 8.)

To meet the requirements of the fourth objective, providing mountings for optional accessories, it was necessary to make composite studies of mountings for all the accessories deemed necessary for passenger cars and trucks.

Accessory Mounting — Placing three tapped holes in the end wall of the cylinder head and two tapped holes on the top of the inlet manifold made it possible to install durable mounts and brackets. These points of attachment are used to mount the compressors for air conditioning and Level Air ride for passenger cars and to mount the air-brake air compressor and the power-steering pump for trucks (Fig. 9).

The group of three holes is placed at both the front and the rear of the cylinder head to eliminate the need for a right- and left-hand unit.

By careful planning for both the present and the future, and coordinating this program with our manufacturing department, we automatically covered the requirements of the fifth objective. We believe we have taken a long stride toward achieving maximum flexibility in the use of automated manufacturing equipment, with resultant long-range economy.

Development Program

This completed the conception and initial design phase of our "W" engine program. Chronologically, we were now at the point when the first prototype engine was completed and ready for laboratory development.

In order to understand the initial direction of the development program, it is necessary to review the overall situation at that time. Chevrolet management felt it was extremely important that an engine meeting the design objectives be available for 1958 production. In view of production tooling lead time requirements, the primary objective of the initial development program was to determine if any problems existed in the design of the engine which might affect major production tooling. It was imperative that this question be answered in a minimum of time. The most serious questions recognized at this point were the following:

1. Was the basic combustion-chamber design satisfactory from the standpoint of specific power output, fuel octane requirement, and fuel economy?
2. Would the location of the combustion chamber

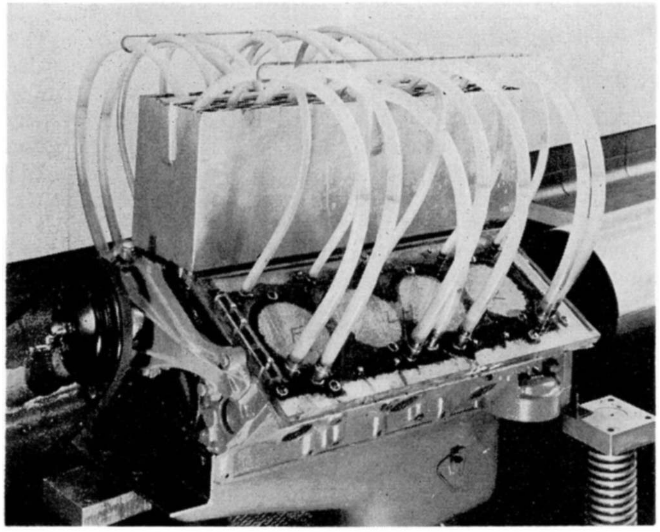

Fig. 10 — Setup enabling visual observation of water flow patterns

inside the cylinder block produce any special cooling problems requiring major tooling changes?

3. Would the larger piston crown area resulting from the gabled head design increase piston temperatures and durability problems?

Cooling System — In order to investigate any of the other questions, it was first necessary to insure adequate combustion-chamber cooling. A total of 40 thermocouples were installed in the first engine at suspected localized hot spots. In spite of an extended program to insure adequate water flow and proper distribution of metering holes in the top deck of the cylinder block, it was not possible by this means alone to eliminate hot spots adjacent to the combustion-chamber surfaces.

Inasmuch as water temperature is only a secondary approximation to heat transfer at a given point in the cooling system, the top deck of the cylinder head was removed and replaced with a plexiglass sheet for visual observation of water flow patterns (Fig. 10). Visual observation of water flow in these areas confirmed our suspicion that there was very little turbulence and, therefore, poor heat transfer along the outer wall of the cylinder block. The nature of the flow and turbulence patterns and hot spot temperatures are shown in Fig. 11.

An experimental modification was made to the water pump to direct the coolant discharge along the outer edges of the cylinder block. This arrangement showed a major improvement in turbulence in the critical areas and also eliminated the hot spots. Fig. 12 shows the turbulence patterns and typical temperatures with the modified water inlet. This arrangement was then released in place of the original design.

