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TECHNOLOGICAL IMPROVEMENTS TO AUTOMOBILE FUEL CONSUMPTION

Volume IIA: Sections 1 through 23

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16. Abstract

This report is a preliminary survey of the technological feasibility of reducing the fuel consumption of automobiles. The study uses as a reference information derived from literature, automobile industry contacts, and testing conducted as part of the program requirements. The design changes, which are recommended for the purpose of maximizing fuel economy, have been derived after lengthy review against a series of constraints including regulatory requirements, technical feasibility, and cost effectiveness.

Several possible technological improvements are identified, documented, and evaluated with respect to fuel economy. Results are reported as percentage improvement in fuel economy by comparison with 1973 model year vehicles. The effect of vehicle emission control systems is considered in the evaluation procedure.

The most promising individual improvements are incorporated into three synthesized vehicle designs, and the projected fuel economy improvement for these vehicles is reported.

The status of the technology reported is that available in the time period of July 1973 to January 1974.

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PREFACE

The transportation sector of the U.S. economy accounts for approximately 25 percent of the total energy demand, predominately in the form of petroleum fuels. The Government has been actively engaged in reviewing the technological and institutional actions that can be taken to reduce our transportation energy demand. One such effort is the preliminary study covered in this report on the technological feasibility of improved fuel economy in automobiles.

The work described in this report was performed by Southwest Research Institute for the U.S. Department of Transportation and the U.S. Environmental Protection Agency. The project was monitored by the Power and Propulsion Branch, Mechanical Engineering Division, Transportation Systems Center, U.S. Department of Transportation. The technical monitor for the project was H. Gould.

The authors recognize the timely significance of this study, and despite warnings to the contrary, information may be taken out of context. For these reasons, the report has been written in an instructive fashion to acquaint the uninitiated reader with facts about automobile design. Hopefully, this instruction will nullify the majority of misconceptions and provide insight into an exceedingly complex issue.

This work does not address the overall automobile transportation energy problem, but it is directed to one of the major components of the American automobile market—the "large" automobile. Specifically, this study is concerned with cars of the 4300- and 3300-lb curb weight classes. These vehicles are frequently identified by Federal Test Procedure inertia weight class with corresponding values of 3500 and 4500 lb.

The status of the technology reported is that available in the time period of July 1973 to January 1974.

The work covered in this report represents approximately a three-man year level of effort and was conducted over a six-month period. The goals of the project are ambitious, and the effort of each member of the project team was vital to the final product. Space does not permit the listing of all participants, but major efforts were contributed by:

- Dr. C. W. Coon, Senior Research Engineer
- B. C. Dial, Senior Research Engineer
- Roger Hemion, Institute Scientist
- R. W. Hull, Senior Research Engineer
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1. INTRODUCTION

The primary objective of the study is to define a series of viable technological improvements which will result in a 30-percent or more reduction in fuel consumption relative to 1973-model year vehicles. It should be noted that the goal may also be stated in terms of a fuel economy increase. The 30-percent reduction in fuel consumption corresponds to a 42-percent increase in the number of miles per gallon of fuel. Another objective of this report is to underscore repeatedly the interrelationships between various automobile design characteristics, as they influence operating fuel economy, and increase the relationship between the vehicle power plant emissions and fuel economy potential.

The conception of automobile design changes for this program was based on the following restrictions:

(1) Recommended technology should be off-the-shelf or in a mature stage of development.

(2) The systems should be demonstrable by 1976 in a small number of vehicles.

(3) Emissions standards (1975 Federal Test Procedure):

NO_X	2.0 g/mile
CO	3.4 g/mile
HC	0.41 g/mile

(4) Safety standards–1973 US/DOT.

(5) Noise–Comparable to reference vehicles.

(6) Reliability–Nearly equivalent to reference vehicles.

(7) Performance–No significant degradation; driveability of any proposed design should be as good as the reference vehicles.

(8) Minimal cost to the consumer including both initial cost and discounted life-cycle cost.

(9) Fuel economy improvements were to be assessed against the LA-4 driving schedule and against road load economy at 20, 30, 40, 50, 60, and 70 mph.

The reference vehicles for the study were selected due to their high sales volume for 1972. All vehicles were equipped with standard V-8 engines, power steering, automatic transmissions, and optional air-conditioning, as these items represented items purchased by the majority of the consumers. The vehicles selected (Figure 1) for the study were:

FIGURE 1. VEHICLES USED DURING TEST PROGRAM

Vehicle	Public scale weight with full gas tank (lb)
Ford Galaxie	4270
Ford Mustang	3470
Chevrolet Impala	4360
Chevrolet Camaro	3560
Plymouth Fury	4190
Dodge Challenger	3490

The 3500- and 4500-lb inertia weight vehicles from each manufacturer were leased equipped with identical power teams including engine, transmission, and rear axle. (See Appendix A for vehicle specifications.) The two Ford vehicles were not directly comparable, however, as two different 351 CID engines were used.

These vehicles were used during this study for two primary purposes: (1) to acquaint the project team with some of the details of 1973-model year technology and (2) to provide test data for in-house use by the Department of Transportation. Approximately one-half of the contract funds were expended in the acquisition, handling, and preparation of the test data.

In addition to the testing of the 1973-model vehicles, this report studies improvements in areas of vehicle design such as engines, transmissions and axles, tires, aerodynamics, vehicle weight, and engine accessories and auxiliaries.

The program was not designed to reduce any fuel use improvement to practice, as such action would properly consume a considerable amount of development funds. The objective of the study effort was to obtain the maximum amount of available data on any candidate improvement, screen these data, and then analytically project and discuss the potential of the fuel conservative on improvement.

The reader is cautioned against amplifying or projecting the results of this study without carefully weighing the realities that actual hardware will demonstrate. Assumed operational characteristics or assumptions about design performance could result in erroneous conclusions, high costs, or design penalties in other areas of the vehicle without the confirmation of prototype assessment.

2. AUTOMOBILE SYSTEMS AND OPERATIONAL REQUIREMENTS: INTERRELATIONSHIPS INFLUENCING ECONOMY

The performance and economy of an automobile is determined by the detailed engineering characteristics of the specific vehicle, the vehicle operating requirements, and the demands of the driver.

In this section of this report, a brief explanation of some of the basic interrelationships of design and operating parameters will be given which forms a generalized background for the interpretation of detailed evaluations of changes in vehicle design characteristics.

The basic physical law governing the motion of an automobile is Newton's Second Law of Motion. In order to place an automobile in motion a force must be exerted through the vehicle driving wheels. The amount of the force depends on the weight of the car and the rate of vehicle acceleration, as well as the magnitude of forces which retard motion such as rolling resistance and aerodynamic drag.

The motive force is applied to the vehicle driving wheels by a combination of the operation of the vehicle engine, transmission, and rear axle. The function of the transmission and rear axle is to deliver the turning effort of the engine to the driving wheels at a speed different from that of the engine. This difference in speed also implies a multiplication of the turning effort of the engine depending on the value of the overall gear ratio.

The turning effort of the engine is termed *torque* and arises as a result of ignition and subsequent expansion of the products of combustion of a hydrocarbon fuel and air mixture within the cylinders of the internal combustion engine. Detail design characteristics of internal combustion engines determine the amount of torque produced by the engine. Basically, it is the quantity of fuel and air ingested by the engine during each revolution that determines the amount of torque produced. Typically, small displacement engines produce less torque than larger displacement engines, all other factors being equal.

Traffic conditions require that a vehicle be operated at varied speeds and acceleration, all of which require varied torques. Ignoring the transmission momentarily, the torque needed to sustain the desired operating condition is obtained by varying the amount of fuel/air mixture processed by the engine. For a vehicle with a carbureted engine, this is accomplished by the modulation of the accelerator pedal. Depressing the accelerator allows more mixture to enter the engine, thus producing more torque. Of course, there is a limit to the amount of torque that any engine can produce with fixed design characteristics. This maximum torque is produced at wide open throttle (WOT). The generalized torque characteristic of a conventional, reciprocating internal combustion engine at wide open throttle is shown in Figure 2.

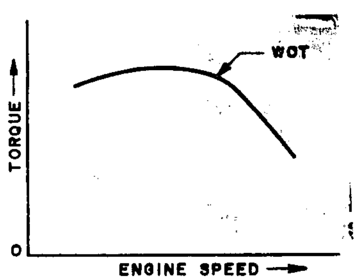


FIGURE 2. GENERALIZED TORQUE CHARACTERISTICS OF AN INTERNAL COMBUSTION ENGINE

The transmission is used to multiply the torque output of the engine so that the vehicle can be subjected to motive forces higher than those developed by the direct connection of the engine to the rear axle. Transmission gear ratios greater than 1:1 allow the torque necessary to move the vehicle at smaller throttle (accelerator) settings. To meet any given vehicle torque requirement, many gear ratios and various combinations of engine displacement and throttle openings can be used.

Another basic concept which must be considered in the discussion of automobile performance and economy is power. To define what is meant by power we must go back to our explanation of the force acting at the wheel of the vehicle. As the force acts on the vehicle, the vehicle begins to move. The force, of course, then acts on the vehicle over the distance through which the vehicle moves. This quantity (force X distance) is termed work, and the rate at which the work is done is termed power.

In engineering terms, the power output of an engine is specified as Power = torque X rpm/constant. The constant is selected to get the units of power into a comparative unit of measure such as horsepower. In essence, power is the price one has to pay when demanding a given force at a given speed. It is the power requirement of the desired motion of an automobile relative to the overall efficiency of the translation of the energy of combustion into meeting this power demand that determines automobile economy.

Before further deliberations on the fuel economy of automobiles, it is necessary to note that other physical laws and rules of standard practice are significant to the subject of fuel economy. Metallurgical considerations dictate the maximum temperatures to which engine components can be exposed. Thermodynamic principles can then be invoked to establish theoretical efficiency boundaries below which real systems can operate. In particular, no practical internal combustion engine can have a thermal efficiency (power out/power available from fuel combustion) over about 50 percent. In other words, it is *not possible* to *use more than 50 percent* (and usually considerably less) of the energy available in the fuel. The remainder is rejected to engine exhaust and cooling systems. Detail design characteristics and operation of various engines dictate how close to this theoretical limit that *real* engines can operate.

The conventional naturally-aspirated (NA) carbureted engine demonstrates a performance (torque vs speed) map under fully warmed-up conditions similar to Figure 3 (adapted from Reference 1). The contours are lines of constant fuel consumption for each unit of horsepower developed (brake specific fuel consumption). This characteristic map is due to two primary effects. The first is the increase in brake specific fuel consumption with decreasing torque output at a constant engine speed due to the higher percentage of friction power at low brake power outputs. Second, brake specific fuel consumption increases with increasing speed (at constant torque) due to increased engine friction. Consequently, the highest fuel use efficiency for the engine occurs at low engine speeds and high torque outputs.

The engine performance map is an extremely significant constituent of the overall economy formulation. This map is the representation of the units of fuel consumed in meeting the power demanded by the motion of the vehicle. The values of the brake specific fuel consumption, as well as the shape of the characteristic fuel consumption curves, are extremely significant in determining accurate estimates of economy. For the purposes of this study, the engine map of Figure 3 has been used as characteristic of a nonemission controlled spark-ignition engine; therefore, the changes in design can be evaluated relative to the same baseline. It is not suggested that this map represents all engines; however it represents a convenient generalization for purposes of comparison. It was found, after test data had been acquired, that the agreement between Figure 3 and the test engines was acceptable.

[1]Obert, Edward F., *Internal Combustion Engines*, International Textbook Company, December 1968.

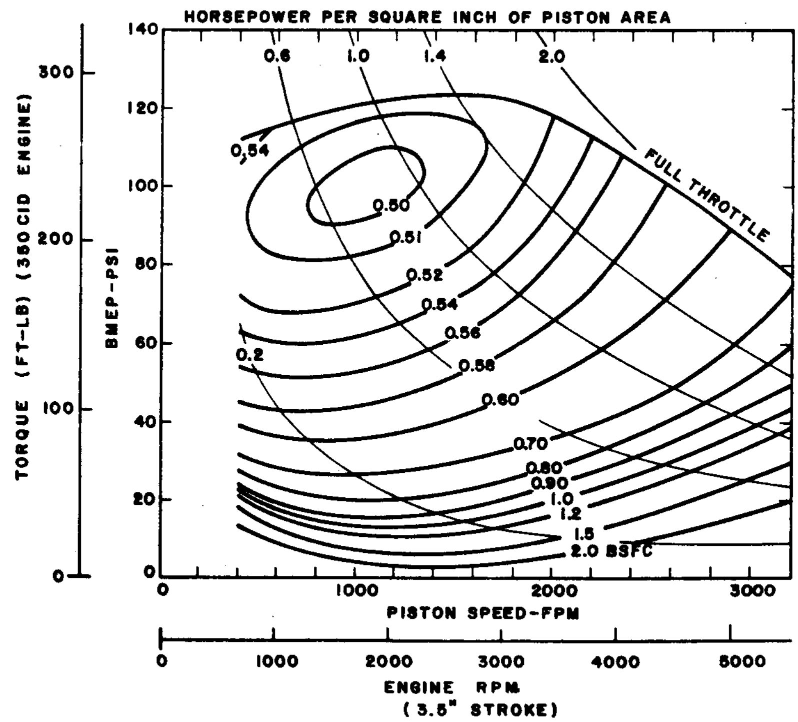


FIGURE 3. PERFORMANCE MAP OF CARBURETED SPARK IGNITION ENGINE

Unfortunately, the conventional application of the engines to vehicles results in a compromise of where the engine operates in the map under road cruising conditions relative to the maximum performance capability of the vehicle. This is illustrated in Figure 4. The road load is met at points of high brake specific fuel consumption. The difference between the torque capability at full throttle and the road load torque value determines the performance (acceleration) capability of the vehicle. If the design characteristics of a vehicle were adjusted such that the road load power requirements could be met by operating near the point of minimum brake specific fuel consumption, then a marked improvement in fuel economy could be gained. However, such a move with conventional technology (such as a very small engine) would also result in substantial reduction in the acceleration and passing capabilities of a full-size automobile. One object of this study was to investigate methods of improving fuel economy while considering the effects such changes may have on the vehicle systems and operation.

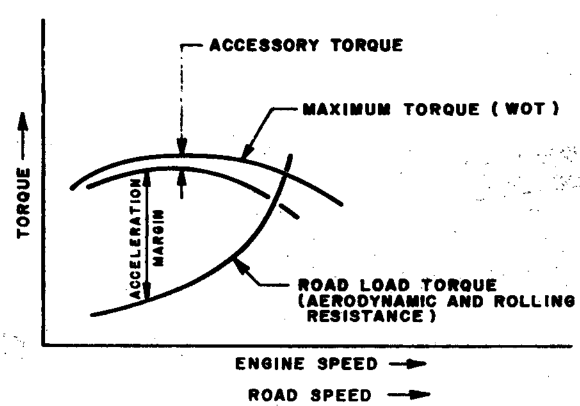


FIGURE 4. COMPARISON OF MAXIMUM PERFORMANCE AND ROAD PERFORMANCE

The above discussion stressed the difference between the full-power capability of a vehicle and that power necessary to maintain level road motion (no acceleration). For a full-size reference, typical road load power requirements are given in Table 1.

This power requirement is determined by a large number of factors including vehicle weight, tire design and construction, road

TABLE 1. TYPICAL ROAD LOAD POWER REQUIREMENTS FOR A FULL-SIZE REFERENCE VEHICLE

Speed (mph)	Power (hp)
20	5.
30	8.2
40	13.0
50	19.0
60	28.0
70	38.5

TABLE 2. 3-PERCENT GRADE POWER REQUIREMENTS FOR 4300-LB VEHICLE

Speed (mph)	Additional grade horsepower	Percent increase over level road
20	6.88	137
30	10.32	143
40	13.75	105
50	17.19	90
60	20.63	74
70	24.07	63

surface conditions, air temperature and density, and vehicle size and shape. The vehicle design must accommodate a range of power requirements which result from operation throughout the U.S. For example, at an altitude of 4000 ft, the road load at 70 mph will be *reduced* by 3.8 hp or about 10 percent as a result of the decreased air density. Operating a vehicle in a 0°F ambient instead of a 100°F ambient will *increase* the road load by 12 percent at 70 mph.

Fuel economy, then, can be seen to be influenced by ambient conditions as well as the vehicle operating requirements.

The economy measured under level road conditions is one method for comparison of vehicle designs; however, the vehicle must be capable of meeting varied road conditions such as grades. "Level" sections of interstate highway are not always level. Often highways include grades, and a 3-ft rise in 100 ft of road is not uncommon. The conventional American automobile can negotiate such grades without requiring a downshift to a lower gear. The significance of grades on vehicle power requirements is illustrated by Table 2.

Negotiating a grade of only 3 percent more than doubles the road load power requirements at speeds up to 40 mph and provides a substantial increase even at 70 mph.

A vehicle cannot be designed to maximize economy under specific level road conditions but must accommodate a wide range of driving conditions encountered in the U.S. The discussion of economy detailed in this report recognizes these factors, however, in order to establish a convenient yardstick, level road power requirements are used in evaluating design changes.

In addition to steady-state driving conditions, transient operation of vehicles is also extremely important. Stop-and-go traffic conditions such as acceleration, braking, and idling constitute typical driving conditions experienced by the public, and these modes of operation must be included in any assessment of fuel economy. The appropriate protocal for simulation of urban driving has been an object of intense discussion among automotive experts; however, no agreement has been reached to date.

One of the most thoroughly documented driving schedules is the EPA Urban Dynamometer Driving Schedule or the LA-4 Dynamometer Cycle. This procedure was developed from observations of urban traffic for use in emission certification tests[2]. Basically, the cycle is a series of non-repetitive accelerations, decelerations, and idling periods, as well as periods of nonsteady cruise operation. The cruise portions are not representative of the road load operation discussed previously, since there is rarely a period of more than several seconds when the vehicle speed remains constant. An illustration of the initial 4 min of the cycle is given in Figure 5. During the entire procedure, a vehicle would operate for about 23 min and cover a distance of 7.5 miles.

[2]Kruse and Huls, "Development of the Federal Urban Driving Schedule," SAE Paper 730553, 1973.

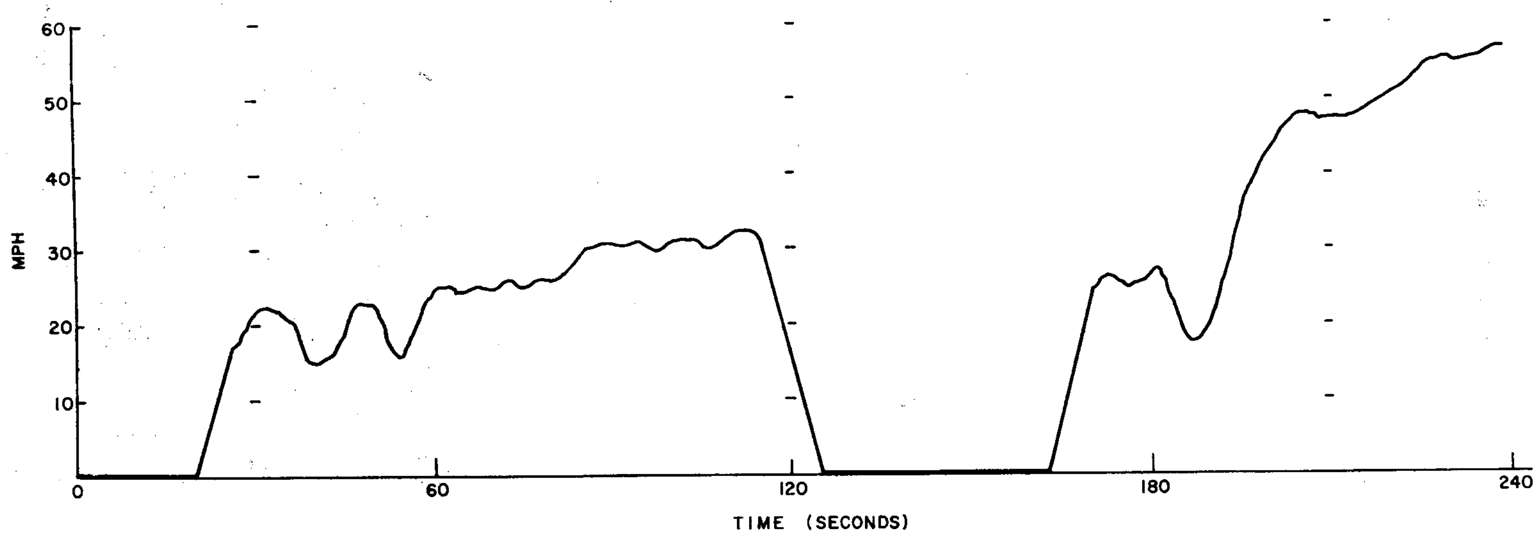


FIGURE 5. PORTION OF DRIVING CYCLE

Although the LA-4 cycle is a chassis dynamometer procedure, it does provide assistance in the analysis of the fuel economy of a vehicle operating in urban traffic. For the purposes of this study, it was assumed that a vehicle could execute the speed-time requirements of the cycle in actual operation. The cycle was divided into component parts as described in Chapter 3, and a procedure was developed for prediction of the fuel consumption.

The LA-4 Cycle emphasizes the parameter of vehicle weight. Under acceleration, weight is the most important parameter influencing fuel consumption. Figure 6 clearly establishes the magnitude of this power demand. Under the maximum acceleration requirements of the LA-4 cycle, the power required to accelerate the 4300-lb vehicle is over six times that required to maintain steady motion at 30 mph. Also shown for reference are the power requirements of accelerating the vehicle at other performance levels. It should be noted that the maximum performance of a full-size reference vehicle lies somewhere between 5 and 6 sec to 30 mph. The performance capability of a given vehicle weight then can be seen to significantly influence the power required and, consequently, the fuel consumption. For further comparison, Figure 7 illustrates the power requirements of a 3300-lb vehicle under the same acceleration loads. The requirements here are 3300/4300 = 76.7 percent of the loads for the 4300-lb vehicle. The influence of vehicle weight can be marked; however, the significance over a driving cycle will depend on the amount of time spent in acceleration modes and the amount of acceleration demand.

Before leaving the subject of vehicle and performance weight, it must be emphasized that the weight of the vehicle is the primary factor that determines the installed power requirements of the vehicle when a given level of performance is specified. For example, in determining the power necessary to accelerate a 4300-lb vehicle to 30 mph in 5 sec, less than a 10-percent error would be encountered in totally ignoring the road load. Conversely, unless the performance level or weight of a full-size vehicle is significantly reduced, the maximum power output of the engine must remain at 1973 reference vehicle levels.

The LA-4 Cycle, when used as a dynamometer procedure, incorporates an additional typical operating feature—a cold start, i.e., the test is performed after the vehicle has soaked overnight and

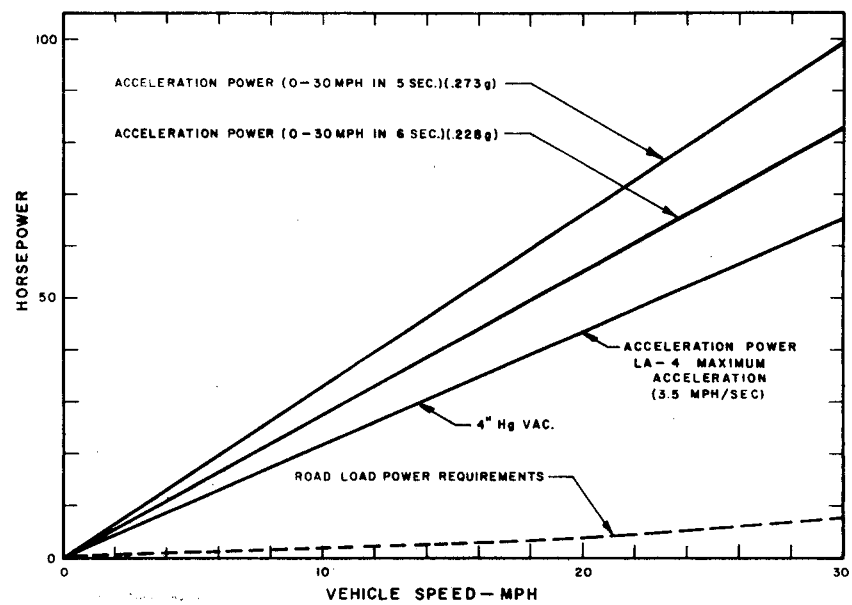


FIGURE 6. MOTIVE POWER REQUIREMENTS (4300-LB VEHICLE)

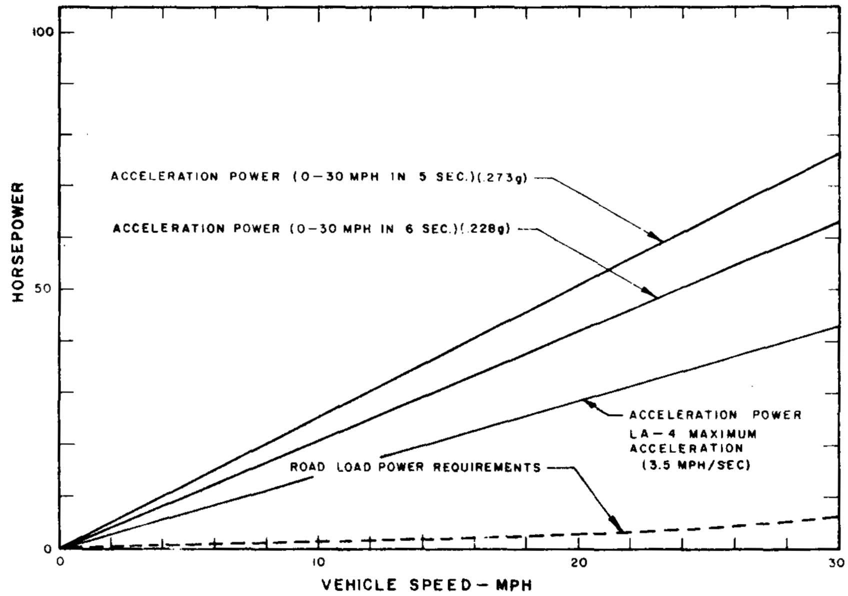


FIGURE 7. MOTIVE POWER REQUIREMENTS (3300-LB VEHICLE)

has come to equilibrium with the ambient temperature of the test lab. The influence on cycle operating economy due to starting and operating a vehicle "cold" as compared to "warm" is significant. The loss in economy is attributable to varied sources, including carburetion enrichment to permit ignition, viscous losses in cold engines, transmissions, and axles and higher heat losses in the engine due to rejection of heat to "cold" coolant. Since the effects discussed above are thermal, ambient temperature would be expected to influence the amount of economy degradation. Figure 8 illustrates this effect (Reference 3). These curves were developed from sequential

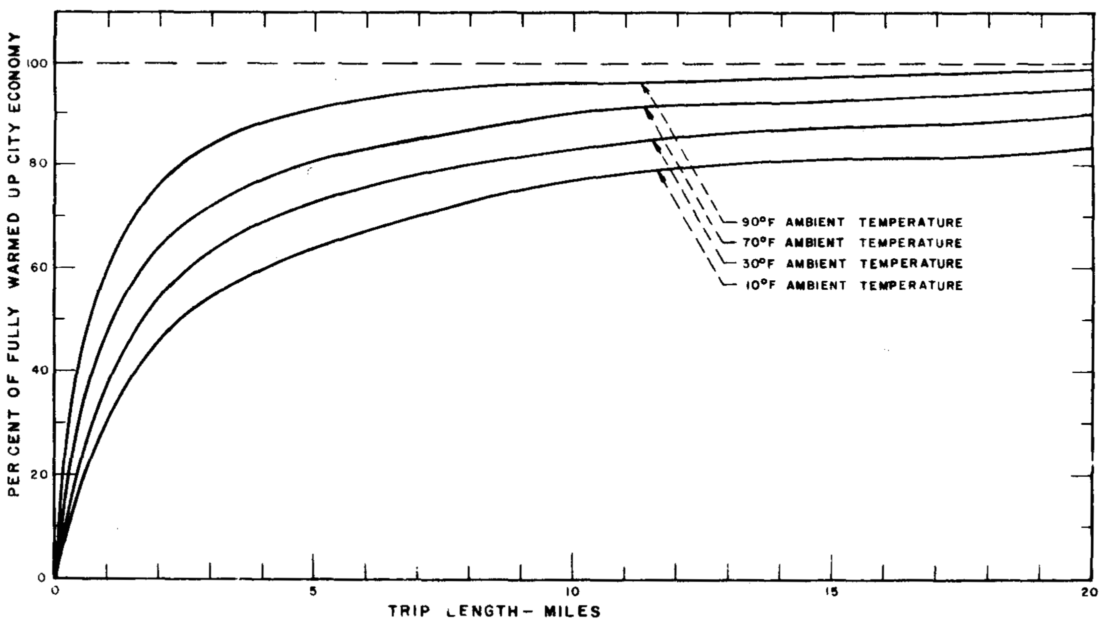


FIGURE 8. WARMUP ECONOMY

[3] Scheffler, C.E. and Niepoth, G.W., "Customer Fuel Economy Estimated from Engineering Tests," SAE 650861.

TABLE 3. AUTO TRIP STATISTICS

Trip length (one-way miles)	Trips (%)	Vehicle miles (%)
Urban mileage		
Under 5	54.1	11.1
5-9	19.6	13.8
10-15	13.8	18.7
16-20	4.3	9.1
21-30	4.0	11.8
31-40	1.6	6.6
41-50	0.8	4.3
51-99	1.0	7.6
100 and over	0.8	17.0
Total	100.0	100.0
Total mileage (%)		
Urban	55.5	
Highway	44.5	

repetition of a driving cycle. Obviously, the driver who makes short trips in cold climates will suffer a significant reduction in his fuel economy from that obtainable from a fully warmed-up engine.

Table 3 taken from Reference 4 illustrates that over one-half of the trips taken in the U.S. are under 5 miles. Although the total miles driven for these short trips are not a major portion of the total travel, improvements in the fuel consumption during warmup will also decrease the overall energy demand.

During the course of this study, some information on vehicle warmup was obtained for LA-4 cycle operation. Fuel economy data for cold and hot starts are provided in Section 3, and a temperature versus time record for two cooling system temperatures is shown in Section 23.

[4] "Motor Vehicle Facts and Statistics," published by Motor Vehicle Manufacturer's Association, 1972.

3. COMPUTATION OF VEHICLE FUEL ECONOMY

There are numerous papers in the technical literature covering predictive techniques and procedures for computing vehicle fuel economy. (References 5 through 9). In this study, we are not concerned with the prediction of absolute fuel economy statistics but with the screening of changes in design to target those changes which will result in marked economy benefits. Consequently, the method employed in this study was a straightforward simplistic procedure amenable to hand calculations. The method results in an absolute fuel economy prediction, but these values are not reported. The results discussed in later sections of the report are expressed in terms of percentage improvements relative to the baseline vehicle rather than absolute numbers.

In this section, details of the computational method are described with reference to the baseline calculations conducted for the full-size reference vehicles. In general, the computation of fuel economy of a candidate vehicle design was based on the major elements illustrated by the flow chart in Figure 9.

Road load fuel economy was based on the power requirements reported in Section 2 for 20 through 70 mph. For the LA-4 cycle, the loads were based on (1) the road load horsepower at 50 mph from Section 2 and (2) the force required to accelerate the vehicle at the specified rates on the LA-4 cycle.

As previously discussed, the LA-4 cycle is a rapidly varying nonrepetitive cycle and, consequently, the forces required to propel any given vehicle are also extremely time variant. To minimize the analytical effort for evaluation of this cycle, an approximation of the cycle described in Table 4 was made. Subsequent evaluation and comparison with other abbreviated versions of the LA-4 cycle indicated that the description in Table 4 was adequate for the present purpose.

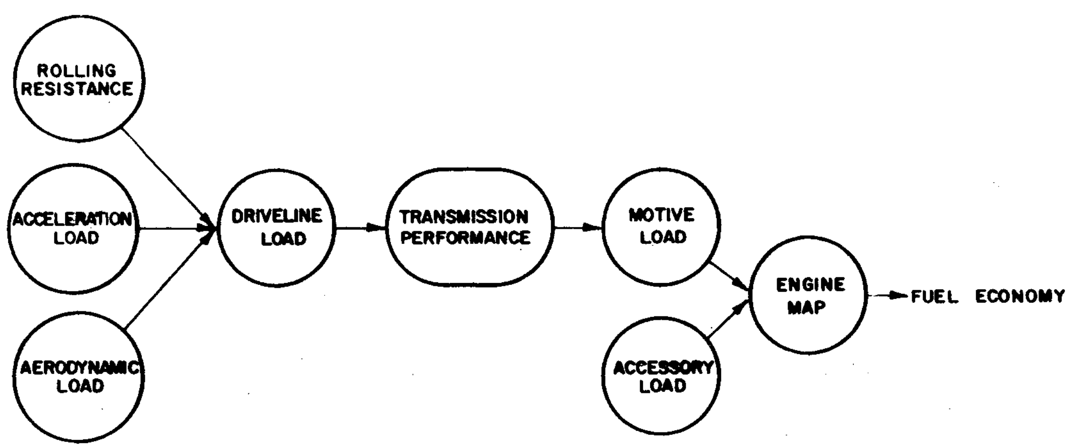


FIGURE 9. ELEMENTS OF CALCULATION PROCEDURE

[5]Jaroslav J. Taborek, *Mechanics of Vehicles* (Reprints from Machine Design) Penton Publishing Co., Cleveland, Ohio, 1957.
[6]J. L. Koffman, "Vehicle Performance: The Effect of Rotating Masses on Acceleration," Automobile Engineer, December 1955, pp. 576-578.
[7]P. M. Clayton, "Forecasting Specific Fuel Economy," SAE Paper 199B SAE Summer Meeting, Chicago, Illinois, June 5-10, 1960.
[8]R. K. Louden and Ivan Lukey, "Computer Simulation of Automotive Fuel Economy and Acceleration," SAE Paper 196A, SAE Summer Meeting, Chicago, Illinois, June 5-10, 1960.
[9]M. A. Ordorica, "Vehicle Performance Prediction," SAE 650623, Detroit Section, May 10, 1965.

TABLE 4. LA-4 CYCLE DATA

Speed (mph)	Time (sec)
Cruise	
20	90
25	312
30	113
35	57
45	15
55	83
Idle	
—	250
Deceleration	
—	240
Acceleration	
0-various	212
Total	1372

The acceleration mode is comprised of a number of accelerations from rest typically to 25 mph and from intermediate speeds to terminal cruise speeds; however, the majority of accelerations are from rest. The acceleration rate is also variable, but our analyses indicate that a typical value of 2.0 mph/sec was representative. The total number of accelerations at all rates on the cycle was 19. These two figures were not consistent with the mode time of 212 sec, so a compromise was performed resulting in the specification of 21 accelerations from 0 to 25 mph at 2.5 mph/sec.

When the LA-4 cycle is executed on a chassis dynamometer, the road horsepower is set at 50 mph, and the power absorbed at other speeds is determined by the dynamometer characteristics. This same procedure was used in the calculations associated with this study in order to preserve sufficient simplicity for hand calculations; however, the actual road horsepower at 50 mph was used instead of the published emission test horsepower setting. The effect of this step is equivalent to the assumption that a constant force is required to maintain any steady vehicle speed; a comparison of horsepower predicted as a result of this assumption with actual horsepower is shown in Figure 10. The force required to propel the vehicle at steady speed was computed from the equation

$$P = FV$$

P = horsepower

F = resistive force

V = vehicle relative velocity

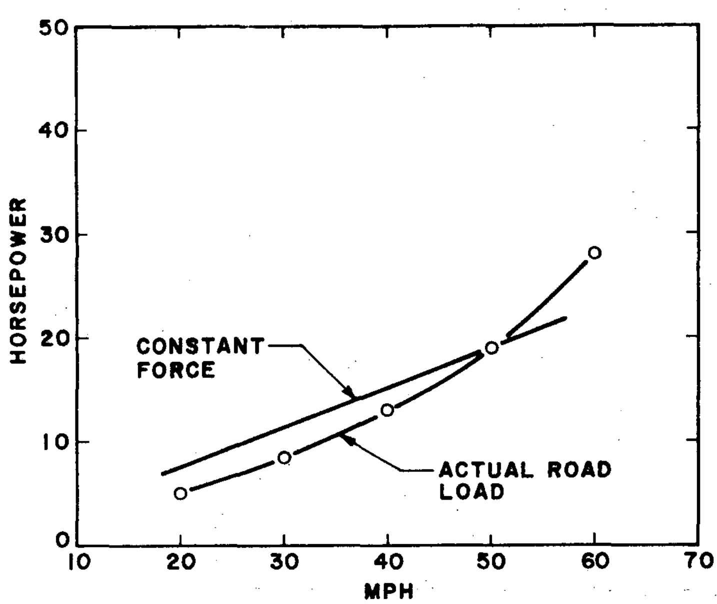


FIGURE 10. HORSEPOWER FOR CALCULATION PROCEDURE

at P = 19 hp and V = 50 mph, F = 142.5 lb.

The force required to accelerate the vehicle was determined by:

$$F_{accel} = \frac{W_{inertia\ wt}}{g} a$$

where

a = acceleration in ft/sec^2

g = gravitational constant

For an acceleration rate of 2.5 mph/sec and a 4500-lb inertia weight, this force was computed to be

$$F_{accel} = 524 \text{ lb}$$

The above two forces, or the variable forces for actual road loads, compromise the external load requirements of the vehicle. Translating these loads to driveline torque was accomplished by assuming a rolling radius of 1.07 ft and a rear axle ratio of 2.75:1. This then provided the transmission output torque requirement.

The losses in the automatic transmission were calculated on the basis of actual slip data taken from the tests conducted on the reference vehicles and the data of References 10 and 11. These loads in combination with the gear reduction of the transmission (2.5:1 first gear, 1.5:1 second gear, 1.0:1 third gear) produce the drive torque requirements of the engine. In the acceleration modes of the LA-4 cycle, it was assumed that 0-15 mph was negotiated in first gear and 15-25 mph in second gear. All cruises were in third gear.

In addition to the torque requirements, the N/V (engine rpm/mph) ratio in third gear was specified as 36 rpm/mph, consistent with the rolling radius and rear axle gearing. These specifications established the motive load torque and speed requirements of the engine.

The accessory torque is added to the drive load torque to produce the net torque requirement for the engine. Due to the limited speed range of the reference vehicle engine when operated on the LA-4 cycle, the accessory torque was also assumed constant. The torque value of 14.4 lb-ft was computed from accessory test data which indicated a total requirement of 5.5 hp at 2000 rpm.

Each operating condition for the vehicle (other than idle and deceleration) has been defined in terms of engine speed and torque requirements. The fuel consumption of the vehicle then is determined by the engine fuel consumption characteristics in meeting these operating conditions. The engine map shown in Section 2 (Figure 3) was chosen as representative of nonemission controlled spark-ignition engines. This map serves as a generalization and was reported in terms of bmep (brake mean effective pressure). To determine the bmep corresponding to the torque, the following equation was used:

[10] Jandasek, V. J., "The Design of a Single Stage Three-Element Torque Converter," SAE SP-186.
[11] *Design Practices* – Passenger Car Automatic Transmissions, Society of Automotive Engineers, Volume 3, 1973.

$$\text{Torque (ft-lb)} = \left[\frac{\text{bmep (psi)} \times \text{displacement (cu in.)}}{150.8}\right]$$

where the displacement of the reference vehicle engine was 350 CID. The map is also given in terms of piston speed so piston speeds were determined by:

$$\text{piston speed (fpm)} = 2 \cdot \text{stroke (ft)} \cdot \text{rpm}$$

for a 3.5-in. stroke,

$$\text{piston speed} = 0.583 \text{ ft} \times \text{rpm}$$

With these parameters, the brake specific fuel consumption was determined for each operating condition.

The fuel consumption during the idle and deceleration modes was assumed to be 6 lb/hr on the basis of test data acquired at SwRI on several 350 CID engines. For other evaluations in this report where the carbureted engine displacement was changed, the fuel consumption during idle and deceleration was assumed to be directly proportional to displacement. There is some evidence that the fuel consumption during idle and deceleration is higher than the value assumed; 1 gal/hr have been reported. Also, some calculations were performed using a fuel consumption of 4 lb/hr during idle and deceleration, and a noticeable effect on the mileage predicted by the calculation procedure was observed. However, the only values reported in this study are percentage improvements by comparison to a reference vehicle. The effect of a change in idle fuel consumption is trivial in this result if the same value for idle fuel consumption is used in both the reference vehicle and improved vehicle calculations.

The summed modal fuel consumption for the urban cycle was assumed to be expended over the standard 7.5 mile cycle length.

The calculations result in a series of economy numbers: Road load economy at 20 through 70 mph, and a composite figure for the urban cycle. A representative figure for the percentage of time spent in each operating mode could not be determined; consequently, on the basis of a recommendation from the Department of Transportation, the composite mileage of a vehicle was computed by assuming that 50 percent of all miles are driven under urban-type conditions and the remaining 50 percent of the miles driven are divided equally by operation at 20, 30, 40, 50, 60, and 70 mph.

The formula for overall economy then becomes:

$$(\text{MPG})^{-1}_{\text{avg}} = \frac{1}{2\,(\text{mpg})\text{LA-4}} + \frac{1}{12}\left[\frac{1}{(\text{mpg})20\text{ mph}} + \ldots + \frac{1}{(\text{mpg})70\text{ mph}}\right]$$

Following the completion of the analysis using the modal distribution listed above, other distributions were suggested as being more appropriate. Specifically, the following were recommended:

$$(\text{MPG})^{-1}_{\text{avg}} = \frac{1}{2\,(\text{mpg})\text{LA-4}} + \frac{1}{20\,(\text{mpg})40\text{ mph}} + \frac{1}{7.69\,(\text{mpg})50\text{ mph}} + \frac{1}{5.26\,(\text{mpg})60\text{ mph}} + \frac{1}{7.69\,(\text{mpg})70\text{ mph}}$$

$$(MPG)^{-1}_{avg} = \frac{1}{2\,(mpg)_{LA\text{-}4}} + \frac{1}{20\,(mpg)_{40\ mph}} + \frac{1}{7.69\,(mpg)_{50\ mph}} + \frac{1}{3.125\,(mpg)_{55\ mph}}$$

In both cases, it is assumed that half of the driving occurs on an urban cycle. The road load distributions correspond to:

5% at 40 mph	5% at 40 mph
13% at 50 mph	13% at 50 mph
19% at 60 mph	32% at 55 mph
13% at 70 mph	

A comparison was made between the percentage improvements predicted with these distributions and the improvement predicted by the distribution which assumes equal occurrence of speeds of 20, 30, 40, 50, 60, and 70 mph. The comparison is shown in Appendix F.

4. TEST PROGRAM DESCRIPTION AND RESULTS

A test program was devised to acquire information on the power demands and fuel consumption characteristics of the reference vehicles. Although one sample of each of several different automobiles is not sufficient to characterize the total vehicle population for the 1973 model year, the data do give insight into the characteristics of "typical" automobiles. Because of the small size of the sample of vehicles, and because association of numerical results with a particular model could lead to erroneous conclusions regarding the entire production of that model, the vehicles are identified by letter (A, B, C, D, E, F) throughout this report.

The literature gives little information on the design characteristics of recently produced automobiles. Since the dominant use of the American passenger automobile is at low load factors (road load), data on vehicle performance and operating requirements were necessary. The variable load demands of the Federal Driving Cycle also represent a "typical" commuter trip, thus operational data for this cycle were also necessary.

Prior to conducting any tests, 2000 miles were accumulated on each vehicle by following the break-in schedule specified below:

- First 100 miles on freeway access roads at speeds not to exceed 40 mph
- Second 100 miles on freeway access roads at speeds not to exceed 50 mph
- Next 300 miles on Interstate at speeds not to exceed 60 mph
- Remaining 1500 miles on Interstate at speeds not to exceed 70 mph.

The test program devised to provide the necessary information consisted of the following phases:

(1) Accessory power tests

(2) Road load fuel consumption and motive power requirements tests

(3) LA-4 chassis dynamometer tests

(4) Engine load/speed BSFC tests.

Accessory Power Tests

Each belt driven accessory was driven by a cradled electric motor (Figure 11) to determine its load/speed requirements. The horsepower requirements of the accessories under these conditions are accurately determined; however, testing a stationary vehicle does result in inaccurate simulation of actual road conditions for some accessories. For example, the engine fan is required to operate against a high static head since there is no relative air flow. At high fan speeds power determinations by this method will be inaccurate. An accurate road load fan test can only be performed in a wind tunnel. Data taken by the method employed at SwRI accurately represent LA-4 test cycle loadings. The air conditioner compressor also poses a problem in that the power requirements of air conditioning are dependent on the ambient temperature, the automobile internal heat load (both latent and sensible), and the condenser air flow, as well as the design characteristics of the particular

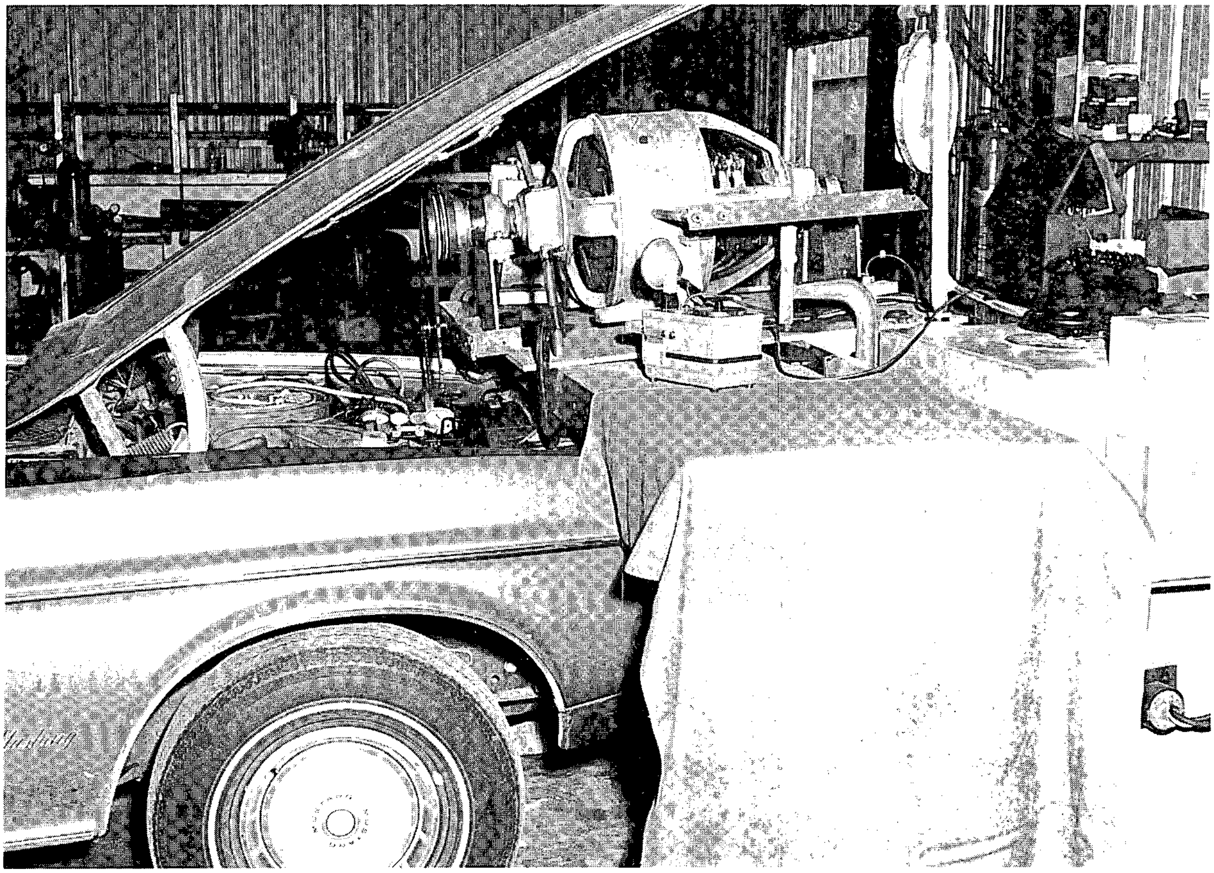


FIGURE 11. ACCESSORY POWER TEST APPARATUS

air conditioning system. To achieve the control necessary for calorimetric in-vehicle testing would have required a significant effort; consequently, it was decided that the compressor tests would be conducted in the controlled environment of the SwRI emissions laboratory. Thus, the compressor tests represent the engine loading experienced during operation on LA-4 (stationary vehicle) test.

Tests of the power steering pump represent the parasitic losses experienced when the wheels are not turned. Testing of the water pump represents the power requirement with the thermostat closed or open and under varied radiator pressures. The torque requirements were found to be insensitive to the above system conditions.

The alternator power requirements depend on the loading (power requirements) and condition of the electrical system. Data were acquired under varied wattage outputs, but a nominal value of 30 to 40 amps draw was chosen to represent a typical alternator characteristic for calculations used for this report.

Exhaust manifold air pumps were also tested on vehicles so equipped. Power requirements were found to be insensitive to position of the flow modulation valve.

Figure 12 illustrates the typical results of the accessory power testing for one of the full-size reference vehicles. Complete data are presented in Appendix B.