Performance Characteristics — With satisfactory combustion-chamber cooling assured, the next objective was to determine whether the performance characteristics of the combustion chamber were satisfactory. Dynamometer measurements of bmep, leanest best torque fuel requirements, minimum spark advance for best torque, and borderline knock characteristics indicated satisfactory combustion-chamber performance in comparison with our 283-cu-in. engine.

Combustion-chamber pressure cards were obtained and compared to our 283-cu-in. engine. Fig. 13 shows the comparative pressure cards; the indication was that combustion-chamber characteristics would be satisfactory. Mbt spark requirements as shown on Fig. 14 also were found to be acceptable.

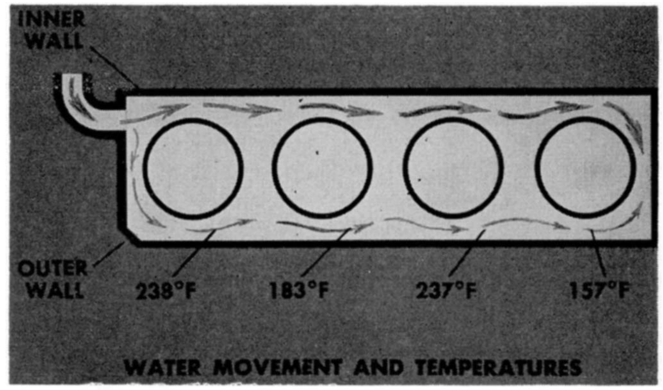

Fig. 11 — Cooling system of original "W" engine

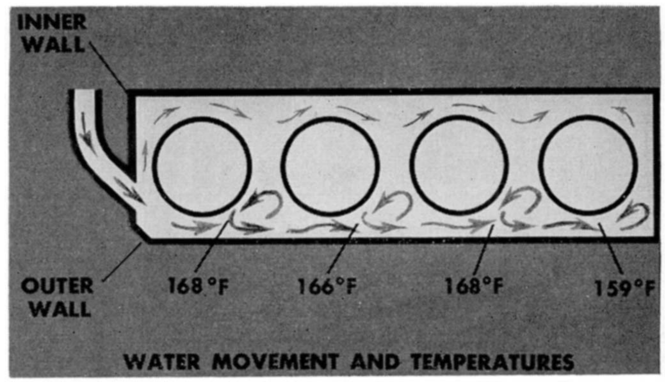

Fig. 12 — Cooling system of revised "W" engine

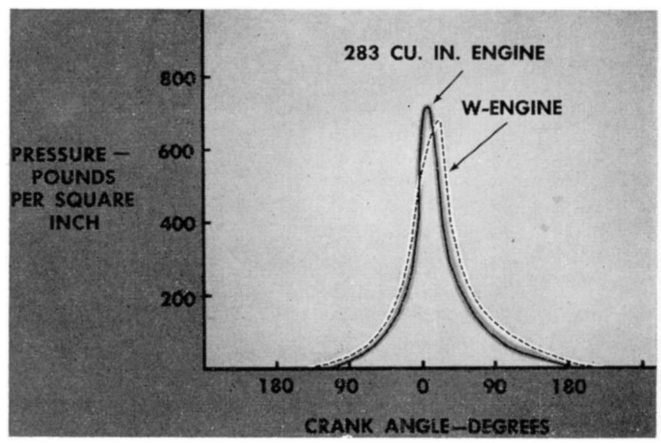

Fig. 13 — Cylinder pressures, 3000 rpm

Piston Durability — The next area of concern was durability of the gabled head piston design. Initial piston durability tests were not wholly satisfactory, but indicated potential for development. Subsequent modification, involving a slight increase in thickness of piston head and ring belt, gave us the needed strength and durability. These early durability tests also gave assurance that the mechanical structure of the cylinder head, block, and crankshaft would satisfactorily carry the imposed loadings. To verify further the conclusion that high piston head temperatures would not present a serious problem, actual piston temperature measurements were made. A comparison revealed that piston temperatures were higher in the "W" engine than in the 283-cu-in. engine. The difference, however, was not much more than might be expected with the greater heating to cooling area ratio inevitable from the larger piston diameter alone.