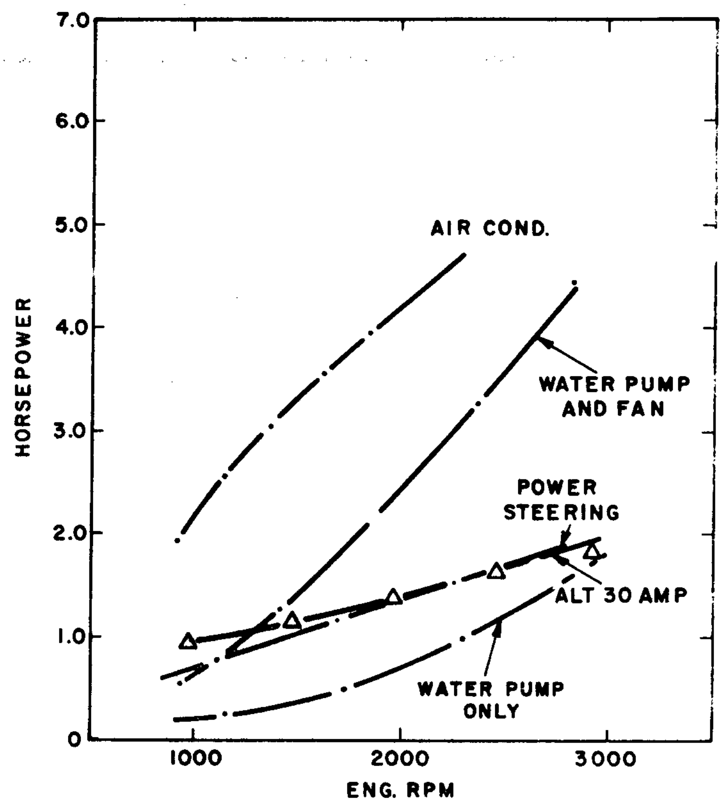


FIGURE 12. TYPICAL ACCESSORY POWER TEST RESULTS

Road Tests

Multiple road tests were conducted to determine road load fuel economy of each of the reference vehicles. The tests were conducted by a skilled test driver on a flat hard surface public road approximately 75 miles southeast of San Antonio, Texas. Tests were conducted in both directions over the course to minimize the influence of wind and grade loads. Tests were conducted at 20, 30, 40, 50, 60, and 70 mph. Each test was replicated at least five times. The individual mileage determinations at each test speed were within ±3 percent of the mean mileage determined at the test speed.

Mileage was determined by using a 1/10 gal burette fuel supply and by measuring the distance traveled through the road course by means of a fifth wheel. A typical vehicle installation is illustrated in Figures 13 and 14. With this information mileage can be easily calculated.

In addition to the mileage (mpg) determination, engine speed, vacuum, driveshaft speed, ambient temperatures, and air conditioning compressor suction and discharge pressures were recorded.

Figure 15 illustrates the mileage determined for the three full-size reference vehicles. Figure 16 illustrates the road load mileage of the sport-type vehicles. There is a remarkable spread in the fuel use efficiency of these automobiles. The differences in fuel economy are apparently due to basic

FIGURE 13. VEHICLE INSTRUMENTATION

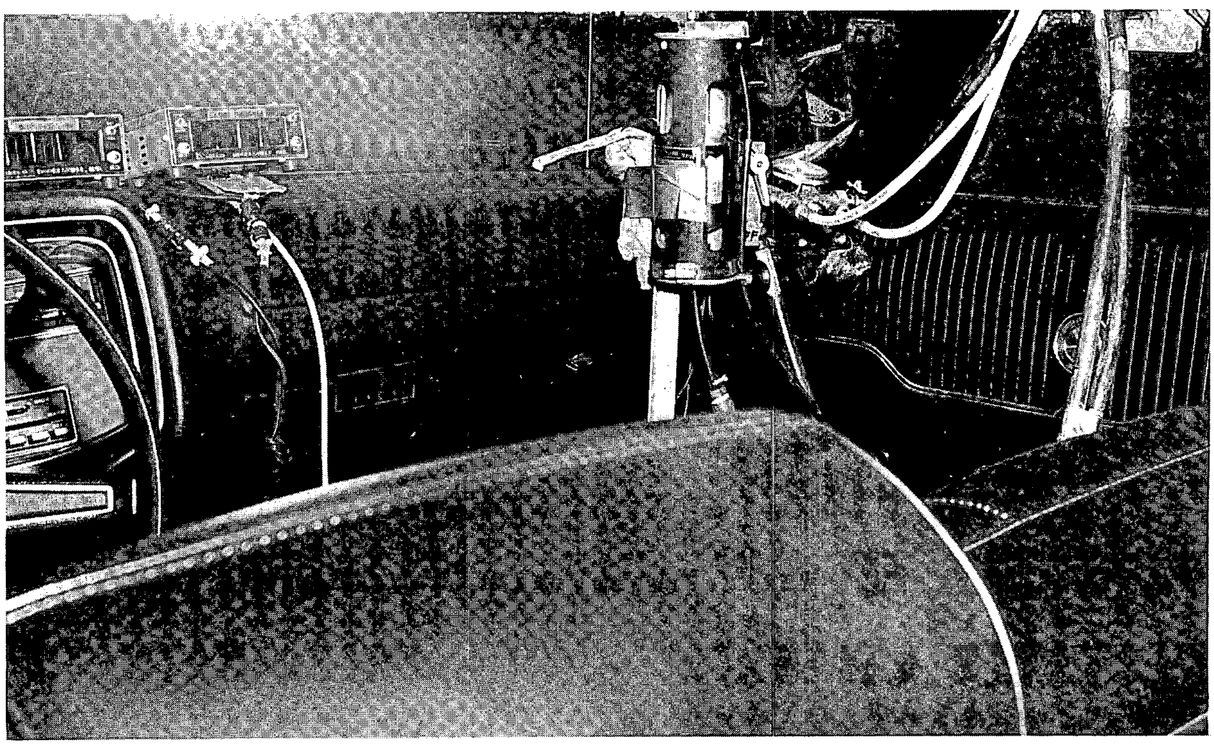


FIGURE 14. VEHICLE INSTRUMENTATION

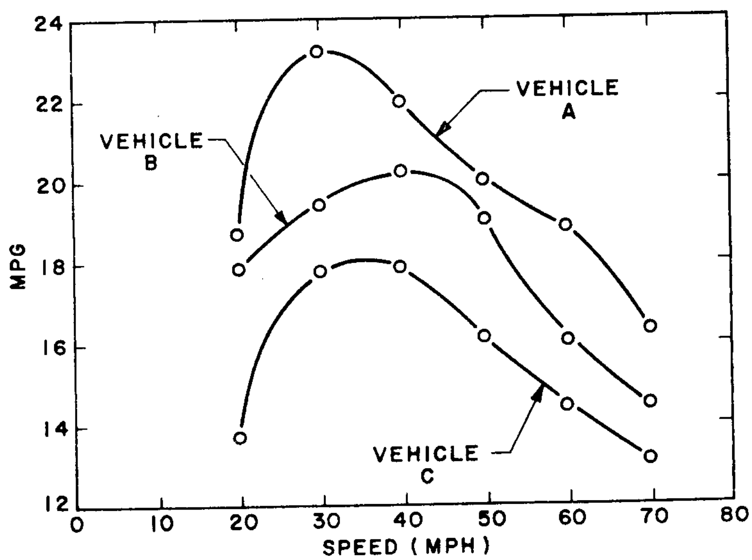


FIGURE 15. ROAD LOAD FUEL CONSUMPTION

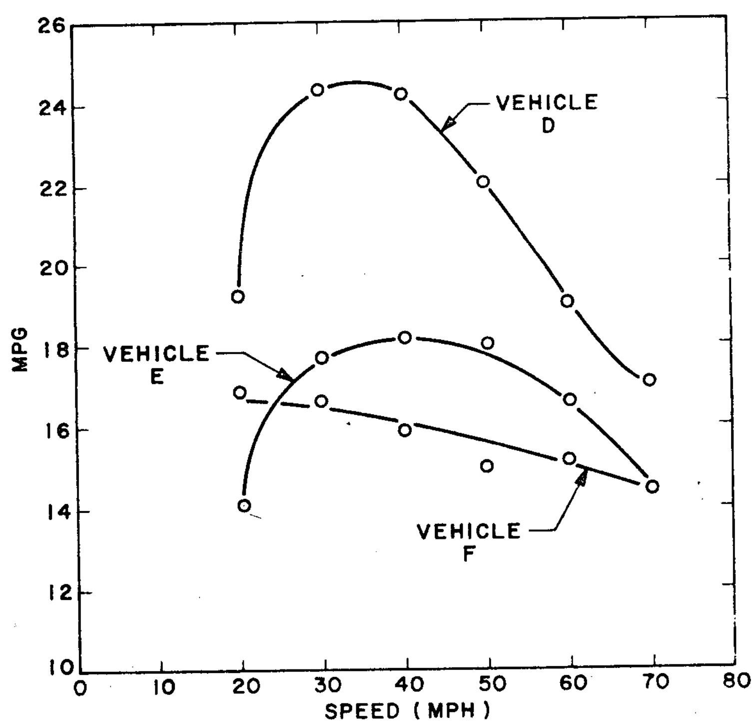


FIGURE 16. ROAD LOAD FUEL CONSUMPTION

differences in engine design and, more significantly, to differences in engine calibration for emission control. In addition, the difficulties associated with maintaining a precise load condition during the experiment have some effect on the results. Complete test data are given in Appendix C.

All tests were conducted during the summer in South Texas under "high" ambient temperatures (~90°F); consequently, the vehicle windows were down to allow an acceptable comfort level for the test crew. This action of course alters the aerodynamic drag of the vehicle but is representative of the action taken by any driver to maintain his comfort. The air conditioning system tests were also conducted with the windows down to provide the highest possible loading on the air conditioning system. This is not a typical operating mode and does not represent a condition that a consumer would see in practice. This approach was taken since the variable loading on the air conditioning system was detrimental to maintaining the desired vehicle speed. The net effect of this choice of test condition is to amplify the apparent penalty of air conditioning systems. A more representative fuel use penalty is advanced in the section on air conditioning systems. The data obtained from the tests, however, are useful from an engineering standpoint for evaluation of the systems.

Tests were also conducted to determine the motive power (road load horsepower) requirements of each vehicle. This was accomplished by measuring the driveshaft speed and input torque to the rear axle of each vehicle. Description of this instrumentation is included in the section describing the LA-4 tests.

Road horsepower requirements for the reference vehicles are illustrated in Figures 17 through 19. Although there are differences in the road horsepower values determined for vehicles in each class, these differences are not felt to be highly significant. The values presented should be regarded only as total road loads; sufficient detail is not present for separation of the total into component rolling and aerodynamic resistances.

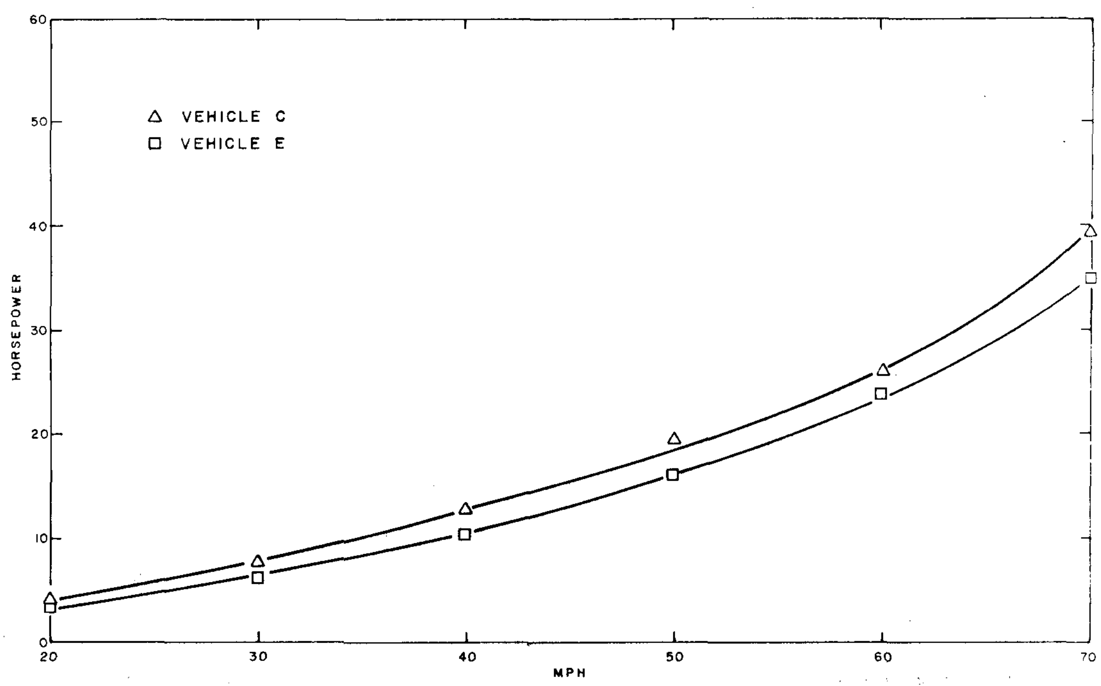


FIGURE 17. ROAD HORSEPOWER REQUIREMENTS

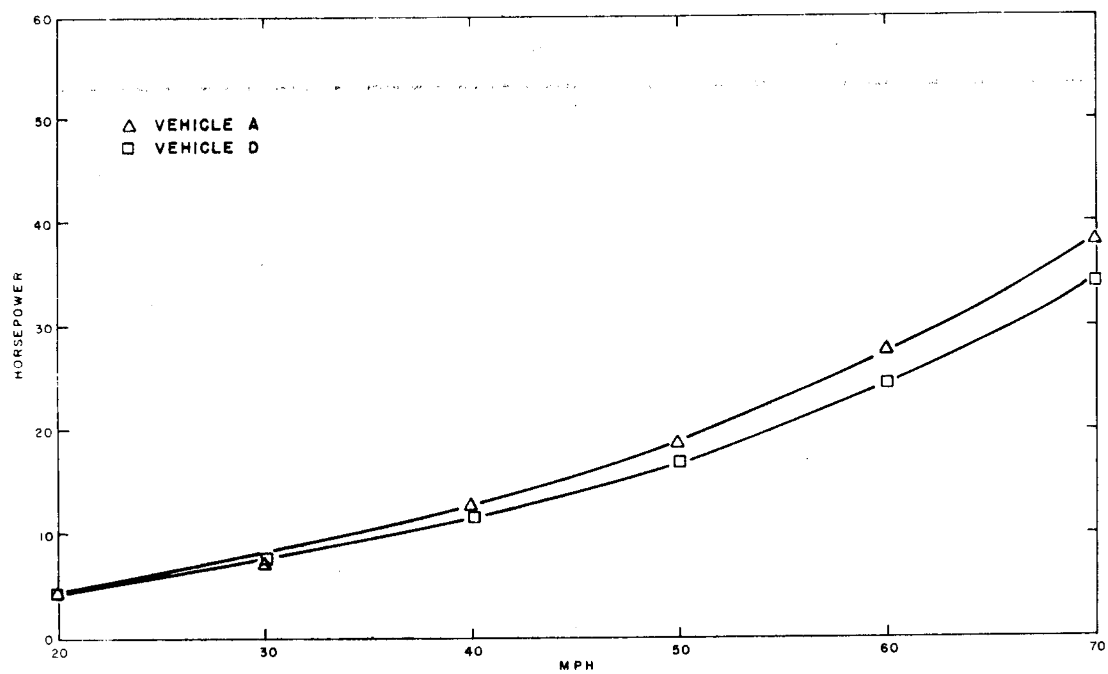


FIGURE 18. ROAD HORSEPOWER REQUIREMENTS

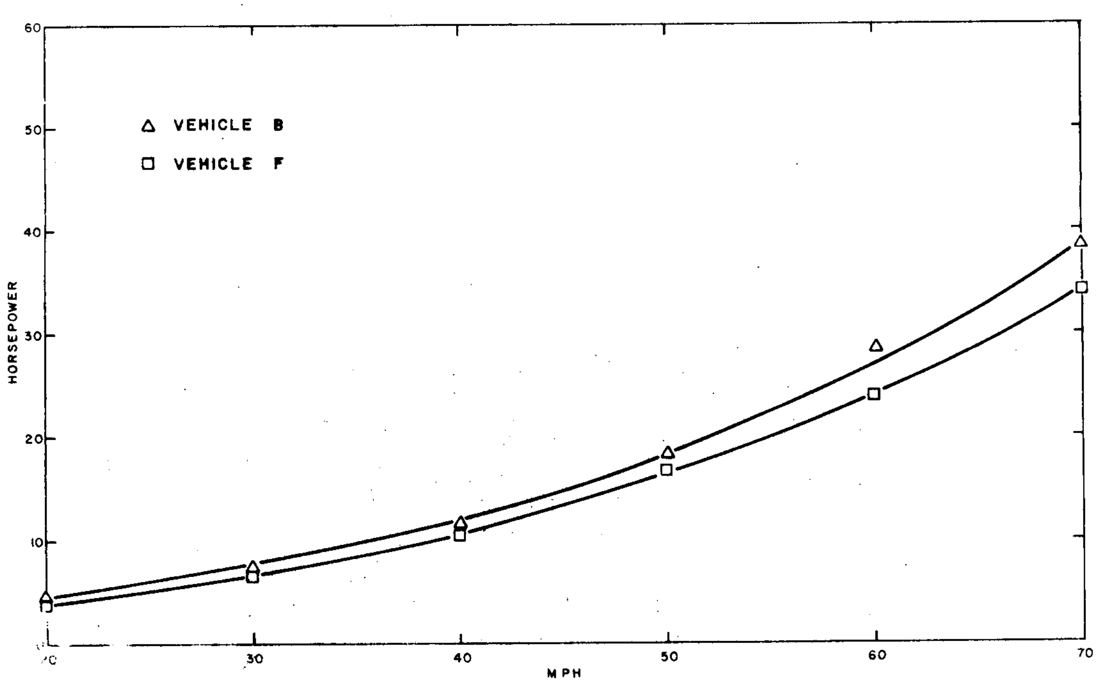


FIGURE 19. ROAD HORSEPOWER REQUIREMENTS

Testing of vehicles on the "level" road demonstrated that barely perceptible acceleration of the test vehicles can result in torque demands that are at least twice that required to sustain steady motion. Slight grades also favor increased torques as would be expected. On the whole, the steady load demand is a difficult quantity to achieve in practice, requiring strict attention to experimental detail.

In general, it is difficult for any driver to maintain the operation of a vehicle at a steady speed even on a level road. There are two causative factors, (1) the fact that road conditions are not constant and (2) current vehicles are designed with a large reserve power factor especially at the low road speeds. A slight movement of the accelerator pedal can result in a substantial increase in acceleration, consequently in power demand and fuel consumption. "Underpowered" vehicles (low power/weight) which operate at a larger percentage of full load are not subject to such operating difficulties, but, of course, such vehicles have much lower performance than the reference vehicles. The tires on each vehicle were checked and adjusted to the manufacturer's recommended pressures for the vehicle loading condition (two occupants and 200 lb of instrumentation) prior to travel to the test site.

TABLE 5. TYPICAL ROAD LOAD VALUES FOR VEHICLE B

Road speed (mph)	Engine speed (rpm)	Driveshaft speed (rpm)
20	815	646
30	1128	1016
40	1433	1353
50	1740	1689
60	2060	2020
70	2390	2353

Road load tests also provided useful information on the performance of the vehicle transmissions. The low torque values transmitted during road load operation are obtained at high efficiency levels as reflected by the high-speed ratio across the torque converter (driveshaft speed/engine speed). Typical road load values for vehicle B are given in Table 5; similar data for the other vehicles are located in Appendix C.

In the case of two vehicles (from the same manufacturer), the transmission would not allow a shift into high gear at 20 mph; consequently, engine speed at 20 mph was approximately 300 rpm higher. This may decrease the fuel economy under road load conditions since the engine torque requirement is decreased and operation is possibly at a point of higher BSFC. It should also be noted, however, that the determination of economy benefit or degradation could be obtained only by testing, and such tests were not performed. Conversation with the manufacturer indicated that the transmission setting preventing shifting to third gear at 20 mph was established as part of the emission control system for the vehicle.

LA-4 Chassis Dynamometer Tests

Six chassis dynamometer tests were conducted on each test vehicle to gather a variety of information. The normal LA-4 emissions test is a cold start test approximately 23 min long. This test was conducted followed by a 10-min period with the engine stopped. The 23-min test was then repeated followed by another 10-min "rest" period and then by another 23-min test. The tests were repeated again on another test day starting with another cold start. The "hot" tests, however, were conducted with the air conditioning system in the maximum cooling mode with vehicle windows open during the second series.

For each test, the chassis dynamometer was warmed up with an auxiliary vehicle by driving for a minimum of 15 min at 30 mph with the dynamometer set at the appropriate inertia. The dynamometer was then adjusted so that the horsepower setting specified in EPA regulations was obtained at 50 mph.

After calibration checks of the dynamometer were completed, the warmup vehicle was removed and the test vehicle was then pushed onto the chassis dynamometer. The rear tires of all vehicles were inflated to 45 psi to minimize the losses incurred by deflection over the dynamometer rolls. The driver then executed the cycle by following a preprinted speed-time trace on a strip chart recorder.

Driver training and familiarity with the test vehicle are necessary to drive the schedule within the EPA specified limits of ±1 sec or ±2 mph of the schedule. The drivers used during these tests were all trained drivers with experience in driving the LA-4 cycle.

The data acquired for each test consisted of engine speed, driveshaft speed, driveshaft torque, manifold vacuum, and fuel weight consumed versus time. Analog transducers for each parameter were installed and were FM-tape recorded on a Consolidated Electronics Corporation 11 channel recorder. All recordings were made at 1-7/8 in. per sec (ips). Subsequently, the analog signals were converted to digital data by playback at 60 ips to a Raytheon MADC15-05 A/D system. These data were then processed by a digital computer. The analysis of these data and some of its implications will be discussed later in this section. (All of the acquired data are not reported, however, typical results are shown in Appendix D.) Figure 20 illustrates the test setup for a typical test. No emission tests were conducted.

Speeds were obtained by pulse generators mounted on the vehicle. Engine speed was obtained from the ignition circuit, and driveshaft speed was obtained from a magnetic pickup which sensed the passage of four small magnets mounted on the driveshaft. Circuitry design limitations prevented accurate data below 375 engine rpm (an acceptable level) and 75 driveshaft rpm (approximately 2-3 mph).

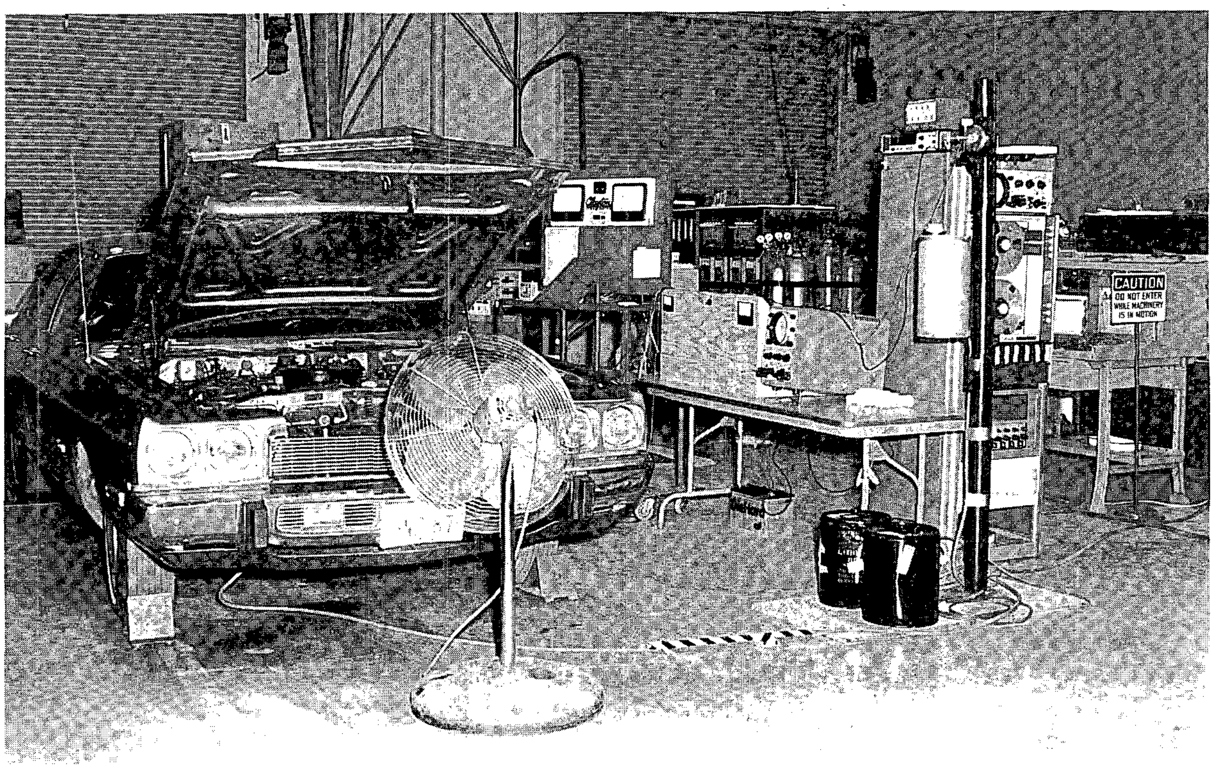


FIGURE 20. CHASSIS DYNAMOMETER TEST

Manifold vacuum was measured by a Celesco model PLC transducer excited and amplified by an Action Pak model 4051-217. The transducer was designed to measure positive gauge pressures; consequently, a calibration was performed using a U-tube manometer and vacuum pump to ensure linearity of the transducer.

Fuel consumption was determined by a system consisting of a fuel reservoir suspended from a 6-in. long aluminum bar. The bar was instrumented with four standard 120 ohm strain gauges. The bridge was then excited and amplified by an Action Instruments module 4051-217.

The rear axle input torque was monitored through a model MCRT-6-02T torquemeter produced by S. Himmelstein and Company.

The driveshaft of the test vehicle was modified to accept the torquemeter by removing a section of tubing and installing two flanges. The torquemeter was then mounted between the flanges. The torquemeter was a strain-gauge type using rotary transformers excited by an alternating current carrier amplifier (Hewlett Packard 8805A). The advertised accuracy of the unit is 0.2 percent; however, two end-to-end calibrations were performed (static and dynamic) to ensure transducer calibration. The torquemeter is shown in Figure 21 installed in its calibration driveshaft. This installation is in the dynamometer test stand for BSFC testing described later in the report.

All instrumentation was calibrated end-to-end to the tape recorder input. The system output was linear with respect to the parameter being measured over the complete range of each parameter.

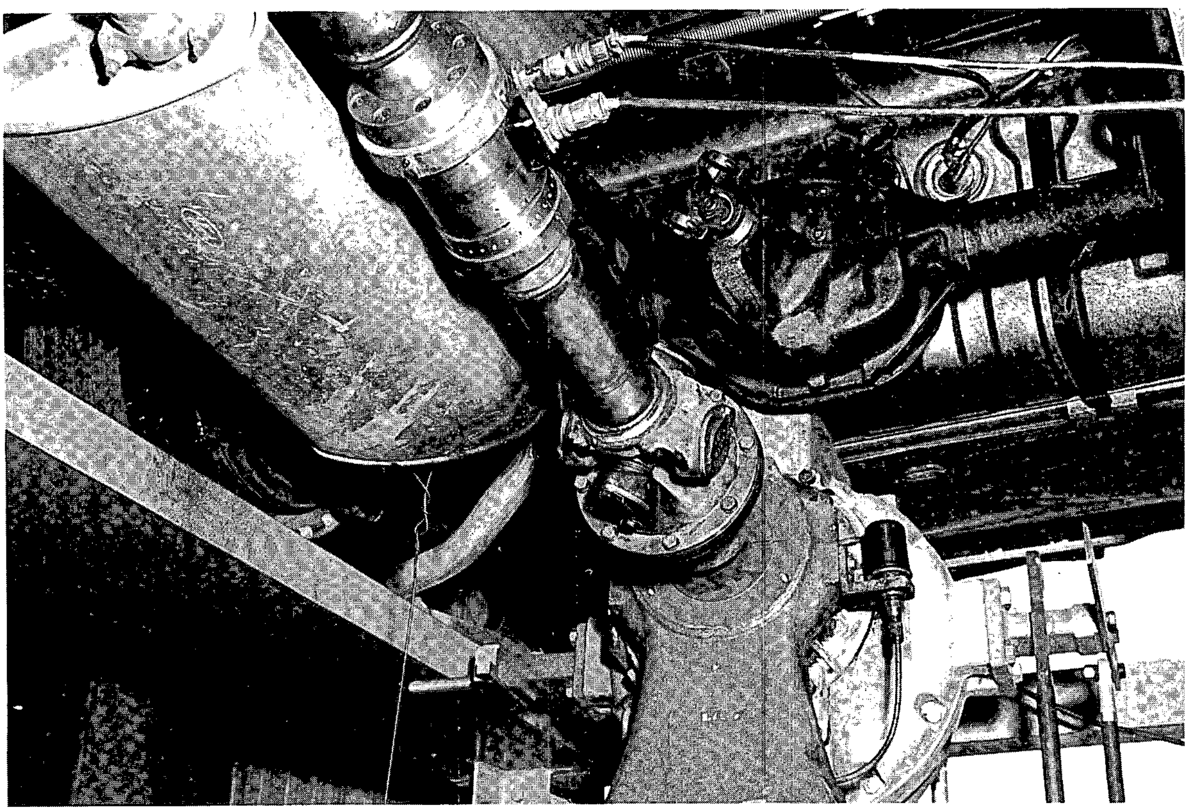


FIGURE 21. TORQUEMETER

The LA-4 cycle, due to its transient nature, represents the most complex combination of operational requirements examined during the test phases of this study. Figure 22 illustrates two engine parameters which are of the greatest significance in determining engine power output and fuel consumption. As can be noted from this Figure, the load and speed are continuously changing, thus, making instantaneous fuel consumption determinations a difficult proposition. To provide greater insight into the fuel use distribution, the cycle was broken into mode intervals over which fuel use was summed. In addition, the driveshaft rpm and torque (Figure 23) were also used to obtain the instantaneous motive power for the vehicle. These power computations were integrated over the corresponding intervals as a measure of the work done in moving the vehicle. Results for a 4500-lb inertia weight vehicle, during a cold start test with a load of 12.7 hp at 50 mph, are given in Table 6.

The acceleration work is 45 percent of the total motive work expended on the LA-4 cycle, and the cruise work is 55 percent of the total. Since there are periods of deceleration and idle where fuel is consumed that is not useful to the motion of the automobile, the percentage of fuel consumed by the acceleration and cruise modes will decrease. It is useful to note, however, that the *total* useful work done during all modes is higher than that indicated in Table 6, due to accessory power consumption. On a energy-use basis, the modal fuel use is given in Table 7.

The reference vehicles negotiate the acceleration ramps (~0-25 mph) of the LA-4 cycle by operating in first and second gear. All cruises and decelerations are performed in third gear. Due to the abrupt change in the shape of the speed required curve, the shift to third gear occurs at the top

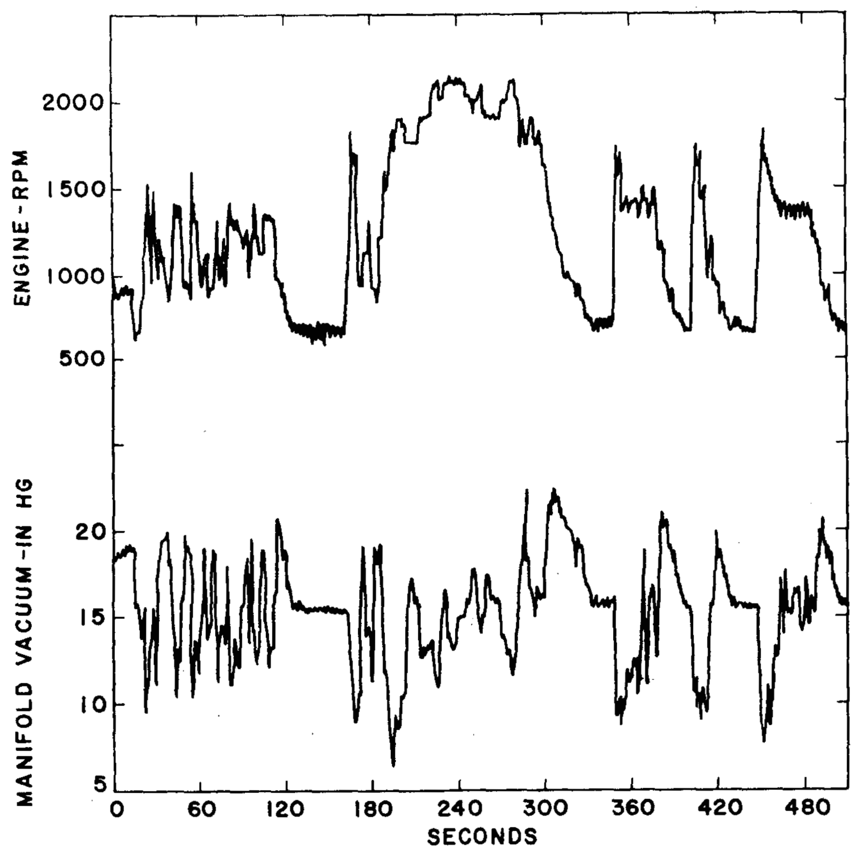


FIGURE 22. ENGINE PARAMETERS

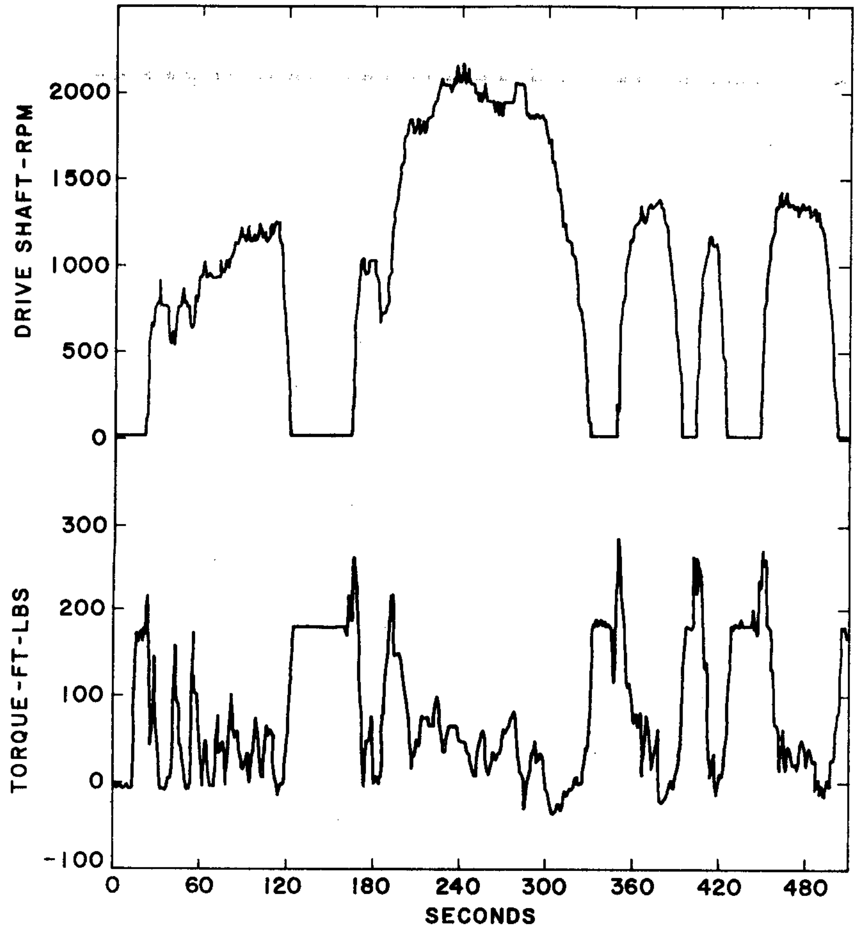


FIGURE 23. VEHICLE PARAMETERS

speed of each acceleration. The load demands of the LA-4 cycle are not excessive with respect to the performance capability of any of the reference vehicles. So little power is required relative to the potential of a given vehicle that engine speeds of about 2200 rpm are the maximum encountered. To further illustrate the power margin between the vehicle capability and the LA-4 demands, a "map" of the vacuum (load) and speed of a 3500-inertia weight vehicle with 350 CID engine was plotted from the test data (Figure 24). Roughly one-half of the maximum torque capability of the installed engine is utilized in negotiating the cycle.

TABLE 6. RESULTS FOR A 4500-LB INERTIA WEIGHT VEHICLE

Mode	Work (hp-hr)	Fuel Used (lb)
Acceleration	1.417	1.20
Cruise	1.736	2.27
Deceleration	---	0.36
Idle	---	0.34
Total	3.153	4.17

TABLE 7. MODAL FUEL USE

Mode	Percent of total LA-4 fuel use
Acceleration	29
Cruise	54
Deceleration	9
Idle	8

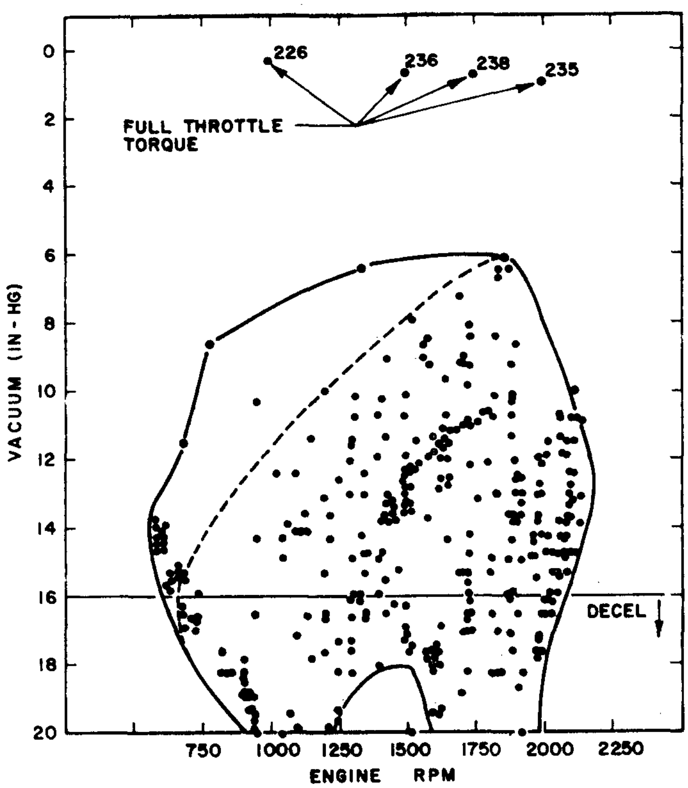


FIGURE 24. LA-4 CYCLE, 350 CID V8 2.73 TO 1 REAR AXLE, 3500-LB INERTIA WEIGHT

The performance capability of the 3500-lb inertia weight vehicle is superior to the 4500-lb inertia weight vehicle. Since the motive loads are lower with the smaller inertia weight vehicle, the fuel consumption is obtained at points of higher brake specific fuel consumption than the minimums achieved by the larger vehicle; this partially offsets the gains obtained by lowering the inertia requirements. Leaving the engine size unchanged as the inertia weight is lowered provides little economy gain as reflected in Table 8. These test results are also confirmed by analytical predictions such as advanced by Ambs (Reference 12) and as illustrated in Figure 25.

As weight is lowered, the engine size must be decreased to prevent an increase in performance capability.

One final factor of interest obtained from the LA-4 test program is the influence of "cold" starts versus "hot" starts. This is illustrated in Table 9.

The influence in commuter type traffic for warmed-up driving versus cold starts can be seen to be as significant an effect as a 1000-lb inertia weight change in the automobile. Although all of the percentage gain due to warmup is not obtainable, the potential significance of the improvement merits the further consideration of methods which will improve warmup economy.

TABLE 8. COMPARISON OF INERTIA WEIGHT EFFECTS ON LA-4 ECONOMY

Vehicle	Inertia wt (lb)	Cold start test (mpg)	Air conditioner off (mpg)	Average mpg
A[a]	4000	11.52	11.92	11.72
D[a]	3500	12.59	12.43	12.51
C[b]	4500	10.97	10.34	10.66
E[b]	3500	12.09	11.16	11.63

(a) 12.5% reduction in wt results in a 7% improvement.
(b) 22.2% reduction in wt results in a 9% improvement.

TABLE 9. INFLUENCE OF "COLD" STARTS VERSUS "HOT" STARTS

(Mileages Are Average of Two Runs)

Vehicle	Cold start	Hot start	Percent improvement in mileage
A	11.72	12.51	6.7
B	10.03	10.64	6.1
C	10.65	12.00	12.6
D	12.51	13.59	8.6
E	11.63	12.19	4.8
F	9.43	9.85	4.4

[12]Ambs, L. L., "Passenger Car Design Influences on Fuel Consumption and Emissions," Paper 739113, 8th Intersociety Energy Conversion Engineering Conference Proceedings, University of Pennsylvania, August 13-17, 1973.

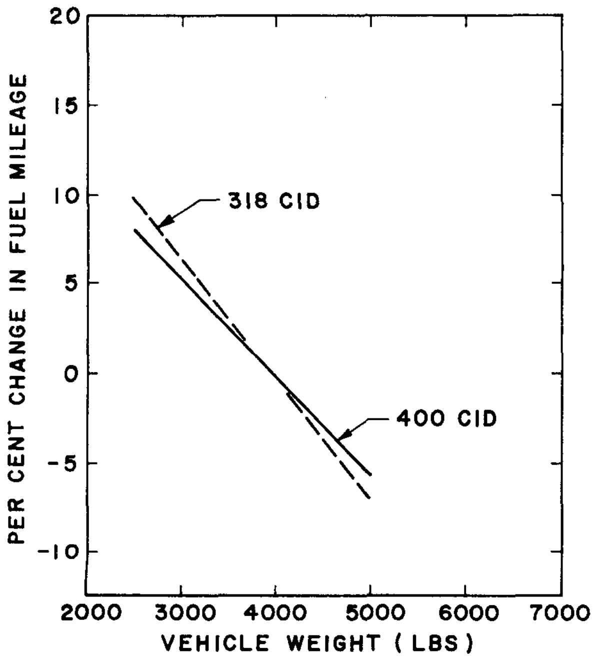


FIGURE 25. RESULTS OF ANALYTICAL PREDICTION

Specific information concerning engine warmup during LA-4 cycle operation is provided in Chapter 23 of this study; it may be observed that the engine cooling system achieves a fully warmed-up condition after about one-third of the cycle. However, it should be noted that the fuel economy associated with the warmed-up condition cannot be achieved on the road until all vehicle components, particularly those which are lubricated, reach an appropriate temperature.

Fuel economy is generally poorer with a cold engine due to the fuel system compromises which must be made to assure operation. A choke is typically employed to provide a fuel/air ratio of approximate unity in order to allow sufficient fuel vaporization. The choking function is diminished automatically as the engine warms up. Both the engine and other vehicle components, such as gears, are affected by lubricant viscosity. Until sufficient energy has been imparted to achieve the relatively low operating viscosity, large friction losses must be endured in these components.

Engine Load/Speed BSFC Tests

In any computational procedure for estimating fuel economy of a vehicle, it is necessary to use engine fuel consumption data for conditions which meet the load and speed demands of the vehicle operation. The calculation procedure described earlier explained these relationships. Data on the performance of the 1973-reference vehicle power plants were not available; consequently, tests were conducted to assess the state-of-the-art.

To facilitate this type of testing, which is highly time consuming, the engines were left in the vehicles. This obviated the need for removal and installation time. The vehicle was positioned on a rack, the automatic transmission and flex plate were removed, and a flywheel was installed. The flywheel was then coupled to a Midwest Eddy Current dynamometer through a two-piece driveshaft to complete the setup. Load was controlled by an external servo system which maintains an established load automatically. Water temperature was controlled to 190 ± 10°F by an external heat exchanger, and fuel consumption was monitored by the breaker and balance method. The test setup for these tests is shown in Figure 26. No vehicle was tested until at least 4000 miles had been accumulated. Prior to testing, ignition timing and dwell were checked and were found to be within manufacturers specifications.

All engines (vehicles B, D, E, F) were run with power steering, water pump, fan and air pumps operating. The alternator was disconnected to eliminate variable loading, and battery voltage was maintained by an external source. Tests were conducted at 1000, 1250, 1500, 1750, 2000, 2500, and 3000 rpm with approximately eight loads replicated at each engine speed. On one engine, some of the higher loads could not be obtained since excessive exhaust manifold temperature repeatedly

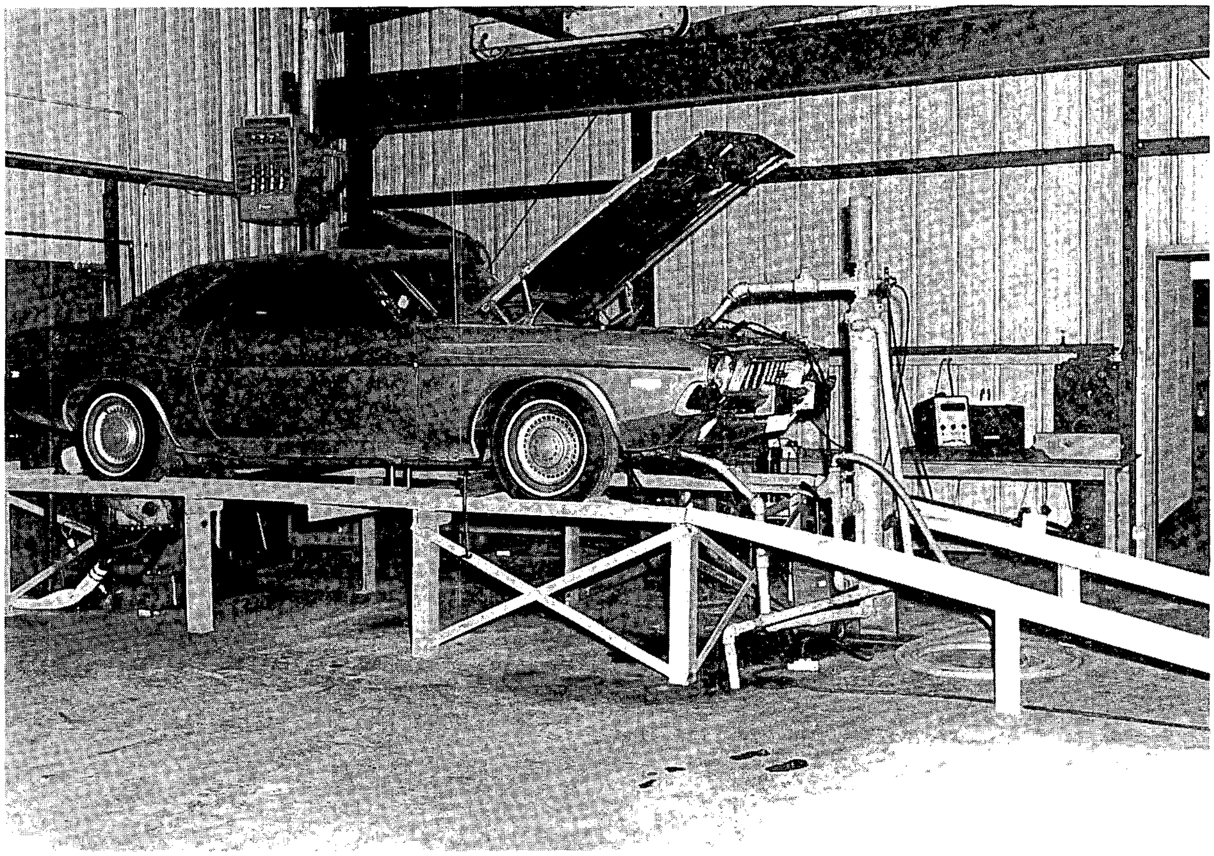


FIGURE 26. ENGINE TEST APPARATUS

ignited the sparkplug wires. At the lower loads, representative of typical driving conditions, no problems were encountered with the engine.

Figure 27 illustrates some of the data acquired on three different 350 CID engines illustrating that substantial differences in brake specific fuel consumption are evident. The reference speed of 1500 rpm will produce a road speed of about 42 mph in the reference vehicles, near the point of maximum road load economy. Since the road loads are approximately equal, the difference in fuel consumption should largely be due to the difference in engine performance. At a load of about 45 ft-lb, the difference in BSFC is 1.05 versus 0.95 lb/hp-hr or 10 percent greater for vehicle C. Referring back to Figure 15, at 40 mph the difference in mileage is 18 versus 20 mpg or 10 percent. The predominant difference in fuel consumption then is attributable to the fuel use efficiency of the engine.

As a further example vehicle F is a "sporty" car and has a lower road horsepower requirement (10.5 hp versus 13 hp); however, a comparison of Figure 15 and Figure 16 illustrates that the mileage of vehicle F is substantially lower than that of either vehicles B or C. The BSFC curve for vehicle F indicates a BSFC of ~1.38 lb/hp-hr at 38 ft-lb of torque, corresponding to 40 mph road load. Calculations with the data demonstrate that over 75 percent of the difference in mileage between these vehicles can be attributed to engine operational characteristics, that is, the difference in calibration of the various engine systems when the engines are used in specific vehicle models.