At this time it was felt that sufficient development work had been done to predict that the basic engine design would make a satisfactory product. This was also the point of no return insofar as the production tooling was concerned, as the basic production machinery was on order. If we made any major engineering changes after this, we would incur large cancellation charges and might find it impossible to meet production deadlines.

Most of the remaining development problems were solved by conventional techniques. However, certain aspects of the programs relating to valve train, camshaft selection, and crankcase ventilation system are worthy of discussion.

Valve Train — The valve train is similar to that used on the 283 engine with the exception that the rocker arm ratio has been increased from 1.50 to 1.75. This change reduces the effective inertia of the push rod and valve lifter, and also makes possible a substantial increase in cam nose radius with a resultant reduction in contact stresses. Modifications to valve springs, valve lifters, and valve train rigidity resulted in a final limiting speed of 5400 rpm. Limiting speed is defined as the speed beyond which the engine will not develop satisfactory power due to valve train malfunction.

A high limiting speed is very desirable to minimize the danger of valve-to-piston interference due to accidental engine overspeeding. The high limiting speed of the Chevrolet engine is a result of good lifter performance, the low weight and high rigidity of the stamped rocker arm construction, and the use of flat wire valve spring dampers. We first used the valve spring dampers in the 1956 models for the 265-cu-in. engine. Without dampers, the spring oscillations continue through the valve-closed portion of the valve train operating cycle, seriously affecting the dynamic characteristics of the valve train as well as the durability of the valve spring. With the dampers installed, the valve spring oscillations are quickly damped out, with a resulting improvement in valve train quietness and durability (Fig. 15).

Fairly late in the development program some difficulty was experienced with valve heads breaking off where the valve head joins the stem. It was suspected that the failures were caused by high dynamic seating loads resulting from excessive valve guide clearances or sporadic lifter leak-down.

A cylinder head was modified so as to include a strain gage load cell arranged to measure valve seating loads. This cylinder head was operated on a dummy setup and valve seating loads were measured with various tappet and valve guide clearances. Fig. 15 shows typical traces taken at 4500 rpm with normal production parts. The effect of valve seating and spring surge are clearly seen. No abnormal

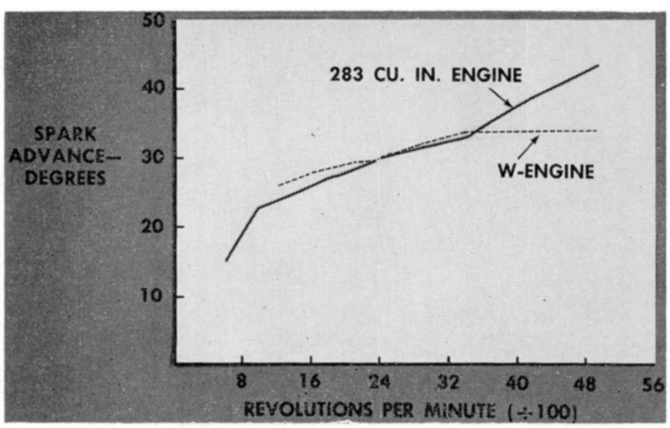

Fig. 14 — Spark requirement, minimum advance for best torque

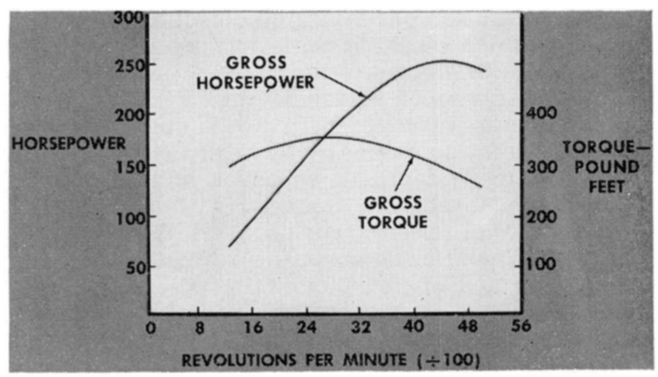

Fig. 16 — Engine characteristics of 348-cu-in. "W" engine with 4-barrel carburetor