At a constant torque of 45 ft-lb (full-size reference vehicle at ~40 mph), the spread in mileage will be 21 percent from the "best" to the "worst" 350 CID engine tested. These engines represent

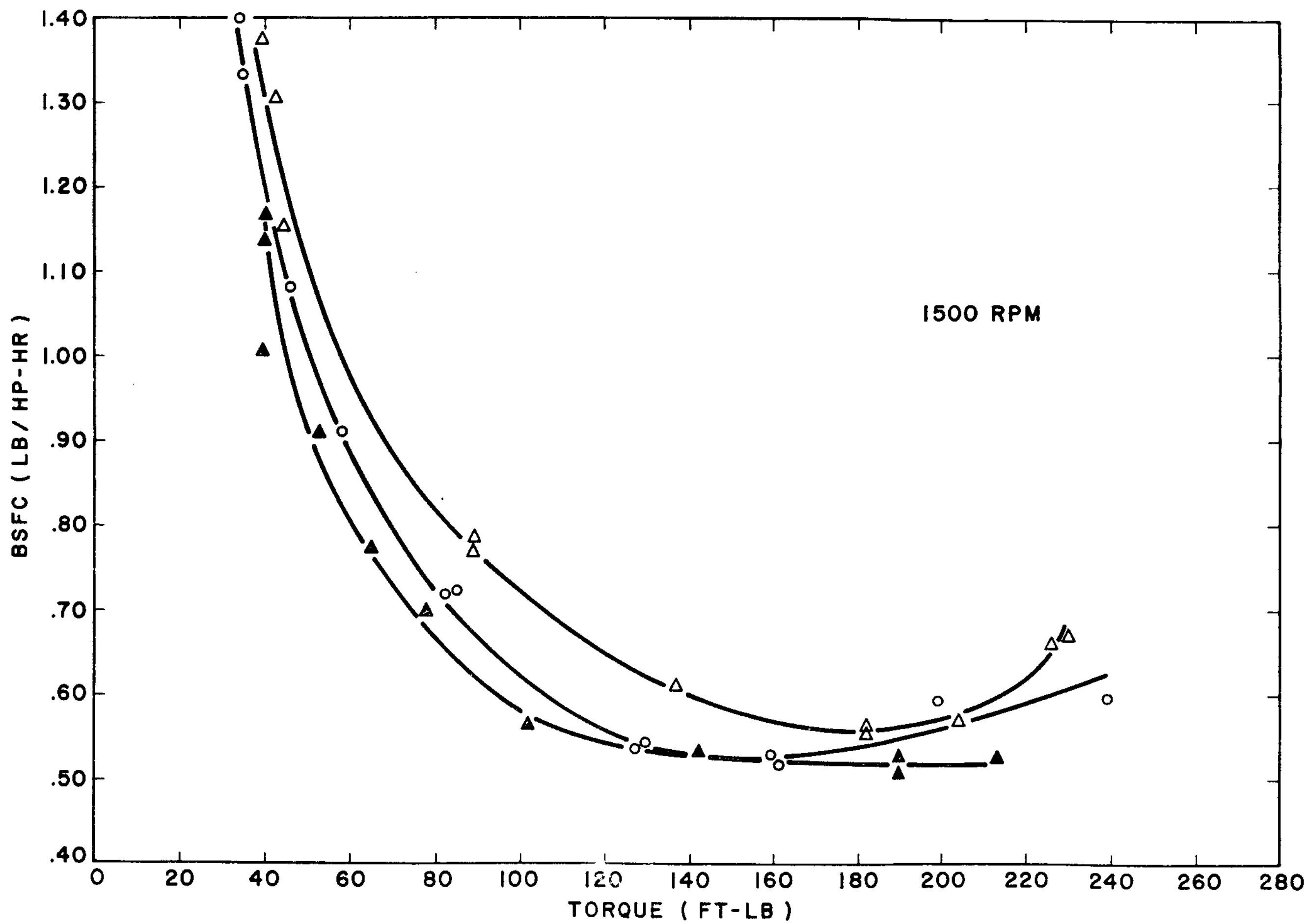


FIGURE 27. DATA FROM THREE ENGINES

only single samples, however, this information does indicate that precise information on the fuel use of the specific engine design, in meeting the load demands, is necessary to establish an accurate mileage estimate.

Another point to consider while reviewing the operation of the engine is the influence of emission controls. The specifics of why various design changes have been incorporated are covered later in this report; however, the significance of one parameter on one engine was briefly tested under this test phase. Figure 28 illustrates the improvement in fuel consumption obtained on the specific test engine by reducing the amount of exhaust gas recirculation to zero. On this specific engine, the improvement would be a 16-percent reduction in fuel consumption under road load conditions at 40 mph. Fuel consumption benefits at higher loads are also evident.

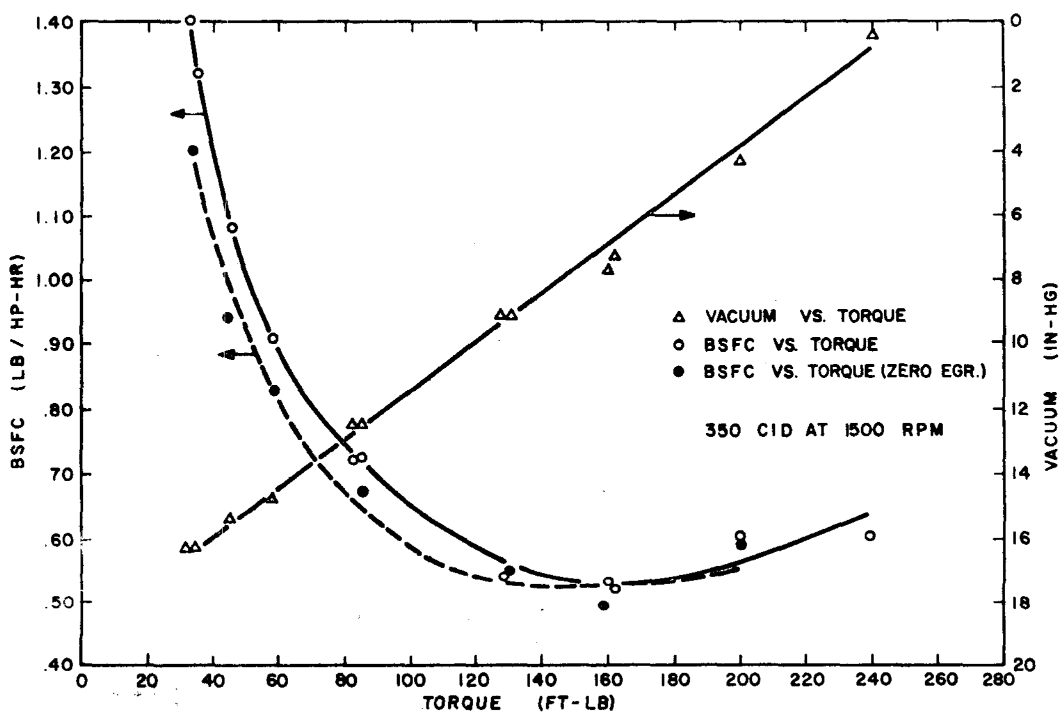


FIGURE 28. ALTERATION OF EXHAUST GAS RECIRCULATION

5. THE COST TO OWN AND OPERATE AN AUTOMOBILE

Methodology for the Analysis of the Cost to Own and Operate an Automobile

Notwithstanding the widespread ownership and usage of American automobiles, there exists no generally accepted method of analyzing the costs of automobile ownership. Therefore, it is necessary to define the methodology by which automobile ownership and operation costs are developed and presented in this study. The methods that are described and used in this analysis draw heavily from prior studies of a similar nature.

The most widely disseminated series of reports of automobile costs have been done by the U. S. Department of Transportation.(13,14,15) In addition to distribution through normal government publications, the results of these studies have been republished in *Consumers Report.*

Consumers Report, in addition to publishing DOT study results, began in January 1973 to include in its evaluation of automobiles "an estimate of typical costs of new car ownership over the first two years of operation."(16) To quantify these economic considerations, many of the assumptions from the DOT studies were used along with fuel consumption data derived from actual driving tests conducted by Consumers Union.

The estimates of automobile ownership and operation costs in this Analysis are based upon: (1) fuel consumption data developed from testing at SwRI; (2) automobile usage assumption from the DOT studies; and (3) all costs analyzed on a pro forma basis discounting future costs.

TABLE 10. CATEGORIZATION OF THE VARIOUS COST COMPONENTS OF AUTOMOBILE OWNERSHIP AND OPERATION

Cost included in analysis	Cost excluded in analysis
• Original vehicle cost • Repair and maintenance • Replacement tires • Gasoline • Oil	• Owner purchased accessories • Insurance • Garaging and parking • Toll fees • Interest lost • Registration & titling

The Cost Included

This analysis of cost of owning and operation of an automobile includes only those costs that are not sensitive to geographic location or consumer use. Table 10 categorizes all of the costs into two groups. The cost items excluded are those that are subject to owner discretionary spending, the specific use of the automobile, and the owner's financial circumstances. The cost items included in the analysis provide a financially conservative estimate of the cost incurred in owning and operating the automobile.

Automobile Lifetime/Mileage Assumptions

In addition to the specific component of costs that are discussed in subsequent paragraphs, there are several assumptions that must be made to ensure an equitable comparison of alternatives and, at the same time, maintain an accurate representation of consumer costs.

[13] U.S. Department of Transportation, Federal Highway Administration, Office of Highway Planning, Highway Statistics Division, *Cost of Operating an Automobile,* April 1972, by L. L. Liston and C. L. Gauthier (Washington, D.C., Government Printing Office, 1972).
[14] U. S. Department of Transportation, Federal Highway Administration, Bureau of Public Roads, *Cost of Operating an Automobile,* February 1970, by E. M. Cope and C. L. Gauthier (Washington, D. C., Government Printing Office, 1970).
[15] Similar studies were published in 1960 and 1968.
[16] "The 1973 Autos," *Consumers Report,* April, 1973.

TABLE 11. DISTRIBUTION (%) OF 100,000-MILE LIFETIME OVER 10-YR INTERVAL

Year	1	2	3	4	5	6	7	8	9	10
%	14.5	13.0	11.5	10.0	9.9	9.9	9.5	8.5	7.5	5.7

Source: DOT, **Cost to operate an Automobile 1972.**

Note: International Research and Technology Corporation in its study **Economic Impact of Mass Production of Alternative Low Emissions Automotive Power System** dated March 1973 used an automobile lifetime of 10+ years and 112,000 miles.

TABLE 12. THE ORIGINAL COST FOR REFERENCE VEHICLES IN 1973 DOLLARS

Description of car (1973)	Estimated original cost (dollars)
Group I full size 4-door sedans	
Ford–Galaxie 500	3880
Plymouth–Fury III	3900
Chevrolet–Impala	3905
Group II sporty 2-door hardtops	
Ford–Mustang	3655
Dodge–Challenger	3750
Chevrolet–Camaro	3700

Source: Southwest Research Institute.

The assumed automobile lifetime and annual driving patterns are taken from the 1972 DOT *Cost to Operate an Automobile*. The automobile has an assumed lifetime of 10 yr and a total lifetime mileage of 100,000 miles distributed as illustrated in Table 11.

Original Purchase Price

The reference vehicles tested during this project are divided into two groups: (1) "sporty" cars and (2) full-size cars.

The estimated original purchase price for each automobile (1973 dollars) is shown in Table 12.

An estimated cost of $3900 is used in this analysis as the estimate of the original cost for full-size sedan; this cost includes:

- Four-door sedan,
- Eight-cylinder engine equipped with 2-barrel carburetor,
- Automatic transmission,
- Power brakes,
- Power steering,
- Air-conditioning,
- 1973 increase (average) in base cost as approved by Cost of Living Council,
- Additional 20 percent of estimated dealer costs to allow for dealer markup (5 to 10 percent), destination charges (4 to 5 percent), makeready (1 to 2 percent), and optional equipment (10 to 15 percent).

An estimated cost of $3700 is used in this analysis for the group of "sporty" type automobiles; this cost includes:

- Two-door hardtop body,
- Eight-cylinder engine, 2-barrel carburetor,

- Power steering,
- Power brakes,
- Automatic transmission,
- Air-conditioning,
- 1973 increase (average) in base cost approved by Cost of Living Council,
- Additional 20 percent of estimated dealer cost to allow for dealer markup (5 to 10 percent), destination charges (4 to 5 percent), makeready costs (1 to 2 percent), and optional equipment (10 to 15 percent).

No salvage value is considered at the end of the 10-yr lifetime. It may be argued that the vehicle still has a value of 3 to 5 percent of its original cost, but this economic benefit to the last owner will be virtually offset by the cost to dispose of the automobile.

Repair and Maintenance

The 1972 DOT *Cost of Operating an Automobile*, estimate of repair and maintenance costs, appears to be the best data source. This cost item is compared in Figure 29 with the 1970 estimate of repair and maintenance costs. (The 1970 estimate has been adjusted by the change in the private auto repair and maintenance consumer price index.) The only significant variation in cost per mile is in the years numbered 4 and 9 when the 1972 estimates reflect additional one time repair.

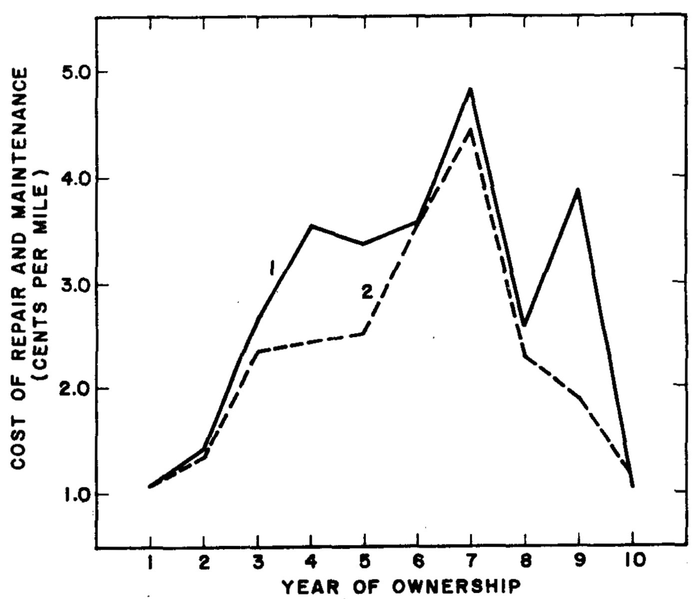


FIGURE 29. COMPARISON OF ESTIMATED COST TO PROVIDE REPAIR AND MAINTENANCE FOR FULL SIZE AUTOMOBILE ADJUSTED TO 1973

Stanford Research Institute conducted a year-long survey of over 3600 vehicles in 1968 for a consortium of clients. Summary data from the proprietary study, *Opportunities in the Changing Automotive After Market*, were entered as testimony before investigation of the Automotive Repair Industry hearings conducted by the Subcommittee on Antitrust and Monopoly (U.S. Senate).(17) While the data are judged to be accurate and valid, the results (adjusted for the different years) were considerably LESS than DOT estimates. In SRI's survey, the maintenance items listed are comparable to four line items in the DOT studies: (1) repair and maintenance, (2) tires, (3) oil, and (4) accessories. A second variation in the study methods contributes to the difference of results. The Stanford study was a cross section of all automobiles age 0 to 9 during a given calendar year;

[17] U. S. Congress, Senate, *Hearings, Automotive Repair Industry*, 90th and 91st Congress, 1969, pp. 3548-63.

whereas, the DOT study method is for one car for 10 yr. If there was a significant variation in the maintainability of more recent autos (which is generally true), then the Stanford results necessarily reflect a lower average cost per vehicle.

The 1972 DOT study results for Repair and Maintenance are used. The repair and maintenance cost per mile have been adjusted by the change in the consumer price index, i.e., the 1972 average private auto repair and maintenance index = 135.1, and the average for the first 5 months of 1973 = 140 (a 3.6-percent increase).

Replacement Tires

The life expectancy of the original equipment tires is estimated to be 25,000 miles. The new car is equipped with five of these tires providing a total of 125,000 miles of tire life. This total life divided by four tires on the ground yields an estimated 31,250 miles of travel before replacement is required.

It is assumed that replacement tires are bought in sets of four every 28,000 miles(18) after the initial replacement. With these assumptions, replacement tires would be required in the third year at 31,250 miles, in the sixth year at 59,250 and in the ninth year at 87,250 miles. At the end of the 10-yr period, 50 percent of the tread on the last set of tires would be remaining. No salvage value is considered, since no provision was made in the change out schedule for a spare tire beyond the time when the original equipment is in use.

Cost of Motor Oil

The assumed consumption rate of oil is 1 qt/1000 miles driven. This consumption rate is derived from an estimated usage of 5 qt/oil change every 6000 miles, plus an additional quart of makeup between each regular oil change. The DOT study used a cost of 49 ¢/qt. For this study, a cost of 67 ¢/qt is used.(19)

Cost of Gasoline

The 1972 DOT study assumed a cost of fuel at 26.9 ¢/gal (added to this is 7 ¢/gal state tax and 4 ¢/gal Federal tax) for a total of 37.9 ¢/gal. The maximum 3 ¢/gal increase in retail price that was allowed by the Cost of Living Council during 1973 is added to the 37.9 ¢/gal to update the total cost of gasoline to 40.9 ¢/gal. The estimate of automobile ownership and operation cost published by *Consumers Report* in 1973 assumes the same price of 40.9 ¢/gal for gasoline. In addition to the 1973 40.9 ¢/gal, two alternative costs of gasoline are used: [1] the 40.9 ¢/gal increased by 5 percent/yr for the 10-yr period (the tenth year cost would then be 53.4 ¢/gal), and [2] a fixed 10-yr price of 50 ¢/gal.

At the time of this writing, officials of the Federal Energy Administration are predicting an 11¢/gal increase to the average 43.9 ¢/gal cost of gasoline within 2 months.

[18] The price of Sears 2 polyester cord ply, 2 fiberglass belted F78-14 tires is used in this analysis. The Sears 1973 Fall and Winter catalog price is $37.94, plus $2.01 Federal excise tax and $1.30 shipping, for a total cost of $41.25 per tire. These tires are guaranteed for 28,000 miles.

[19] The estimated cost of 67 ¢/qt of oil is derived from the Sears 1973 Fall and Winter catalog price for 10 W-40 oil, plus shipping charges.

In lieu of predicting future gasoline costs, the three alternative gasoline prices are used because they yield a financially conservative estimate of the additional new car costs that could be incurred to achieve fuel conservation.

If it is found that the overall acceptability of a particular fuel economy innovation hinges on this cost factor alone, then a more detailed analysis will be conducted.

Analysis of the Estimated Cost to Own and Operate an Automobile

The basic assumptions and definition of the reference vehicles have been previously established and include: (1) the lifetime and annual driving pattern, (2) the vehicle initial cost, (3) the anticipated repair and maintenance cost, (4) the cost of replacement tires, and (5) the cost of motor oil. These costs are tabulated and summarized in Table 13.

TABLE 13. NON-VARIABLE COSTS TO OWN AND OPERATE A 1973 REFERENCE VEHICLE–BY YEAR

Item	Year											Total
	0	1	2	3	4	5	6	7	8	9	10	
Miles driven (thousands)		14.5	13.0	11.5	10.0	9.9	9.9	9.5	8.5	7.5	5.7	100.0
Initial cost	$3900.00											$3900.00
Repair & maintenance		$84.10	$119.60	$251.85	$306.00	$285.12	$302.94	$412.30	$177.65	$253.50	$30.21	$2223.27
Replacement tires				165.00			165.00			165.00		$495.00
Motor oil		9.72	8.70	7.70	6.70	6.63	6.63	6.37	5.70	5.03	3.82	67.00
Total	$3900.00	$93.82	$128.30	$424.55	$312.70	$291.75	$474.57	$418.67	$183.35	$423.53	$34.03	$6685.27

The analysis of the cost to the consumer to own and operate a 1973 reference vehicle (full-size, four-door sedan) using the methodology heretofore developed must be examined to identify its sensitivity to three remaining variables. These variables are (1) the discount rates for reducing costs to present worth, (2) the cost of gasoline, and (3) the fuel consumption rate or automobile mileage.

Discount Rate of Future Costs

To examine the sensitivity to variations of the rate at which the future costs are discounted to present value, a quantity for each of the variables (gasoline cost and mileage) must be selected and held constant as the discount rate is allowed to change. The suggested values of discount rate shown in the Statement of Work used are 0, 6, 10 and 18 percents.

Time value factors for annual compounding at an effective rate (i) per year applied to the end of year amounts are used. These rates may be derived from the appropriate tables or the expression;

$$\text{Present Worth of Future Cost} = \frac{1}{(1 + i)^n}$$

For this examination of the variations caused by changing discount rates, the assumed cost of gasoline is \$.409/gal and the fuel consumption rate is 14 miles/gal. Using these assumptions, the total cost of automobile ownership and operation for the four discount rates is tabulated in Table 14.

From the data illustrated in Figure 30, the total cost of automobile ownership, with the specific assumptions and operating patterns described, may be expressed as a value of \$9607 to \$6531,

TABLE 14. EFFECTS OF DISCOUNT RATES ON EVALUATING TOTAL COST OF AUTOMOBILE OWNERSHIP

Item	Year 0	1	2	3	4	5	6	7	8	9	10	Total
Nonvariable costs	$3900.00	$ 93.82	$128.30	$424.55	$312.70	$291.75	$474.57	$418.67	$183.35	$423.53	$34.03	$6685.27
Gasoline costs[a]		423.60	379.79	335.95	297.15	289.20	289.20	277.55	248.30	219.10	166.50	2921.34
Total	$3900.00	$517.42	$508.09	$760.50	$604.85	$580.95	$763.77	$696.22	$431.65	$642.63	$200.53	$9606.61
Factor–0%	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	
Present worth	$3900.00	$517.42	$508.09	$760.50	$604.85	$580.95	$763.77	$696.22	$431.65	$642.63	$200.53	$9606.61
Factor–6%	1.0	0.9434	0.8900	0.8396	0.7921	0.7473	0.7050	0.6651	0.6274	0.5919	0.5584	
Present worth	$3900.00	$488.13	$452.20	$638.52	$478.56	$434.14	$538.46	$463.06	$270.82	$380.37	$111.98	$8156.23
Factor–10%	1.0	0.9091	0.8264	0.7513	0.6830	0.6209	0.5645	0.5132	0.4665	0.4241	0.3855	
Present worth	$3900.00	$470.39	$419.89	$571.36	$413.11	$360.71	$431.15	$357.30	$201.36	$272.54	$ 77.30	$7475.11
Factor–18%	1.0	0.8475	0.7182	0.6086	0.5158	0.4371	0.3704	0.3139	0.2660	0.2295	0.1911	
Present worth	$3900.00	$438.51	$364.91	$462.84	$311.98	$253.93	$282.90	$218.54	$114.82	$144.91	$ 39.32	$6531.66

(a)Fuel costs: $0.409 per gallon. Fuel consumption: 14 mpg.

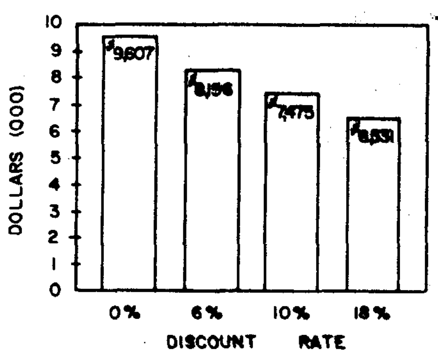


FIGURE 30. PRESENT WORTH OF TOTAL COST OF AUTOMOBILE OWNERSHIP AT SELECTED DISCOUNT RATES

depending upon the rate at which future costs are discounted. The 0-percent discount rate reflects costs as though they occur simultaneously, i.e., completely neglecting the idea that future dollars are worth less today. The 18-percent discount rate most appropriately reflects the consumer cost of financing nondurable goods and is, therefore, too high a discount rate for consumer transportation. The 6-percent discount rate is not high enough; 6-percent more nearly reflects the yield on consumer savings which is always 2 to 4 percent below the cost of money to the same consumer. Therefore, for the remainder of this analysis of automobile costs to the consumer, all future costs will be discounted at a rate of 10 percent/yr.

The Cost of Gasoline

The second variable to be considered is the cost of gasoline. No attempt is made to predict what gasoline prices will be at any future time. Three alternative gasoline costs are assumed, and their impact on automobile ownership costs is examined. First, the estimated 1973 gasoline price of $0.409 ¢/gal is used; second an arbitrary $0.500/gal is chosen; and the third alternative is the $0.409/gal increased at 5 percent/yr compounded annually. (The 5-percent annual increase equals the average annual increase in the Consumer Price Index for the years 1967 through 1972.)

The annual gasoline cost in 1973 dollars (discount rate equals 0 percent) is illustrated in Figure 31 for the three alternative pricing assumptions.

As with the prior sensitivity analysis, as the price of gasoline is allowed to vary, all other variables are fixed. In this case, the fixed costs include the nonvariable costs from Table 13 with all future

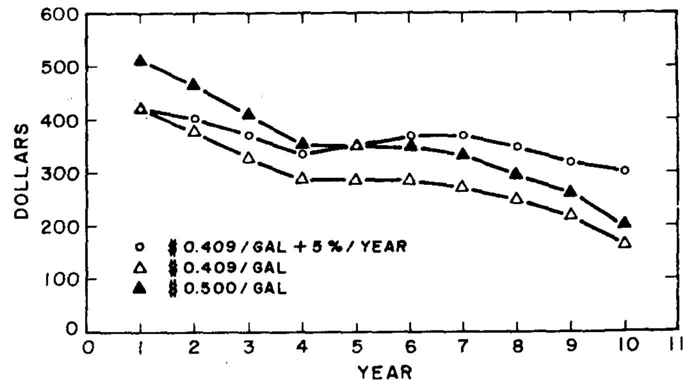


FIGURE 31. COST OF GASOLINE BY YEAR FOR THREE PRICING ALTERNATIVES (1973 DOLLARS)

costs discounted at 10 percent. The tabulation of the remaining variable, total gasoline cost, at the three selected prices is shown in Table 15.