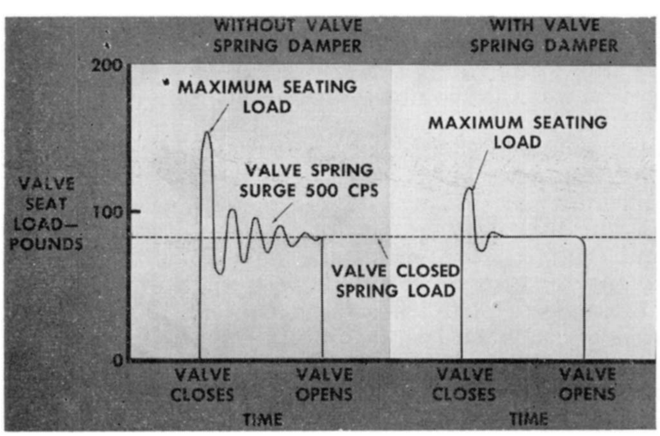

Fig. 15 — Valve seating loads, dynamic measurements at 4500 engine rpm

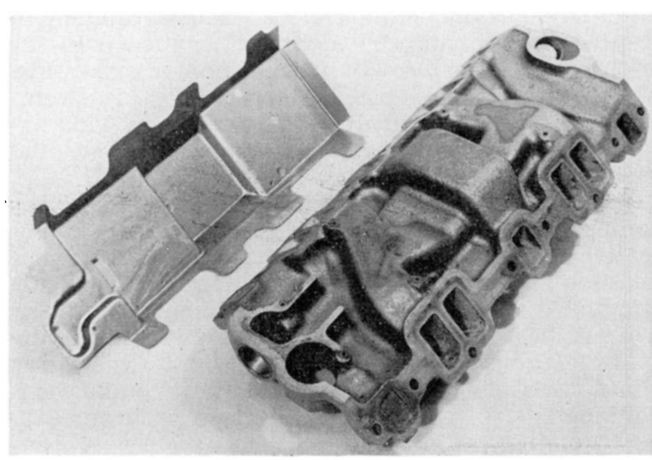

Fig. 17 — Cast rib and shield of manifold

loading was found with normal valve guide clearances. A combination of a badly worn guide and lifter leak-down was found to produce loads in the region of 2500 lb, approximately the hot strength of the valve. The valve guide wear problem was solved by careful attention to valve guide surface finish and valve stem lubrication. An additional factor of safety was obtained by a design modification to eliminate a stress raiser where the valve stem grind run-out blends with the underside of the valve head.

The final selection of the camshaft was based on a study of overall powerplant characteristics in combination with the Turboglide transmission. The speed range of 2400–3200 engine rpm was found to be vital in producing good low-speed and mid-range performance; therefore, an effort was made to provide ample torque in this speed range. The final powerplant performance accomplishes this objective as shown in Fig. 16.

Crankcase Ventilation — The original design crankcase ventilation was satisfactory, except that modifications had to be made in the oil separator. The oil separator was incorporated as part of the intake manifold assembly. It consists of integral cast ribs which serve as walls for air passages, plus an attached shield riveted to the underside of the manifold (Figs. 17 and 18). This simple design worked out well after two basic problems were solved: (1) entrance air velocity was too high for effective oil separation, corrected by deepening the rear section to provide more passage area; (2) drainage of the oil after separation. Effective drainage was achieved after we replaced the two original ¼-in. holes in the shield by two louvered slots extending the full width of the passages. With these changes, the oil separation characteristics had improved to the extent that air could be drawn through the separator at the rate of 10 cfm at 5000 engine rpm without oil pullover. This figure, when compared with normal engine blowby of approximately 2 cfm, gives a safety factor of 5.

Fuel Requirement — The distributor centrifugal advance curve is shown in Fig. 19. With this advance curve, loss from best torque is only 5% at 1200 rpm, 3% at 2400 rpm, and 2% at 3600 rpm. Fig. 19 also shows the fuel octane requirements of an engine with representative combustion-chamber deposits and cooling water temperature at 190 F. A gasoline with a Research octane number of 97 satisfies the engine's requirement throughout the usable speed range.

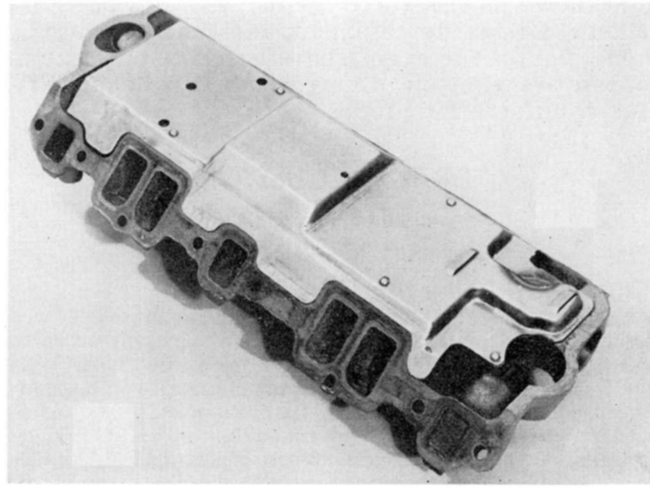

Fig. 18 — Shield riveted to underside of manifold

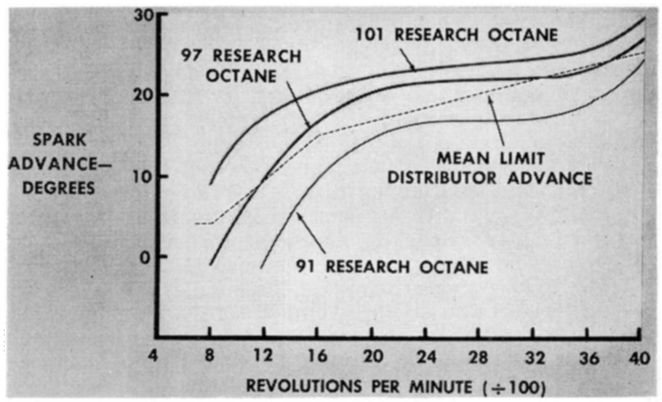

Fig. 19 — Fuel octane requirements of 348-cu-in. passenger-car "W" engine

Durability Test Program

The extremely large volume of Chevrolet production makes it imperative that any product placed on the market be free from defects. Therefore, the durability testing of the engine was of paramount importance throughout the program. In the laboratory, two separate types of dynamometer durability tests were run. Valve train, pistons, crankshaft, and bearings were evaluated on a 4500-rpm full-throttle basis. Total engine test evaluations were made on a long-term cycling durability test.

By the time construction of pilot line engine had begun, more than 40 experimental engine assemblies had been put through tests of varying length and severity over a period of about 18 months. One of the engines built in the Chevrolet Engineering experimental shops had successfully completed a 200-hr wide-open throttle test at 4500 rpm — equivalent to 20,000 miles at 100 mph — as well as a 1000-hr cycling durability test.

The pilot line prototype engines were built by the Tonawanda engine plant, using production tools wherever possible. The purpose of the test program using the pilot line engines was to determine whether any durability problems might develop as a result of variations from experimentally built engines to production engines.

Tests were conducted independently by the Tonawanda engine plant and the Chevrolet Engineering Center. One result of this phase of the testing program was that minor changes were made in the cylinder head and block castings to correct for foundry core shift variations.

Durability testing of experimentally built and preproduction engines on the road was carried out both in General Motors Proving Ground passenger cars and trucks, and in fleet vehicles. Engines in commercial fleet vehicles underwent a wide variety of service conditions over almost 200,000 road miles. At the time it entered volume production, the new engine had a succesful durability test history equal to over a million vehicle miles of operation.

As a result of the advanced pilot engine test program and the manufacturing program at Tonawanda, we were able to supply this engine in the quantities required by our sales department for the start of the 1958 model year.

Customer acceptance of the new engine, which is now known as the Turbo-Thrust V-8, has been excellent. Design, development, and timely release by production of the engine have been the outcome of cooperative effort by Chevrolet design, laboratory, and manufacturing groups.