TABLE 15. TOTAL GASOLINE COSTS AT SELECTED FUEL COSTS[a]

Item	Year											Total
	0	1	2	3	4	5	6	7	8	9	10	
Present worth factor–10%		0.9091	0.8266	0.7513	0.6830	0.6209	0.5645	0.5132	0.4665	0.4241	0.3855	
Gasoline cost:												
$0.409 per gal		$423.61	$379.79	$335.96	$292.14	$289.22	$289.22	$277.54	$248.32	$219.11	166.52	$2921.43
Present worth		385.09	313.86	252.40	199.54	179.56	163.25	142.44	115.83	92.92	64.19	1909.08
$0.500 per gal		$517.85	$464.28	$410.71	$357.14	$353.57	$353.57	$339.29	$303.57	$267.86	$203.57	$3571.23
Present worth		470.78	383.69	308.57	243.93	219.53	199.59	174.12	141.62	113.60	78.48	2333.91
$0.409 per gal plus 5% annual inflation		$423.61	$398.82	$370.46	$338.29	$351.66	$369.27	$372.06	$349.53	$323.84	$258.41	$3555.95
Present worth		385.09	329.59	278.33	231.05	218.35	208.45	190.94	163.06	137.34	99.62	2241.82

(a) Fuel consumption rate: 14 mpg

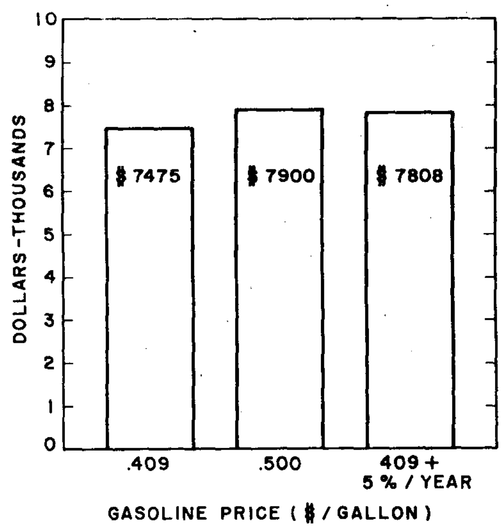


FIGURE 32. EFFECT OF GASOLINE PRICING ON THE TOTAL COST OF AUTOMOBILE OWNERSHIP AND OPERATION

The nonvariable costs and the fuel cost for the three pricing alternatives are combined and discounted to illustrate the effect of various gasoline prices on the total cost of automobile ownership and operation. These results are illustrated in Figure 32.

From Figure 3, it is apparent that there is no significant difference in the total cost of automobile ownership and operation if a gasoline cost of $0.500/gal for 10 yr, or the alternative of the initial cost of $0.409/yr, inflated at a rate of 5 percent/yr for the 10-yr period is used. With all the existing uncertainties concerning gasoline prices, the most reasonable alternative is the one which starts with a good estimate of present cost and increases at a uniform rate over the study period. Therefore, for the remainder of this analysis the cost of gasoline is assumed to be $0.409 inflated at a rate of 5 percent/yr.

Gasoline Consumption Rates

The final independent variable, and the primary objective of this an alysis, is gasoline consumption rates. As with the previous examinations, the determination of the sensitivity of ownership cost to various fuel consumption rates will be made with all other variables fixed. All future costs are discounted to present worth at a rate of 10 percent. Gasoline prices start at $0.409/gal and increase at a rate of 5 percent/yr compounded annually. The tabulation of gasoline costs at consumption rates of 6 to 30 miles/gal is illustrated in Table 16.

TABLE 16. TOTAL GASOLINE COST AT SELECTED FUEL CONSUMPTION RATES IN 1973 AND DISCOUNTED DOLLARS

Cost	Miles per gallon						
	6	10	14	18	22	26	30
Present dollars (1973)	$8297.22	$4978.34	$3555.95	$2765.75	$2262.88	$1914.76	$1659.45
Present worth at 10% discount rate	5230.92	3138.56	2241.83	1743.62	1426.61	1207.12	1046.19
Fuel price: $0.409 per gallon, increased 5% annually.							

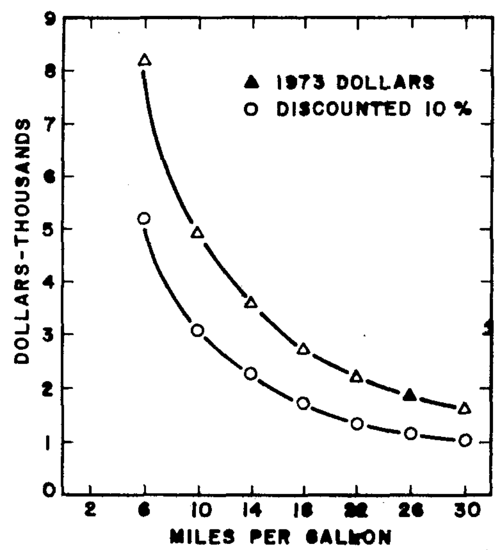


FIGURE 33. TOTAL GASOLINE COST FOR 100,000-MILE LIFETIME AT VARIOUS FUEL CONSUMPTION RATES

Total cost of gasoline over a 10-yr period at various consumption rates is shown graphically in Figure 33.

The impact of various fuel consumption rates on the total cost of ownership is illustrated in Figure 34.

Conclusion

The apparent additional increment first cost that the consumer can pay for fuel conserving innovations and not increase his total ownership and operation cost is shown in Table 17. These allowable additional first costs are shown for vehicle mileage improvement with respect to 10, 14, and 18 mpg.

Using the data from Table 17, three examples of the incremental additional initial purchase price that could be paid if fuel consumption were reduced are illustrated in the upper section of Figure 35. The three examples show the various incremental first costs allowable as a function of the increased mileage with respect to initial mileage rates of 10, 14, and 18 mpg. For an automobile which presently operates at 14 mpg, if a fuel consumption rate could be achieved at an initial cost to the consumer of $500 the *total* cost of automobile ownership and operation would not be increased.

In the lower section of Figure 35 the composite fuel consumption for each of the six reference vehicles (A through F) is illustrated. The derivation of the composite fuel consumption data for each vehicle is described elsewhere in this report.

TABLE 17. ALLOWABLE INCREMENTAL ADDITIONAL FIRST COSTS FOR INCREASED FUEL ECONOMY ABOVE 10 MPG, 14 MPG, AND 18 MPH

Vehicle mileage (mpg)	Non-variable costs ($)	Gasoline costs ($)	Total costs ($)	Additional costs with respect to 10 mpg ($)	Additional costs with respect to 14 mpg ($)	Additional costs with respect to 18 mpg ($)
10	5,566	3,138	8,704	0	--	--
14	5,566	2,242	7,808	896	0	--
18	5,566	1,744	7,310	1,394	498	0
22	5,566	1,427	6,993	1,711	815	317
26	5,566	1,207	6,773	1,931	1,035	537

Note: All costs are discounted to present worth at 10%.

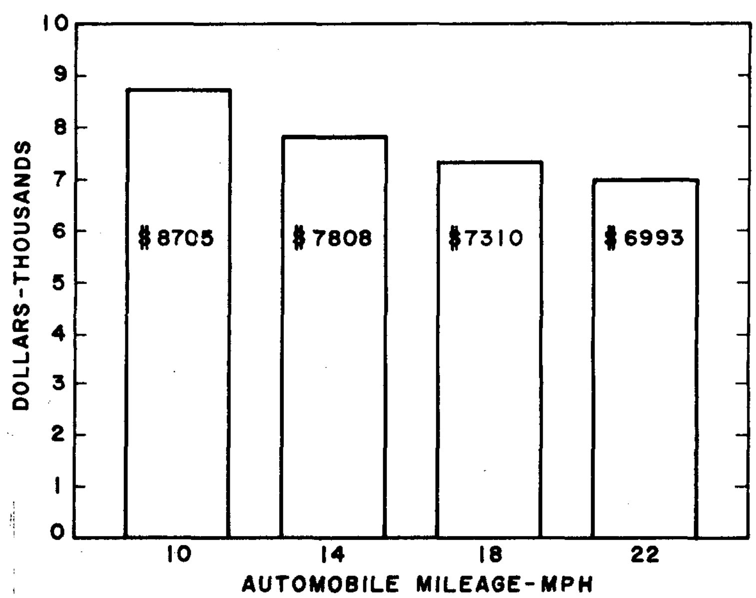


FIGURE 34. THE EFFECT OF VARIOUS FUEL CONSUMPTION RATES ON THE TOTAL COST OF AUTOMOBILE OWNERSHIP AND OPERATION

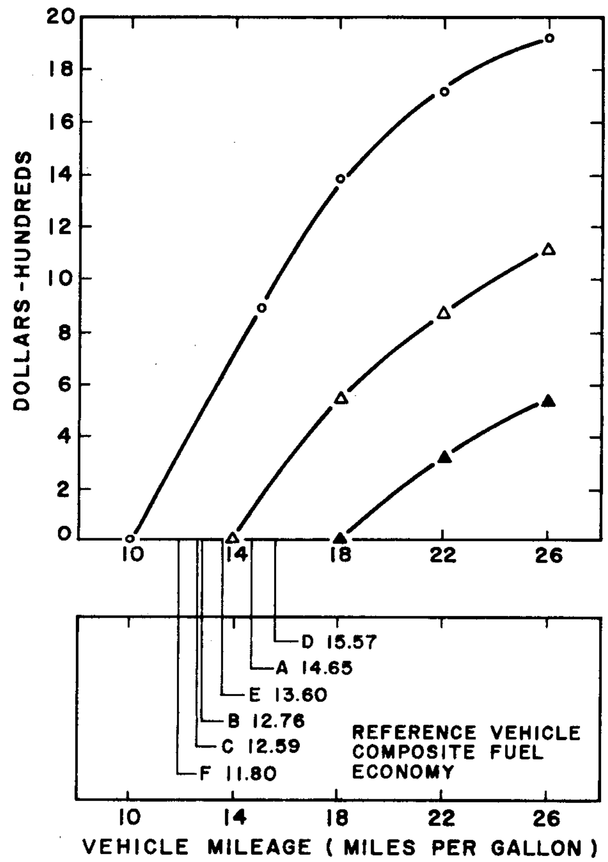


FIGURE 35. ALLOWABLE INCREMENTAL ADDITIONAL COSTS TO REALIZE INCREASED FUEL ECONOMY AT CONSTANT TOTAL COST TO THE CONSUMER WITH REFERENCE VEHICLE COMPOSITE FUEL CONSUMPTION DATA SHOWN

6. CONSIDERATIONS IN PREDICTING PREPRODUCTION LEAD TIME FOR INCORPORATION OF FUEL ECONOMY INNOVATIONS

Introduction

Assuming that the automobile manufacturers are motivated for economic or competitive reasons to incorporate innovations which would increase automobile fuel economy, consideration must be given to the amount of time and effort which must be expended before these innovations could be included in a production model. The purpose of this chapter is to examine the preproduction lead time necessary for the incorporation of a selected group of fuel economy innovations.

Product Development Lead Time

Many of the concepts and innovations which must be considered in increasing automobile fuel economy have not yet been developed to the point where they can be included into a production line automobile. For this reason, when considering lead times as necessary to design, develop, and construct vehicles incorporating fuel conserving innovations, it is necessary to allow time for necessary basic research and development. Once an innovation has been developed to the point that feasibility is assured, it can then enter into a normal product development lead time cycle of the given automobile manufacturer. Product development lead time is defined as the total time required for the development of a product from the initial formulation of the production design concept to the production of the first unit. This lead time is generally 36 to 48 months in duration and can be divided into eight major steps or phases. Therefore, by including the basic R&D necessary, the total lead time required to develop more efficient automobiles consists of nine phases with a total time duration of at least 4 yr. These nine phases are described as follows:

(1) *Basic Research and Development.* Basic R&D is a continuous ongoing process in the automobile industry in which all ideas must compete for R&D funds. It is often difficult to determine when work on a specific idea begins or how long it will take to develop this idea to the point that it can be included in the planning for a production automobile. For the innovations included in this paper, consideration will be given to the amount of R&D which may be necessary to prove feasibility for a production operation.

(2) *Research and Advanced Development.* This phase has its inception in the infinite R&D timespan. It can best be defined as beginning at the time products of the R&D effort are identified as needed or desirable and are included as a part of a specifically identified production package.

(3) *Product Concept Development.* This is the time at which the basic product concepts are developed into a package, or a set of basic assumptions, which begin to look like an automobile, has specific features and dimensions, etc., and costs. It is at this stage that determinations are made on product lines, styles, performance characteristics, power plants, and other things of that nature. This phase involves marketing people, engineering people, and management people.

(4) *Structural Development/Preliminary Design.* This is the phase in which the product concepts are put into a tentative final form. The results of this phase are the models and/or prototypes to be presented to management for approval of the program.

(5) *Program Approval.* At this point, management must decide whether or not to continue with the product program development. Once the decision is made, there is a full commitment to detailed engineering, tooling, and resource allocation. As this approval constitutes a commitment to basic concepts and design, the ease of effecting the changes beyond this point steadily decreases.

(6) *Detailed Engineering.* During this phase, the approved program is converted into the necessary tools, dies, jigs, fixtures, parts, and facilities which are necessary to produce the product. Designs are finalized and prototypes are tested. Also at this time other areas, such as marketing, are beginning to make their plans for selling the product.

(7) *Parts Procurement/Tool Construction Installation and Testing.* This phase involves making the necessary arrangements with various suppliers and vendors for the parts necessary in the production of the product. Arrangements must also be made with tool makers to obtain the necessary major forming and other dies. While some of the larger companies have tooling capabilities, all manufacturers to some extent rely on outside tooling sources to obtain the necessary equipment. Plans are also made during this time for the manufacturing facilities and production lines necessary for the product production.

(8) *Pilot Assembly.* Pilot assembly begins when tools are received, and facilities are made available for the construction of the necessary assembly lines. Pilot assembly determines if the product will go together in a production operation. A few pilot models are used to test the output of the production tooling. These models are also used to ensure that engineering specifications will be met in the production model.

(9) *Production Buildup.* This is the final step in the product development lead time. It is during this time that the facilities which were formerly used for the production of earlier model products are committed to the production of the present line. This phase is primarily concerned with the mechanics of large-scale production and rarely, if ever, has impact on the final configuration of the product.

Product development lead time in the automotive industry has an average length of approximately 48 months. Major innovations or significant product changes, such as the development of a new type of engine, could require a considerably longer development time. This is due mainly to the fact that more than one product development cycle might be involved. Also, there is a rather nebulous dividing line between basic research and development and the research and advance development which marks the beginning of the planning of a particular product line. It must be remembered that the phases listed above are not necessarily sequential but rather are overlapping, the amount of overlap depending on the particular product being developed. A typical automobile development cycle is shown in Figure 36.

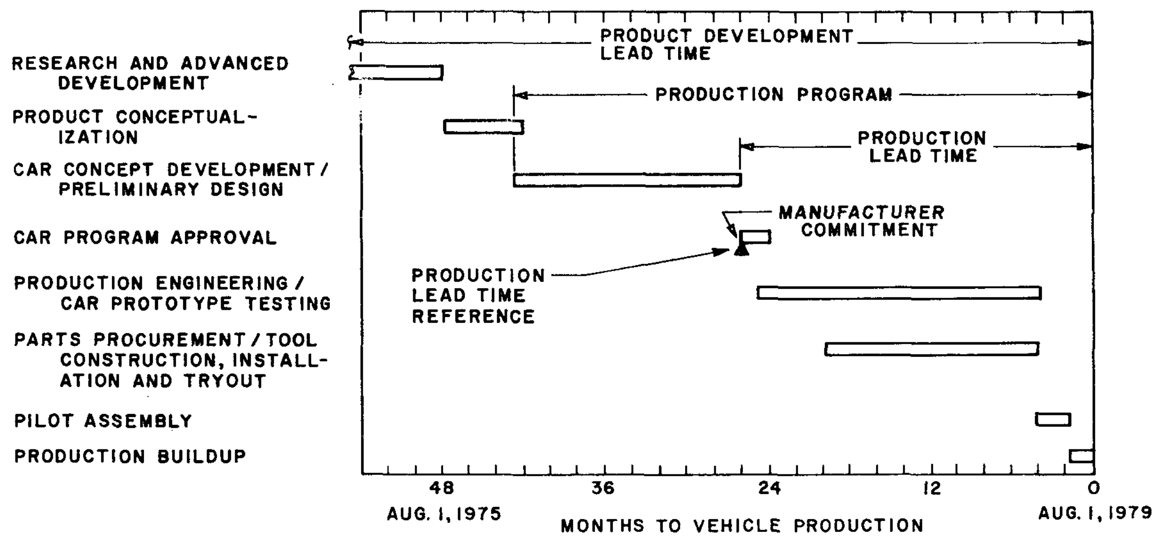


FIGURE 36. AUTOMOTIVE INDUSTRIAL ENGINEERING PROCESS

7. FUEL ECONOMY OF CARBURETED SPARK-IGNITION ENGINES

General

This section will discuss in general terms the performance and the factors that influence the performance of the conventional automobile engine. This engine is a carbureted, spark-ignition (S.I.) engine with multiple reciprocating pistons.

Although the consequences to engine efficiency and power of the primary effects to be discussed below are well documented in textbooks and technical papers, a brief review of these effects is considered necessary in order to clearly delineate the characteristics of the engine, as well as to provide a foundation for later discussions of engine modifications.

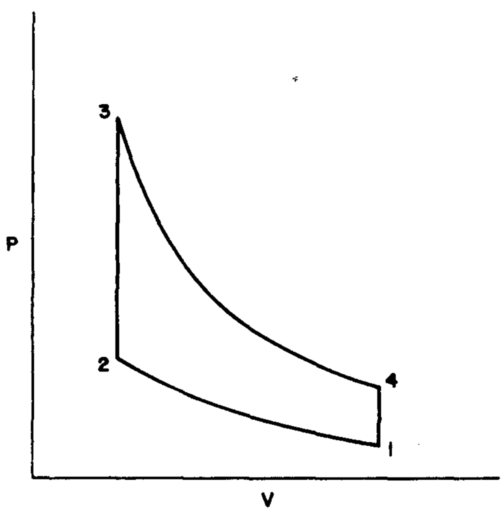


FIGURE 37. PRESSURE-VOLUME DIAGRAM

Theoretical Cycle Efficiency

The theoretical operating cycle of the carbureted S.I. engine is shown in Figure 37 as a plot of combustion chamber pressure and combustion chamber volume.

A gas pressure at P_1 and volume V_1 is compressed by a piston to pressure P_2, volume V_2. At this point, heat is added to the gas and the gas pressure, consequently, increases to P_3 at constant volume. The piston is then allowed to retract while the gas performs work on the piston as it expands to pressure P_4 and volume V_1. The gas is then cooled at constant volume to pressure P_1 to begin the cycle over again.

A simple thermodynamic analysis of this cycle shows that the cycle efficiency (η_t) is:

$$\eta = 1 - \left(\frac{1}{r}\right)^{k-1} \qquad (1)$$

where r is the compression ratio (V_2/V_1) and k is the ratio of the specific heats of the gas being used. Thus, for the theoretical cycle, the efficiency is dependent only upon the compression ratio and a gas property. The power (P) developed by the theoretical engine is:

$$P = Q\eta_t \qquad (2)$$

where Q is the heat added to the gas, per unit time.

For later reference, the effect of equation (1) is shown in Figure 38 using a mixture of fuel and air in the combustion chamber.

Note that the efficiency first increases sharply with compression ratio, then less strongly as the compression ratio becomes larger.

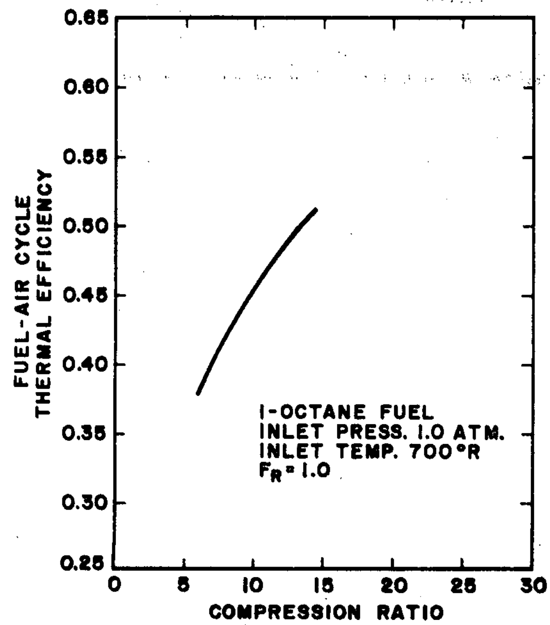


FIGURE 38. EFFICIENCY AS A FUNCTION OF COMPRESSION RATIO

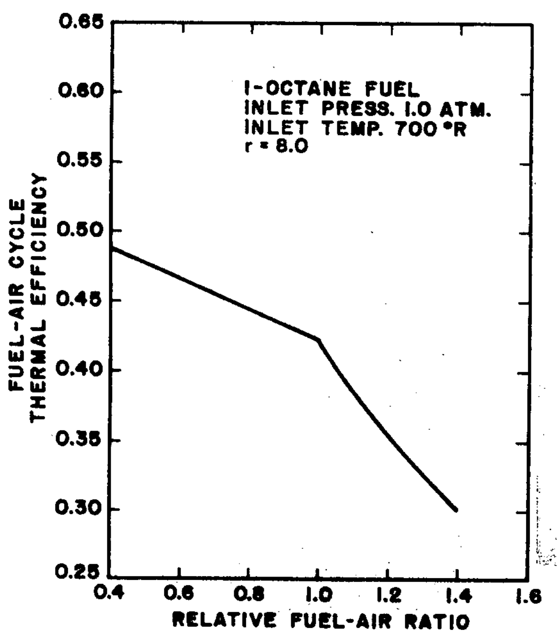


FIGURE 39. EFFICIENCY AS A FUNCTION OF FUEL AIR RATIO

Factors Affecting Actual Engine Performance

The theoretical cycle just discussed cannot be achieved in practice, and Figure 38 represents the idealized theoretical limits of efficiency. The various phenomena that affect the efficiency of a real engine will be discussed below.

Actual Working Fluid

In the idealized theoretical cycle, the gas was heated to increase the cycle pressure from P_2 to P_3. In a real engine, the working fluid is a mixture of fuel and air which is burned to increase the pressure. Equation (2) can be expanded to account for the fact that the heat added (Q) comes from the combustion of fuel:

$$Q = C(V_2 - V_1)\, N\rho_1 F h_v \qquad (3)$$

where C is a constant; $(V_2 - V_1)$ is the engine "displacement"; N is the rotational speed of the engine in revolutions per minute; ρ_1 is the density of the air at the pressure P_1 and temperature T_1; F is the weight ratio of fuel to air in the mixture; and h_v is the "heating value" of the fuel, i.e., the quantity of heat released by burning a unit weight of fuel. Then,

$$P = C(V_2 - V_1)\, N\rho_1 F h_v \eta_t \qquad (4)$$

The very high gas temperatures attained during combustion cause an increase in the specific heat of the gases as well as dissociation of some of the molecules of burnt products. Both these effects limit the maximum temperature below the theoretical temperature that would be attained in the absence of the effects and hence reduce efficiency. From equation (3), one would expect that a reduction in F would reduce these effects and, therefore, improve the efficiency. That this is correct can be seen from Figure 39, which shows calculated efficiencies for a range of fuel/air ratios.

In a real engine, all of the products of combustion are not exhausted at the end of the expansion stroke, and the fresh gases in the next cycle will include a small fraction of burned gases from the previous cycle. The presence of this residual fraction can be detrimental to performance. Further, the mixing of fuel and air is never complete, and the fuel is frequently not completely vaporized. Thus, the actual working fluid may consist of fuel/air mixture, gas residuals, air, vaporized fuel, and liquid fuel.

Incomplete Combustion

In every real engine, the combustion products passing the exhaust valve contain some compounds that could be further oxidized to release heat. If these compounds were burnt at V_1, the efficiency and power output of the engine would be improved. However, even though these incompletely reacted emissions (unburned hydrocarbons, CO, C) are judged to be environmentally hazardous, the quantity of these emissions is such that the engine efficiency and power is only slightly affected. One estimate[20] is that theoretical efficiency is lowered about 1.5 percent by this cause at 40 mph steady cruise. Note that this improvement is not necessarily gained by the addition of emission controls, since burnup of combustible emissions may not be carried out near piston top center.

Time Losses

The theoretical cycle (Figure 37) required that burning (or heat addition) be accomplished at constant volume or equivalently, that combustion be instantaneous. High burning rates in a real engine cause high-frequency engine vibration and noise, as well as high stresses on engine parts, and are, therefore, undesirable. (Diesel engines usually have relatively high burning rates. This characteristic contributes to engine noise but may contribute a small amount to their increased efficiency.)

Again, the theoretical cycle calls for constant volume exhaust of the burnt products. This characteristic is only approached by real engines due to limitations in valve opening speed and valve port area.

These real effects will be called "time losses," since they involve rate processes. The deviation from the constant-volume events of the theoretical cycle reduces the area within the cycle loop of Figure 37; hence the power output and efficiency are reduced. At high power outputs, these losses may represent about a 5-percent loss in efficiency from the theoretical cycle, although this loss will vary widely from engine to engine, as well as with speed and load.

Heat Losses

The engine structure (cylinders, cylinder heads, pistons) must be maintained at temperatures much lower than the peak combustion gas temperature in order to preserve structural integrity, to prevent excessive deterioration of lubricating oil, and to reduce detonation tendency. Therefore, the high temperature combustion gas loses heat to these components, which reduces the efficiency and power output of the engine.