ORAL DISCUSSION

Reported by Vincent Ayres

Eaton Manufacturing Co.

J. D. Turlay, Buick Division, General Motors Corp.: The new engines under discussion appear to be very well planned, sound designs, and I am sure they will prove to be excellent engines. The new Chrysler line is certainly more conventional than the hemispherical combustion-chamber models and does show the influence of other competitive engines. This is no reflection on the designers for the best products usually result from adoption of the best features of existing designs. We are all deeply involved in this game of comparing, refining, making improvements, and reducing cost, and we are all indebted to others for some of our design details. Also, once a general design is established, certain advantages and disadvantages inevitably accrue and in some cases desirable features cannot be adapted to a given design; for instance, we have not been able to work out a satisfactory stamped rocker arm for our engine.

Reference is made in both papers to the effect on design of the physical environment in which the engine is placed. We, too, have found that this is a most important factor, in fact in many cases chassis considerations seem to be more important than the ideal functioning of the engine components. The reduced height of the engines is very desirable in view of the styling trend toward lower and lower hood lines.

The weight saving of the new Chrysler engine is indeed worthy of comment. I agree that all materials cost something and that a reduction in weight usually results in overall manufacturing economy in addition to the many engineering advantages obtained. One feature influencing weight which was not mentioned is the necessity for designing for machine tool equipment. For example, we have had to increase the rigidity and weight of our crankcase considerably, not for durability reasons, but merely to withstand the rapid metal removal of the big broaches.

The Chrysler combined intake manifold gasket and push rod cover appears to be a very clever and inexpensive design. It is apparent that all the new "V" engines are adopting means of this sort to eliminate the necessity for dry sand cores in the push rod compartment of the cylinder block and thus make possible weight saving and foundry economies.

In all the new engines, the structural rigidity of the crankcase and crankshaft was evidently a major consideration, and they appear to be very adequate.

There are several features of the Chrysler engine of which I heartily approve since they were used in Buick straight eight and continued in the V-8, such as the series flow cooling system, the extended crankcase skirts, and the extended cylinder bores to shorten the water jackets. Evidently Chevrolet is not convinced of the value of series flow cooling or extended crankcase skirts as these features were not incorporated in their new engine.

The new Chrysler cylinder head design provides a minimum of water capacity and should reduce the heat rejection and yet provide high water velocity and excellent cooling with the series flow system. The exhaust ports are commendably short and the air jacketed exhaust crossover is a novel design.

Either there has been a complete reversal of opinion at Chrysler in regard to spark plug location or, as I think is more probable, the new location at the extreme edge of the chamber was a compromise which was found necessary in designing the remarkably compact and light new head. I would be interested in hearing the experiences of the Chrysler authors with this plug design as compared to their almost ideal centrally located and well-cooled plug in their hemispherical combustion chamber, especially in regard to fouling tendencies.

The new flat Chevrolet cylinder head would seem to provide manufacturing economies, as well as excellent breathing potential. I wonder how many recall that the Chevrolet "490" introduced in 1915 had a similar flat cylinder head with the combustion chamber in the upper cylinder bore. Of course, the 1915 version had a pancake-shaped combustion chamber with no squish area, as combustion-chamber development had not yet progressed very far.

The spark plug is more nearly centrally located in the Chevrolet than in the new Chrysler head, but like the Chrysler appears to be only partially water jacketed. It would seem that some difficulty might be encountered in keeping the combustion-chamber face flat and the valve seats square with their guides, because of the thermal and gas pressure effect on the large flat face of the Chevrolet chamber. Would the authors like to comment on their experience with these problems?

Apparently, the relative position of the spark plugs and exhaust manifolds has not been stabilized. In all the competing engines these two parts seem to be playing a game of leap frog; first one is on top and then the other. I note that in the new Chevrolet engine the plugs have moved from below the manifold to above, but in the Chrysler engine they have moved from above to below. They have also reversed positions on the new engines of another competitive line which is not at present under discussion. Undoubtedly, the plugs above the manifolds are easier to service, but with a full quota of engine accessories the engine is so completely covered that there is little advantage either way. Certainly by the time these engine accessories are removed to gain access to the plugs the exhaust manifolds should be cooled sufficiently to present no hazard to the mechanic.

A statement was made in the Chrysler paper as to the use of "desirable" flat top pistons. As one of the earliest users of bump-top pistons I feel we should rise to their defense. Pistons with contoured tops present no cost handicap as permanent mold top surfaces are satisfactorily smooth and accurate without machining, and compression heights can be held sufficiently accurately. We have found that with some piston dome shapes a considerable improvement in structural strength is achieved, permitting the use of a lighter head thickness and an actual saving in weight. The Chevrolet paper pointed out the manufacturing advantages of modifying the piston top instead of the cylinder head to obtain changes in compression ratio necessary with multiple model usage. Finally, piston bumps provide a very convenient method for changing the volume distribution of the combustion chamber and greatly increase the flexibility of the chamber design.

Both papers implied that provision had been made in the new designs for further displacement increases. Evidently, the displacement race is not over, despite all the statements to the contrary.

Mr. Moeller: Reference was made by Mr. Turlay of Chrysler having abandoned the hemispherical combustion-chamber design engine for one more conventional. This change is certainly admitted, but perhaps an additional comment is desirable. There is no change in the thinking that for maximum high-speed volumetric efficiency and power output, the larger valves and unrestricted port shapes in the hemispherical engine are ideal. However, the weight savings and general simplification possible with a line valve type of cylinder head made it attractive for consideration in the new engine. After development of the valve and port details of the new engine, it was found that engine output losses, as compared to the hemispherical engine, were negligible except for a small percentage difference at high speed in the range of 4000 rpm.

In regard to the spark-plug position in the new engine, it has been very satisfactory with no greater tendency towards fouling or overheating than any other engine we have tested. The generally favorable characteristics of the series flow cooling system undoubtedly contributed to these desirable results. The flat top configuration of the piston reduced the heat input, which is desirable.

Mr. Rausch: The tendency of a flat-bottom cylinder head is to show most trouble with gaskets. The use of six bolts per cylinder successfully eliminates leakage of gas. There also is no excessive bore or cylinder head distortion.

The design of crankcase split at the crank centerline is used to reduce overall cost of manufacture. Its strength is adequate with the bulkhead as designed.

The metered-type water system is adequate based on past performance. Piston temperatures were not compared with those of other types of piston design, nor was it felt necessary since durability was satisfactory.

Spark-plug temperatures were checked as satisfactory and no problems arose as a function of new plug location. The problem of plug durability is best resolved by issuing bulletins recommending plug types to be used in the field for the type of service involved.

P. M. Clayton, Ford Motor Co.: The advantage to Chevrolet of using their valve location is probably related to their ability to machine the bores. The new combustion chamber used by Chevrolet appears to be satisfactory but Ford still prefer the in-line valve arrangement and see no disadvantages in it.

The engine compartment seems to be full regardless of make, which complicates service and is a challenge to the designer. Chevrolet engineers are to be congratulated on the excellent high-speed valve gear dynamics demonstrated in their motion photography. The use of high-speed photography to study this phenomena appears well-recommended.

The continued use of metal valve stem seals by Chevrolet might be questioned inasmuch as it seems like this could be improved by synthetic materials.

Ventilation of engines is usually last in any new design evaluation and, therefore, becomes a compromise.

In regard to fuel economy, the large displacement engines have an advantage when compared to smaller engines because of lower friction at their lower operating rpm assuming that proper axle ratios are used.

Mr. Rausch: The valve position adopted by Chevrolet was used to obtain necessary quench area in combustion chamber. The subject of valve stem seals is always under investigation for possible improvement. The design we are using is best for our conditions. I agree that ventilation is always a problem, one eventually resolved in a compromise.

Mr. Moeller: Large engines operating with low numerical axle ratios have a lower noise level than smaller engines running at higher speeds. However, our experience indicates that there is a limit where the combined effects of lower friction and poorer thermal efficiency of the slow speed engine have a detrimental effect on fuel economy. This result is particularly emphasized when automatic transmission torque converter slip is taken into consideration. Consequently, an engine size should only be large enough to obtain adequate car performance with an axle ratio which allows a satisfactory engine noise level.