In general, percentage heat losses (heat loss compared to total heat release in the cylinder) decrease strongly as engine speed increases. This decrease in percentage heat loss is evident even though the thermal resistance of the heat path is reduced. The reduction occurs because the fuel input increases with speed at a greater rate than the heat loss. Also, percentage heat loss decreases slightly with load increase (at constant speed), because the heat loss does not increase as fast as the total heat input with increasing load. Heat losses have widely varying effects upon engine efficiency, depending upon engine design and operating conditions, but a rough estimate for

[20]Cleveland and Bishop, "Several Possible Paths to Improved Part-Load Economy of Spark-Ignition Engines," SAE Paper 150A, March 1960.

efficiency loss is 6 percent at midload, mis-speed conditions, compared to an engine without heat losses.

Pumping Losses

The air that passes through the engine experiences pressure drops due to the restrictions of the air cleaner, carburetor, throttle valve, inlet and exhaust manifolds, inlet and exhaust valves, muffler and tailpipe. These pressure losses, when multiplied by a suitable volume flow rate, give the power required for the pumping losses. Pumping losses increase with increasing speed and decreasing load (the latter due to the increased loss across the throttle valve).

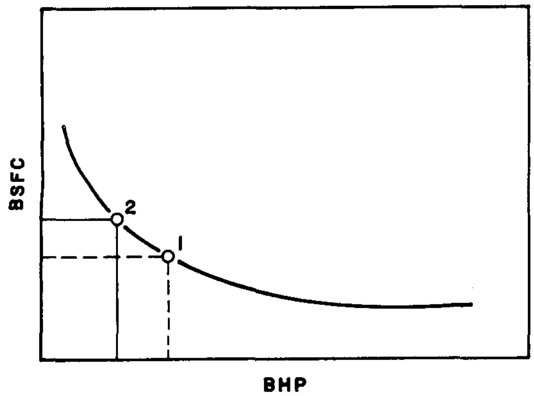


FIGURE 40. CARBURETED ENGINE BSFC AT CONSTANT SPEED

Pumping losses represent a very significant portion of the overall engine friction at light loads. Part of the fuel economy advantage the diesel enjoys over the carbureted engine is due to the reduction of pumping losses.

The increase of pumping losses with reduced load in a carbureted engine, combined with the use of richer fuel/air mixtures at low load (see Section C.7.), can lead to some apparently inconsistent engine behavior. Consider the curve of Figure 40 for a carbureted engine at constant speed. Assume that the engine is operating at point 1 and that the required horsepower of the engine is reduced to point 2 by, say, the reduction of aerodynamic drag or rolling resistance. The engine now operates at a *higher* bsfc ($bsfc_2 > bsfc_1$). Even so, it would be expected that the fuel consumption rate (lb/hr) would be reduced. The fuel rate (in lb/hr) will be termed f:

$$f \propto (isfc)(ihp) \propto \frac{ihp}{\eta t}$$

where ηt is the thermal efficiency.

Since ihp = bhp + fhp:

$$f \propto \frac{bhp + fhp}{\eta t}$$

then

$$\frac{f_1}{f_2} = \frac{\eta t_2}{\eta t_1} \frac{(bhp_1 + fhp_1)}{(bhp_2 + fhp_2)}$$

In moving from point 2 to point 1, throttling losses increase ($fhp_2 > fhp_1$), brake horsepower decreases ($bhp_2 < bhp_1$), and the thermal efficiency may decrease due to richer fuel/air mixtures ($\eta t_1 > \eta t_2$). If the effect of the decrease in thermal efficiency and the increase in friction horsepower is greater than the effect of reduced brake horsepower, f_2 will be greater than f_1, and the reduction in load will actually increase fuel consumption.

This effect is not unusual at low engine loads where throttling losses are a major portion of the total engine friction loss.

Mechanical Friction Losses

Mechanical friction losses include journal-bearing friction, piston friction, piston ring friction, and valve gear friction. Other losses such as oil sloshing, oil pumping, and water pumping are included. Of these, the most significant friction loss is attributed to the piston and rings. Data[21] from widely diverging types of engines indicate that of the total loss attributed to pistons, rings, bearings, and valve gear, the piston and ring loss comprise some 75 percent over the speed range of the engine.

All friction losses tend to become larger with increased piston speed. Losses from pistons and rings are no exception and these losses are further influenced by cylinder pressure. As cylinder pressure increases, the compression rings are forced outward with greater force, and friction losses are higher. Thus, increases in engine power enlarge the friction losses of the piston and rings. Increases in oil viscosity also result in higher friction losses.

The effect of engine stroke and bore is a little more involved. Piston speed(s) is defined as

$$s = 2NS$$

where N is engine speed in rpm and S is the piston stroke. Engines with the same bore but different strokes have the same piston and ring friction at the same piston speed. At the same *engine speed*, the engine with the smaller stroke-bore ratio has the lesser friction. However, since maximum engine speeds are limited by piston speeds for acceptable life and adequate ring performance, this distinction has little real importance.

If two engines of different size, similar in all respects, are run at the same engine speed and imep with the same lubricant, the engine with the larger bore will have a reduced percentage friction loss (percentage friction loss is friction horsepower divided by indicated horsepower). The reduction in percentage friction loss will be in proportion to the ratio of the bores of the two engines. This effect is small in the range of cylinder bores used in automobile engines.

It should be noted that some aircraft engines have percentage friction losses that are significantly lower than conventional automobile engines. This is primarily due to light reciprocating parts, large piston-to-cylinder clearances, short pistons, and light ring pressure. However, these changes result in a higher level of engine noise and possibly reduced lubricant life and higher lubricant consumption.

Automotive engines typically use two "compression" rings (rings designed primarily to seal combustion chamber pressures) and one oil control ring (a ring that limits the oil flow into the combustion chamber). Without a breakthrough in ring design, a decrease in the number of rings presently used will lead to high blowby and oil consumption.

Fuel/Air Ratio Variations

In the section on actual working fluids, page 52, the benetical effect of reduced fuel/air ratios upon efficiency was discussed. In this section, some practical considerations of fuel/air ratio will be made.

[21] Taylor, C. F., **The Internal Combustion Engine in Theory and Practice**, 2nd Edition, M.I.T. Press.

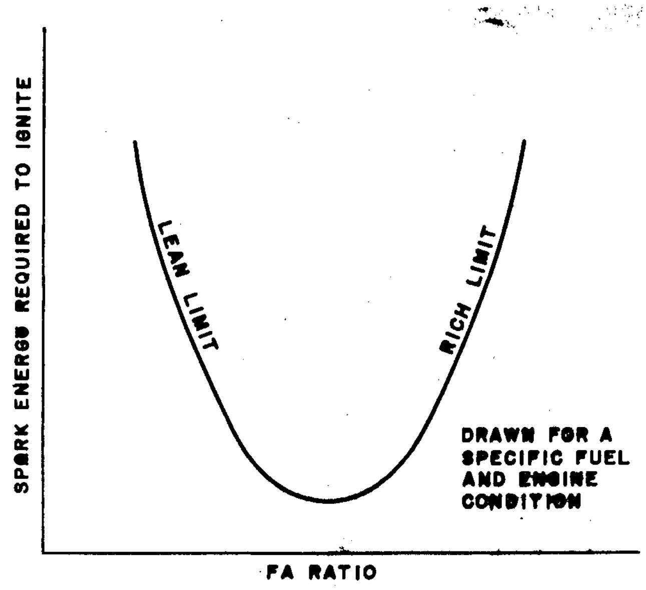


FIGURE 41. ENERGY REQUIRED FOR IGNITION

One of the most important phenomena associated with the carbureted, S.I. engine is the general relationship shown in Figure 41.

This curve shows that, given a certain level of spark energy in the engine ignition system, a certain range of fuel/air ratios are ignitable. Higher spark energies will ignite a wider range. At some minimum spark energy, there is no fuel/air ratio that can be ignited. It is also important to note that different fuels have widely differing characteristics in this regard. For instance, hydrogen-air mixtures will ignite over a much broader range of fuel/air ratios than gasoline/air mixtures. A great many other factors influence the shape of the curve as well, and some of these will be discussed below.

Conventional S.I. carbureted engines operate in the relative fuel/air ratio range of from about 0.9 to 1.3. (Relative fuel/air ratio, F_R, is the ratio of the actual fuel/air ratio used to the stoichiometric fuel/air ratio.) These limits are dictated by a number of things, other than the ignitable limits of the mixture. Reduction in F_R reduces the flame speed and thus contributes to increased time losses (see Time Losses, page 53). Carbureted multicylinder engines are troubled by problems in obtaining even fuel and air distribution, consequently, the fuel/air ratio varies from cylinder to cylinder. As one tries to operate the engine on leaner mixtures, misfires will begin first on the leanest-running cylinder. This will occur at an overall *engine* fuel/air ratio well above the lean limit of ignition determined by more controlled and precise methods.

A number of other factors influence the lean limit of operation of a carbureted S.I. engine. Increase in inlet air temperature lowers the lean limit[22]. An increase in charge turbulence at the time of ignition can increase the ignitable mixture range, but has detrimental effects upon heat losses and pumping losses. Charge turbulence tends to make the fuel/air mixture more homogeneous and reduces the chance of misfires due to the ignition spark firing in local zones of too-rich or too-lean mixtures.

The fuel/air ratios used in automobile engines vary with operating requirements:

At idle and low loads, a rich mixture is required for consistent engine operation without misfires. This is due to the reduced intake manifold pressure (closed throttle) which causes a portion of the exhaust gases to expand into the intake manifold so that the "fresh" charge of fuel/air mixture contains a relatively high percentage of burnt gases from the previous cycle. This "charge dilution" affects the heat content per unit volume of the mixture so that reliable combustion requires richer fuel/air mixtures. High valve overlap further increases the charge dilution and increases the required fuel/air ratio. At mid-range or cruise conditions, charge dilution is small and leaner mixtures can be used to improve fuel economy.

[22]Warren, Glenn B., "Fuel Economy Gain from Heated Lean Air/Fuel Mixtures in Motorcar Operation," ASME Paper 65-WA/APC-1.

At full load, the mixture is again made richer. This is done to develop the maximum power capabilities of the engine. Qualitative explanation of this requirement can be obtained from Equation (4) and from Figure 39. From Equation (4), it is seen that power is proportional to the product $F\eta_t$. The product $F\eta_t$ is maximized at a F_R of about 1.1 (reference Figure 39) which is somewhat richer than stoichiometric. In real engines, due to fuel distribution problems, the actual relative fuel/air ratio used is somewhat higher.

The real engine also has a set of transient operating mixture requirements. For cold starting, a very rich mixture is required, since the low air temperatures produce reduced fuel evaporation and the cold walls of the inlet manifold make the problem more severe. Fuel/air ratios of 1.0 ($F_R = 15$) may be needed under these conditions, and the carburetor is provided with a choke to produce this mixture. During warmup, while the inlet manifold walls become warmer, the fuel/air ratio must be progressively decreased.

In steady-state operation, the inlet manifold walls are covered with a film of liquid fuel. At higher manifold pressures (higher loads), the quantity of liquid fuel on the wall is greater because of the greater total fuel flow and the reduced propensity for evaporation at the higher manifold pressure. Upon acceleration by the sudden opening of the throttle, the mixture in the cylinder will become leaner because of the relatively low velocity of the fuel film on the wall. This effect will cause misfiring of the engine, unless provisions are made to provide a richer mixture during acceleration. This is done by the accelerator pump, which throws in a quantity of fuel when the throttle is opened. Note that the need for acceleration enrichment is greatly reduced or possibly eliminated by fuel injection systems that introduce the fuel into the combustion chamber or into the inlet port very near the inlet valve.

Spark Plug Effects

Spark plug design affects the mixture ratios at which ignition can occur. The so-called "quench" distance (the distance from a cool surface within which flame will not propagate) plays a role in the choice of sparkplug gap location, the size of the plug gap, and the detail design of the plug gap and surrounding insulation.

The voltage at the sparkplug gap has two primary characteristics: rise time to the "breakdown" voltage where the arc is established, and the duration of the arc. Fast voltage rise time is helpful in firing fouled plugs. Short arc duration (at high energy levels) tends to cause higher gas temperatures between the plug gaps without detriment to plug life. However, long arc duration has proven to be beneficial in achieving reliable ignition of lean mixtures[23]. The reason for this is found in the previous discussion of fuel/air ratios: the actual fuel/air mixture is nonhomogeneous, and a long arc duration has a better chance of "finding" a suitable mixture ratio for ignition. This effect is enhanced as the fuel/air mixture becomes more fuel-lean.

As is well known, spark advance or retard has a strong effect upon engine performance. The goal is to achieve a close approach to constant-volume combustion (see Theoretical Cycle Efficiency, page 51), without excessive engine noise caused by high pressure rise rates. Since complete combustion of the mixture requires a finite time due to the relatively slow flame speed, the spark must occur before the piston reaches top center. The number of degrees of crankshaft revolution required for complete burning of the fuel/air charge increases slightly with engine speed, decreases with load, and increases with leaner fuel/air mixtures. At every operating condition, there is a single spark

[23] Craver, R. J., et al, "Spark Plug Design Factors and Their Effect on Engine Performance," SAE Paper 700081.

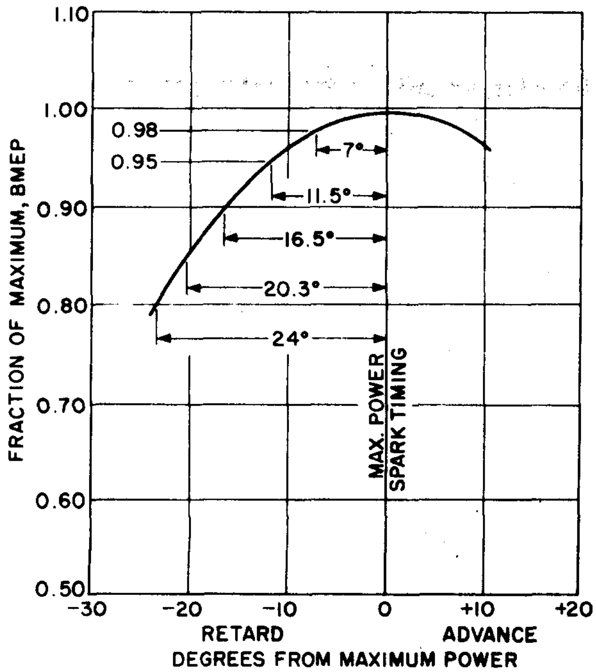


FIGURE 42. CORRELATION FOR ALL SPEEDS AND LOADS

timing for best power output (see Figure 42). Since for constant speed and throttle setting, bsfc is inversely related to bmep, the point of maximum bmep is also the point of minimum bsfc. Spark timing also has a strong effect upon exhaust emissions, as will be discussed later.

Fuel Effects

The carbureted S.I. engine is affected by the following fuel characteristics:

Volatility
Detonability
Pre-ignition characteristics
Heat of combustion
Heat of evaporation

Volatility affects starting and warmup performance of the engine, with fuels of low volatility causing difficult starting and long warmup periods (the period necessary to reach the designed steady-state inlet manifold temperature). Fuels of low volatility also cause crankcase oil dilution, while high volatility fuels can produce vapor lock, evaporation losses, and fire hazard.

The detonation characteristics of fuels have been extremely influential in the development of the carbureted S.I. engine. The compression ratio of engines has, in general, increased over the years in response to the availability of higher octane fuels. Detonation, or engine "knock," is a result of the extremely fast combustion of the last of the fuel/air mixture to burn (the "end gas"). As the flame moves across the combustion chamber, the end gas is compressed by the increasing combustion pressures and is thereby heated. Given enough time and a high enough temperature, the end gas will spontaneously and simultaneously explode, and this explosion produces a pressure wave in the combustion gas that is audible as a knocking sound. The resistance of a fuel/air mixture to knocking performance is measured by the *octane number*. Higher octane numbers reflect higher resistance to knock. Given a certain octane number, the engine load at which knocking begins will decrease with any change that increases end gas temperature or increases the time between the induction of the fuel/air mixture and the arrival of the flame at the end gas location. Thus, the knocking tendency is enhanced when compression ratio is increased (higher temperature), engine speed is decreased (longer time), inlet air temperatures are increased (higher temperature), spark timing is advanced (higher temperature), coolant temperature is increased (higher temperature), fuel/air ratio is decreased (slower flame speed, therefore longer time), and so forth. Knock is objectionable, not only from the noise it produces but due to the fact that prolonged knocking operation can seriously damage engine pistons, as well as lead to pre-ignition.

Pre-ignition is the early ignition of the fuel/air mixture before the spark. It is caused by the presence of an overheated surface in the combustion chamber. Knocking operation tends to overheat surfaces of the combustion chamber, which leads to pre-ignition. Since pre-ignition has the same effect as an overly advanced spark timing, pre-ignition further strengthens the knocking phenomena. Pre-ignition is very sensitive to fuel composition but it is not a simple function of octane number.

The product of heat of combustion and fuel/air ratio affects directly the power output of the engine [see Equation (4)]. In a carbureted S.I. engine, the range of fuel/air ratios is limited to those in the ignitable range, as already discussed. Both these factors depend upon the fuel type. However, the product of $F_c h_v$, where F_c is the stoichiometric fuel/air ratio and h_v is the lower heating value of the fuel, has a range of only 1210 Btu/lb to 1755 Btu/lb for an extremely wide range of fuels, from heavy diesel oils to hydrogen gas. For typical liquid fuels, the product ranges only from 1210 to 1275 Btu/lb.

The heat of evaporation of the fuel influences engine operation through the reduction of inlet air temperature. Reduction in inlet air temperature increases inlet air density (ρ_1) and increases power output [see Equation (4)]. This effect is defined by this relationship:

$$\frac{P_1}{P_2} = \frac{\sqrt{(T_i - x\Delta T)_1}}{\sqrt{(T_i - x\Delta T)_2}}$$

where P is engine power at a given speed, T_i is inlet air temperature, x is the fraction of fuel evaporated in the inlet manifold, and ΔT is the temperature drop of the stoichiometric mixture as a result of complete fuel evaporation. The subscripts indicate two cases: an engine running with fuel type 1, and the same engine at the same speed using fuel type 2. For gasoline, ΔT is about 40°F, for comparison the ΔT for methyl alcohol is 300°F.

The inlet air density of Equation (4) is a direct function of the partial pressure of the air, which is in turn influenced by the weight of fuel evaporated in the air and the molecular weight of the fuel. The air density ρ_1 is thus affected by the fuel/air ratio and the fuel molecular weight, and the effect on engine performance is:

$$\frac{P_1}{P_2} = \frac{[1 + F(29/m_f)]_2}{[1 + F(29/m_f)]_1}$$

where m_f and F are fuel molecular weight and fuel/air ratio, respectively. This effect tends to favor fuels with low stoichiometric fuel/air ratios and high molecular weights, although with the range of liquid fuels used in engines today, the effect is small.

8. EXHAUST EMISSIONS OF AUTOMOTIVE SPARK-IGNITION CARBURETED ENGINES

General

Any useful consideration of engine performance and economy cannot be accomplished without a simultaneous consideration of engine exhaust emissions and the desired level of emission standards. This section provides the reader with a general background of the source of exhaust emissions, the effect upon emission levels of variations in engine parameters, and the various control techniques. One of the major objectives in providing this description is to show the degree to which exhaust emissions genesis and basic engine operation are connected so that later evaluations of fuel-saving concepts, with respect to expected changes in exhaust emission levels, can be better understood.

Source of Exhaust Emissions[24]

Unburned Hydrocarbons

It is generally agreed that unburned hydrocarbons found in the exhaust gases result from the failure of the combustion flame to propagate all the way up to a wall surface. This "wall quench" effect is explained through molecular chain-breaking reactions (reactions, made more likely by the presence of the wall, which modify combustion-promoting molecules to reduce the activity of the combustion process), as well as the cooling of the mixture adjacent to the wall, which is always cooler than the gases. Therefore, after the combustion process is ended, a layer of unburned fuel/air mixture about 0.04 in. thick (very roughly) is left adjacent to the walls of the combustion chamber. During the exhaust stroke, a portion of this unburned mixture leaves the chamber through the exhaust valve. The quench distance increases with both richer and leaner deviations from fuel/air stoichiometry, although the quency distance increases faster with leaner mixtures (away from stoichiometric) than it does with richer mixtures. The quench distance increases with decreasing temperature of the fuel/air mixture, with decreasing pressure of the fuel/air mixture, and with decreasing temperature of the wall.

A second source of unburned hydrocarbons is, of course, the unburned fuel/air mixture from an engine with partial or complete misfires. Conditions resulting in misfires have been discussed earlier: fuel/air ratios outside the rich or lean limits, poor fuel/air mixing, spark with insufficient energy or a too-short duration, or excessive residual gas.

Carbon Monoxide

When hydrocarbon fuels burn, carbon monoxide is produced. If time, temperature, and oxygen availability are adequate, the carbon monoxide reacts to carbon dioxide. Carbon monoxide emissions are tied very closely to the fuel/air ratio actually existing in the engine cylinder (see Figure 43).

Lean mixtures produce very little carbon monoxide; that any at all are produced is a result of fuel maldistribution (from cylinder to cylinder, from cycle to cycle, and from location to location with the combustion chamber volume), as well as a result of the relatively slow reaction rate of carbon monoxide to carbon dioxide.

[24] Henein and Patterson, **Emissions from Combustion Engines**, Ann Arbor Science Publishers, Inc., 1972.

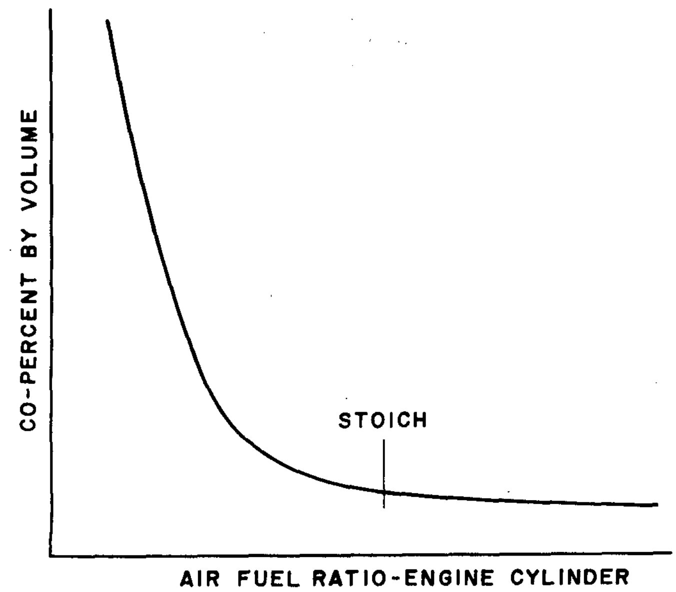


FIGURE 43. CARBON MONOXIDE EMISSIONS

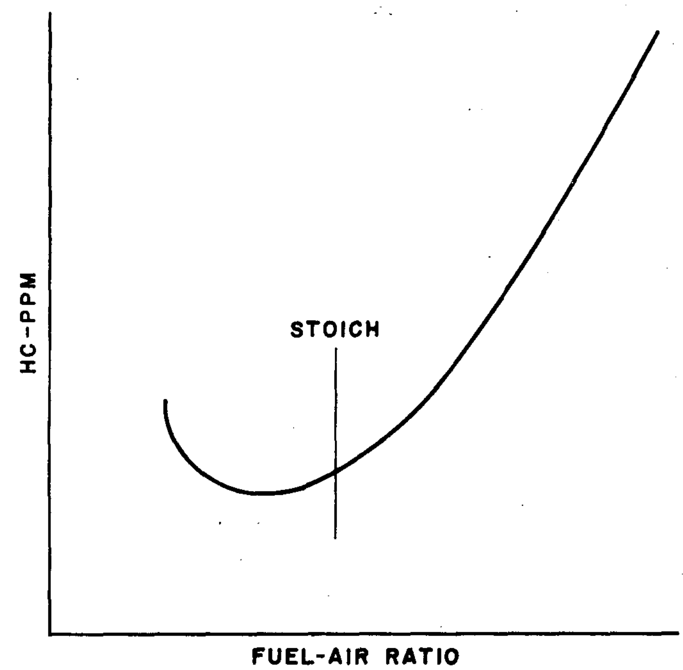


FIGURE 44. UNBURNED HYDROCARBON CONCENTRATION

Oxides of Nitrogen

The formation of oxides of nitrogen (NO_X) involves the nitrogen and oxygen available in the combustion air and requires high temperatures and time for the reactions to take place. Unfortunately, the reactions producing NO_X are more active than those in the reverse direction; so that very little of the NO_X produced in the high-temperature part of the cycle dissociates during the expansion stroke or in the exhaust manifold.

Since the flame front originates at the sparkplug and progresses to the remote parts of the combustion chamber, the burnt gas near the sparkplug is hotter than in any other location in the chamber because of the subsequent compression heating of the first-burned gas. It has been theoretically determined (and verified experimentally) that the NO_X concentration is highest in those gases that are first-burned, due to the higher temperatures and enhanced reaction rates.