W. D. Innes, Ford Motor Co.: The 1958 V-8 Chrysler engine has many new and novel features. It is interesting to compare this engine with its predecessor which, if memory serves me correctly, was introduced in the year of 1951. This new engine is relatively the same size engine in cubic-inch displacement as the 1951 engine; however, package-wise, it has changed considerably. The 1951 engine was essentially a square bore and stroke engine; this engine follows the modern trend of short-stroke design.

A major benefit of a short-stroke engine compared to an engine of similar cubic-inch displacement but with a square bore and stroke relationship is the increased efficiency of the engine itself. This generally will be indicated in increased usable power and should result in an improvement in fuel economy, provided the engines are placed in identical circumstances. No data has been presented on this comparison today; however, it would have been a very interesting comparison in basic engine design.

The novel features of this new Chrysler engine are: (1) cylinder head — reduced water jacketing, (2) new carburetors, (3) one-piece intake manifold gasket and valve chamber cover, (4) stamped rocker arm mounted on a rocker arm shaft, (5) cylinder block — rigid deep skirt design, and (6) throwaway-type oil filter.

The cylinder head design is unique in regard to the amount of water jacketing it has. This V-8 engine has approximately 6 qt less coolant than the conventional V-8's; however, in some areas of the cylinder head this reduction might be harmful. In looking at the design of the cylinder head in the intake port area, it is possible that water jacketing could be helpful in maintaining proper mixture temperatures particularly in the cold weather operation. The small water jacket section in the cylinder head would seem to create a foundry problem, handling water jacket cores with reduced sections—however, not knowing the casting procedure of this engine, it is difficult to tell. Another novel feature of this cylinder head is the air space around the exhaust crossover port. This certainly looks like an advantage in keeping heat out of the cylinder head proper.

Another feature of the new Chrysler engine is the cylinder block with its rigid deep skirt design. Our experience with this type of construction parallels the benefits mentioned in the paper.

The induction system of this new engine is more-or-less conventional with the exception of the carburetor and automatic choke combination. However, I did notice the bottom of the intake manifold risers has been squared-off now in comparison to rounded corners present in the 1957 engines. This, I feel, is an aid to better distribution with the heights of risers we now have to live with in these lower package heights.

The new carburetors used on this Chrysler engine have certainly been reduced in height. This new trend in carburetion opens entirely new concepts in the induction design field. I noticed in the paper that the dual carburetor has a mechanically operated bowl vent which is open only at idle and is closed at normal operating condition, making the carburetor fully balanced. This would certainly eliminate any air cleaner buildup or restriction from enrichening the fuel flow under operating conditions. However, in the 4-barrel carburetor a combination of internal and external vents are used, making an unbalanced carburetor, which could affect fuel flow under increased air cleaner restriction. In our experience with a similar type of air cleaner, restriction under normal driving conditions buildup is so slow that after 20,000 miles the increase is only 0.5–1.0 in. of water; therefore, I do not believe this is a serious problem.

The new econo-choke which we have heard so much about is certainly a novel arrangement. The two-staged vacuum piston mounted in the air horn is definitely a new feature. The basic premise of reducing the choke enrichment under operating road load conditions is excellent. The only difficulty would seem to be a possible hysteresis when operating under road load conditions and then suddenly going to wide-open throttle which could possibly cause a leaning out condition. This hysteresis is not the choke plate closing but is the metering system itself recognizing this change in plate closing. In regard to conventional chokes coming off fast under slow driving conditions and slow under fast driving conditions, it might be pointed out that the difference in manifold vacuum running road load at say 20 and 50 mph is 3 in. of Hg and the variance in total airflow over the choke bimetal spring is only 0.3 of a cfm. However, the amount of heat available to pass over the bimetal spring is determined by the road load condition at which the vehicle is being run. The heat available is generally the same for any type of choke arrangement excluding the possible losses in tubes to heat boxes and the like. The Chrysler setup, however, with its heated mass which will keep the choke off under warm engine conditions proves very beneficial.