Effect of Engine Variables Upon Emissions

Unburned Hydrocarbons

Fuel/Air Ratio—An increase in fuel/air ratio first decreases the thickness of the quench layer, then increases it as fuel/air ratio is further increased. The minimum quench layer thickness lies at a mixture somewhat richer than stoichiometric. However, the concentration of fuel in the quench layer increases linearly with the fuel/air ratio. The product of quench layer thickness and fuel concentration in the layer gives the unburned hydrocarbon concentration in the chamber. A typical curve is shown in Figure 44.

The curve of Figure 44 begins to "hook" upward at lean mixtures due to incomplete combustion in the chamber as the mixture approaches the lean limit. Best fuel economy occurs at a fuel/air ratio very nearly equal to the ratio for minimum HC emissions. The mass emission curve (lb/hr of HC) is similar in shape to Figure 44.

Engine Power—If fuel/air ratio and speed are held constant, while the power output of the engine is varied, the HC emission concentration stays very nearly constant. This is due to a balance of a number of effects. Increased power increases the airflow through the engine and exhaust system (increasing throttle opening) and, therefore, decreases the time the exhaust gases are in the exhaust manifold. This tends to increase HC because of the reduced burnup in the exhaust. On the other hand, power increase increases the average cylinder pressure and reduces the thickness of the quench layer, as well as increases the temperature in the exhaust manifold. These effects reduce HC emissions. The net effect, as observed above, is a more or less constant HC emission concentration with load.

Since the mass airflow rate increases with engine power (and since the mass emission rate is a product of exhaust flow rate and concentration), the mass HC emission rate increases linearly with load.

Engine Speed—Increased turbulence, both in the combustion chamber and in the exhaust manifold, tends to reduce HC concentration as speed increases. This effect is aided by higher exhaust manifold temperatures. The reduction in HC concentration is very nearly linear with speed increase. The mass emission rates also decrease with speed increase, but not as fast as the concentration. This is because of the higher engine airflow due to the increased engine friction at the higher speeds.

Spark Timing—The retarding of spark timing increases the exhaust temperature (later burning), and thereby promotes HC burnup in the exhaust manifold. HC concentration is reduced, as are mass emission rates, except that at highly retarded spark timing the throttle must be opened to maintain power and the mass emission rates start to increase.

As already noted, fuel economy is strongly related to spark timing, and retarded timings decrease thermal efficiency.

Charge Dilution—The increase in the residual fraction of burnt gases in the combustion chamber is connected with a reduction in HC concentration. This is because the increased residual fraction occurs when some of the burnt gases expand into the inlet manifold when the intake valve opens and are pushed back into the chamber on the intake stroke. These residual gases are rich in HC because the last gases to be exhausted are those gases closest to the walls of the chamber containing the quench layer. This effect is not strong, since chamber turbulence tends to even out the distribution of HC within the chamber.

Increased residual fractions occur with increased exhaust pressure to inlet pressure ratio and with increased valve overlap.

As residual fraction is increased further, a point is reached where poor combustion occurs. At this point, HC increases strongly. This effect is most predominant at lean mixtures.

Carburetor and Automatic Spark Advance—In a conventional automobile engine, the carburetor is designed to vary the fuel/air ratio over the load range. Mixtures are rich at idle (to ensure combustion at high residual fractions) and at full load (to obtain maximum power output). Spark is retarded at idle (to reduce idle emission levels), advanced in the mid-range loads, and may be again retarded near full load for control over detonation.

If HC concentration is measured at a constant speed from no-load to full load, a curve similar to Figure 45 will be obtained. The mass emissions, as influenced by exhaust flow rate, are shown in Figure 46.

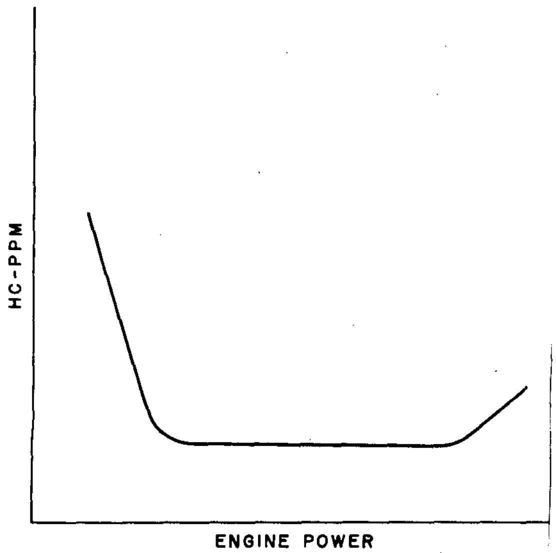


FIGURE 45. HYDROCARBON CONCENTRATION

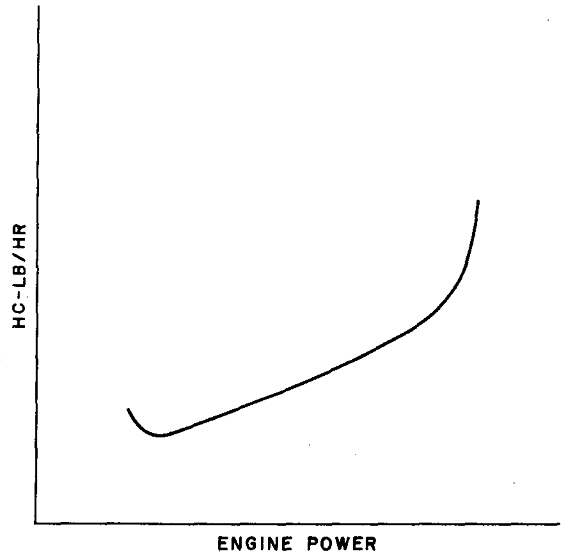


FIGURE 46. HYDROCARBON MASS EMISSION

Engine Design—Since the quench layer is the primary source of HC emissions, the concentration of HC can be reduced by reducing the ratio of the wall surface area to the combustion chamber volume. This can be accomplished in a number of ways:

(a) Cause the shape of the combustion chamber to approach, as nearly as possible, the shape of a sphere.

(b) Increase the stroke-to-bore ratio at the same displacement.

(c) Increase the cylinder displacement. For the same engine power, this is equivalent to reducing the number of cylinders while holding the total engine displacement constant.

(d) Decrease the compression ratio.

It will be obvious that all of these design changes have strong ramifications to engine and vehicle performance, excluding emission consideration. Compact combustion chambers provide good performance but have high pressure rise rates and, consequently, a high level of engine noise. Long stroke engines must operate at lower crankshaft speeds and maximum performance suffers. Large bore engines are susceptible to knock, have higher torque variation (if the cylinder number is reduced), and have higher inertia stresses at a given engine speed. Lower compression ratios decrease thermal efficiency.

Other Effects—Higher combustion wall termperature reduces HC concentration by reducing the thickness of the quench layer. This reduction amounts to perhaps 0.5 ppm HC/°F rise in wall temperature. High wall temperatures increase detonation tendency, reduce volumetric efficiency, and are detrimental to lubricant life.

Combustion chamber deposits increase HC concentration by increasing the effective wall surface area and by absorbing raw fuel. Since deposits accumulate with driving, HC emissions tend to increase with the life of the vehicle.

Carbon Monoxide

The effects of engine parameters upon CO emissions are less complex than for HC or NO_X. This is due to the mechanism of formation being primarily a function of fuel/air ratio without dependence upon wall quench effects. The effect upon CO formation has been discussed in the above section.

Engine power does not influence CO concentration. Since exhaust mass flow rate increases with engine power, CO mass emission rates also increase linearly with power.

Speed has little effect on CO concentration, but mass emissions will increase slowly with speed increase due to the higher exhaust flows as a result of the increased friction.

Spark timing has little effect on CO concentration, except at abnormally retarded timings where late combustion provides insufficient time to complete CO oxidation. This increase is enhanced when CO mass emissions are considered, since the retarded timings cause power loss which must be remedied by opening the throttle and increasing exhaust mass flow.

The carburetor characteristics normally found in an automobile engine produce higher CO concentrations at both idle and full load than at mid-loads, again due to richer mixtures at these operating conditions. The curve for CO concentration is somewhat similar to that for HC.

Engine design has little effect on CO emissions, as does deposit buildup and wall temperature.

Nitric Oxide

Fuel/Air Ratio—Since both the combustion temperature and oxygen content of the combustion gases affect NO_X concentration, the fuel/air ratio is an important NO_X parameter. Figure 47 shows this relationship in a qualitative fashion. Peak NO_X concentration occurs at a fuel/air ratio somewhat leaner than stoichiometric. Richer mixtures lack oxygen and leaner mixtures have reduced peak temperatures.

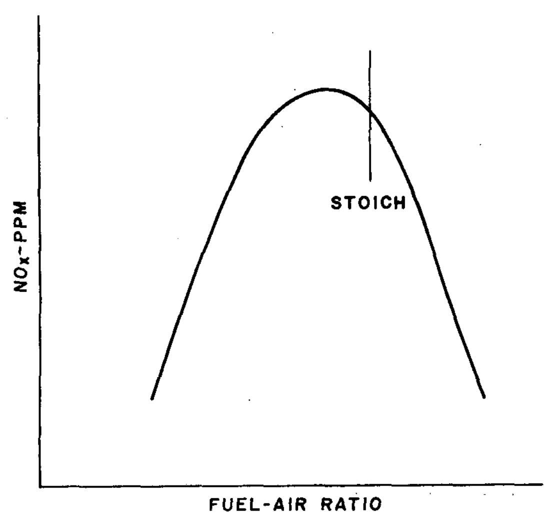


FIGURE 47. FUEL/AIR RATIO

Spark Timing—Spark advance increases peak cycle temperature due to the compression of hot burnt gases. This factor causes NO_X concentration (and mass rate) to increase strongly with spark advance.

Engine Power—An increase in engine power is obtained with higher inlet manifold pressure (lower manifold vacuum). This reduces the residual fraction in the chamber and increases flame speed which increases peak temperature and NO_X concentration. Increased peak pressures also cause an increase in NO_X. Spark retard with increasing load reduces the strength of this effect. The increased exhaust gas flow with higher engine power makes the rate of increase of mass emission rate much higher than the rate of concentration increase.

Engine Speed—Peak combustion temperature rises with increased engine speed because of reduced percentage heat loss. On the other hand, as speed increases more of the combustion process occurs on the expansion stroke at lower temperatures. For very lean mixtures, flame speeds are low and the latter effect predominates, so NO_X decreases with increased engine speed. For rich mixtures, flame speeds are high and the decreased heat loss effect overcomes, with the result being lower NO_X concentration with increased speed. In between, the effects cancel out and the NO_X concentration is more or less unaffected by speed.

Other Effects—Reduction in heat losses by raising the coolant temperature increases NO_X concentration. Compression ratio increase causes higher peak temperatures and higher NO_X levels. Modification of the combustion process by the addition of water (or high ambient humidity), or the addition of inert gas (such as recirculated exhaust gas), reduces peak temperature and NO_X concentration. Figure 48 summarizes the effects discussed above.

Emission Control Systems

The methods used in conventional automobile engines for emission control are briefly discussed below.

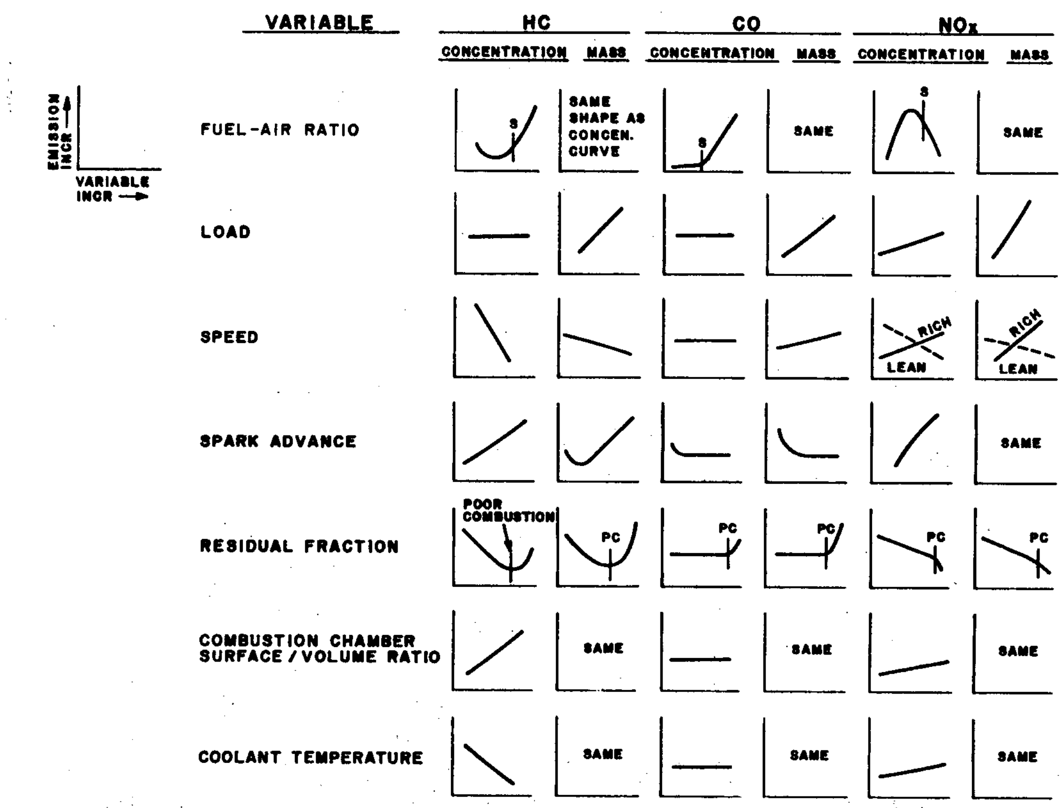


FIGURE 48. EMISSION TRENDS

Positive Crankcase Ventilation—The blowby gases in the crankcase are rich in hydrocarbons and were formerly vented into the atmosphere. The positive crankcase ventilation (PCV) system ducts the crankcase gases through a control valve into the inlet manifold. The PCV valve prevents excessive flow into the manifold at idle, yet allows adequate flow at low manifold vacuum.

Lean Mixtures—The use of a mixture slightly leaner than stoichiometric reduces HC and CO emissions but increases NO_X.

Retarded Spark—This adjustment is effective for both HC and NO_X (unless excessive retard impairs combustion). Power output and efficiency are detrimentally affected.

Reduced Compression Ratio—Again, effective for both HC and NO_X, but this results in a power and efficiency loss.

Reduced Surface-to-Volume Ratio in the Combustion Chamber—This reduces HC but may increase NO_X, due to a decrease in heat loss to the combustion chamber walls.

Increased Coolant Temperature—Reduces HC, but increases NO_X.

Increased Idle Speed and Retarded Spark at Idle—This requires a larger throttle opening at idle, reduces the residual fraction, and reduces HC through a reduction in residual fraction. With these changes, the mixture may be made leaner, which improves CO. Auto ignition at idle is more probable by these changes which results in "dieseling," or continued firing without spark. A solenoid is used to close the throttle upon ignition shutoff to alleviate this problem.

Limited Vacuum Deceleration—During deceleration at high speed, manifold vacuum is high leading to high HC. Devices to prevent complete throttle closing during these conditions improve this situation.

Spark Advance—To obtain maximum spark retard at conditions where the full power of the engine is not required, a solenoid valve prevents the vacuum advance from becoming operative until the vehicle is in direct drive. This has obvious undesirable effects upon fuel economy under certain driving conditions.

Exhaust Gas Recirculation—The introduction of exhaust gas into the combustion chamber reduces the peak combustion temperature and is effective in reducing NO_X. The mechanism by which the reduction occurs is based upon the use of an inert substance to dilute the charge; other gases and water have also been used for this purpose. Significant reductions in the concentrations of nitric oxide formed during the combustion process have been observed for diluted charges that have an increased thermal capacity by comparison with typical air/fuel mixtures. The reduction in nitric oxide formation is attributed to the reduction in flame speed and the lower peak cycle temperature. Although exhaust gas recirculation is effective as a means of reducing oxides of nitrogen, the lower flame speeds result in a time loss which can adversely affect fuel economy. In addition, the use of a richer mixture to overcome driveability problems resulting from slow burning can lower the fuel economy.

Carburetor and Inlet Manifold Modification—Carburetor changes have had as their objective better fuel atomization and mixing and closer control over fuel/air ratio. The latter result has been furthered by air preheating in order to provide a constant summer and winter inlet air temperature. In addition, a modified choke arrangement is used that reduces choking after cold starts. Quicker warmup is achieved by manifold redesign.

Exhaust Reactors—Given adequate temperature, time, and oxygen, CO and HC in the exhaust stream can be oxidized and thereby eliminated. Exhaust reactor systems are designed to provide the necessary requirements for this oxidation process. In general, exhaust reaction may be divided into two categories: those requiring the engine to operate with rich fuel/air mixtures with air injection into the thermal reactor, and those using lean mixture operation without air injection. The latter are more difficult to operate since the quantity of combustibles in the exhaust is lower, and the problem of sustaining the reaction is thereby increased. The former method has the disadvantages of reduced fuel economy due to rich mixture operation, and the added cost and complexity of an air pump.

Neither method of operation reduces NO_X and, in fact, may actually increase NO_X if the reaction temperature in the reactor is high enough.

The design of optimum thermal reactors is not a simple problem; at low loads the reaction must be maintained in spite of lower reactor temperatures, optimum air injection rates are not necessarily the same for CO and HC, and the reaction rate is influenced by exhaust pressure, airflow rate, reactor volume, and reactor temperature. Even so, injection of air into more or less standard exhaust manifolds is used and does result in significant reductions of CO and HC.

Catalytic Reactors—Certain materials act as catalysts in the oxidation of CO and HC and in the reduction of NO_X, although a single material is not necessarily the best for all three functions. Reactions are carried out at temperatures lower than in the thermal reactors. The oxidizing reactions require excess oxygen which may be supplied by lean mixture operation of the engine or by an air pump. The reducing reaction requires rich mixtures (or at least a reducing atmosphere).

Tetraethyl lead, used to increase fuel octane number, has been found to cause deterioation of the performance of the catalyst. In any event, catalyst durability appears to be a major problem. Other problems involve heat conservation: quick warmup of the reactor upon cold starting, and means to prevent overheating at high loads.

Fuel Evaporation Control—Equipment has been designed and installed on production vehicles to route fuel vapors from the fuel tank into the engine during running and into a storage device during hot soaks.

Conventional Automobile Exhaust Emissions and the Federal Emission Standards

Introduction

For the purposes of this study, the Federal Emission Standards of primary interest are the Interim Standards for 1976:

0.41 g/mile for HC emissions

3.4 g/mile for CO emissions

2.0 g/mile for NO_X emissions

The Standards require that the vehicle pass these standards when operated on the CVS-CH test cycle, and that it continue to pass the Standard for 5 yr or 50,000 miles, whichever occurs first.

In this part of the report, various modifications of the conventional engine to reduce exhaust emissions will be considered. The purpose of this review is to show the direction that emission reduction development has taken, with the attendant penalties in fuel economy. It is believed that this review will be enlightening when other power plant types are investigated later in this report.

Engine with Improved Carburetion, EGR, and Air Injection

In this section, the arrangement without exhaust treatment (catalytic or thermal reactors) that has the lowest overall exhaust emissions will be considered.

The modifications include an improved carburetor for lower fuel/air ratio tolerances, density compensation, and early-off choke; a quick-heat inlet manifold; EGR; air injection into the exhaust manifold; modified idling conditions; and rich carburetion. This system has the *potential*[25] of the following emission levels:

HC: 1.5 g/mile

CO: 25 g/mile

NO_X: 1.5 g/mile

With lean carburetion and without air injection, the engine has potential exhaust emissions of:

HC: 1 g/mile

CO: 10 g/mile

NO_X: 2.5 g/mile

It should be emphasized that these values are considered to be the *best potential* emissions that can be achieved without exhaust treatment and they have not been consistently demonstrated. Some prototype systems and some control systems for small automobiles have demonstrated values lower than those already listed at low mileage, and subsequent developments may allow revision of the estimates. However, at the present time, the values are considered valid for the types of vehicles considered in this study.

A fuel economy penalty of approximately 10 to 15 percent, by comparison with an uncontrolled vehicle, will be paid by the rich-mixture version of this arrangement, due to the fuel mixture as well as the EGR. A lesser penalty will accrue to the lean-mixture vehicle due to the leaner mixture, but EGR will still cause fuel economy deterioration. In any event, neither of these arrangements qualifies for consideration as a method to improve fuel economy.

Oxidizing Catalyst System

Automobile manufacturers have concentrated on this type of emission control for 1975 Federal Standards (HC: 1.5 g/mile; CO: 15 g/mile; NO_X: 3.1 g/mile). It makes use of improved carburetion, a quick-heat manifold, EGR of about 10 percent, an air pump to inject air into the exhaust ports for HC and CO reduction, and a catalytic converter in the exhaust system to further HC and CO oxidation. It is almost certain that such a system will be produced by 1975 that will meet the required standards for that year. Low-lead gasoline is a requirement for adequate catalyst durability.

[25] "Report by the Committee on Motor Vehicle Emissions," National Academy of Sciences, 1973.

There is some evidence that the use of catalytic reactors on 1975 model year vehicles will result in an improvement in fuel economy by comparison with 1973 model year vehicles due to the opportunity for changes in spark control. However, specific data concerning the magnitude of the improvement or the implications for subsequent model years are not available.

Dual-Catalyst System

For reduction in NO_X to levels considerably below the 1975 level, automobile manufacturers are concentrating on the dual catalyst system in which the exhaust gas is first passed through a NO_X catalyst reactor under reducing exhaust gas conditions (richer than stoichiometric fuel/air mixture). An air pump then adds air to the exhaust stream and the exhaust gas, now oxygen rich, passes through the CO and HC oxidizing catalyst; lead-free fuel is required.

At this time, the NO_X catalyst suffers from durability inferior to that of the oxidizing catalyst, and the fuel/air ratio must be maintained within close limits to prevent ammonia formation in the NO_X catalyst.

Effect of Emission Controls Upon Fuel Economy

The emission control devices now used and those planned for future use have a strong effect on engine performance and fuel economy. EGR reduces flame speed, increases cycle-to-cycle combustion variation, and leads to poor driveability. A remedy used for this problem is an increase in the fuel/air ratio which leads to higher fuel consumption. Reduction in compression ratio to accommodate lower octane gasoline causes deterioation in performance and fuel economy. Increased engine displacement to recover lost performance further increases fuel consumption due to engine operation at lower (and less efficient) loading.

In the discussions to follow that are concerned with engine and vehicle modifications, it is impossible to avoid consideration of the effect of emission controls upon fuel economy of these modifications. The base of comparison required by this contract makes the task somewhat more comples, since all modified vehicles synthesized in this work are to meet the 0.41–3.4–2.0 (hydrocarbons, carbon monoxide, nitrogen oxides in g/mile, respectively) emission standard and are to be compared against the standard vehicle meeting the 1973 emission standard. In this case, the standard vehicle is a 4300-lb vehicle with a 350-CID carbureted spark-ignition engine substantially identical to the comparable 1973 vehicles tested in this program.

In order to perform the comparison of fuel economies in a repeatable and logical manner, the following technique will be used. The terms below are first defined:

A – fuel economy (miles/gal) of the synthesized vehicle meeting the emission standard of 0.41–3.4–2.0.

B – fuel economy of the standard vehicle meeting the 1973 emission standard

C – fuel economy of the synthesized vehicle with uncontrolled emissions

D – fuel economy of the standard vehicle with uncontrolled emissions

E – fuel economy of the standard vehicle meeting the 0.41–3.4–2.0 emission standard.

The desired comparison is the ratio $\frac{A}{B}$.

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

The value of the C/D can be obtained by analysis of engine fuel maps, road load, vehicle weight, accessory load, and so forth, over a chosen driving cycle for both the synthesized vehicle and the conventional vehicle.

The value of A/C will, in most cases, be chosen after consideration of the factors involved in the synthesized vehicle that tend to increase or decrease the difficulty in achieving the desired emission levels and, hence, influence the fuel economy. In most cases, the factor A/C is an *educated guess* since actual data for emissions versus economy frequently are either not available or are for operating conditions widely different from those we are concerned with. In those cases where emissions versus economy data are available and reliable, they have been used.

The value of the factor D/B is somewhat controversial in that different sources give different values. This is not surprising in view of the widely varying test conditions (different driving cycles, different vehicle weights, different accessory loads, etc.) not only between different observers but between the same observer for different model years, as well as the sometimes small samples of vehicles tested. Based upon conversation with representatives of Ford Motor Co., General Motors Corp, Chrysler Corp. and from data published by the National Academy of Sciences and the EPA, the value for the ratio of D/B was chosen to be 1.09. Another ratio that will be used in this report is the ratio of the fuel economy of the standard vehicle with uncontrolled emissions to the fuel economy of the standard vehicle with emissions controlled to the level of 0.41–3.4–2.0 (D/E). The value of this ratio is even less well established than the ratio D/B since fewer data are available, but based upon the information available, we have chosen the value of this ratio to be 1.15.

9. EMISSION CONTROL HARDWARE

In order to accomplish the emission control goals specified in legislation, the vehicle manufacturers have employed a wide variety of devices and systems. It should be helpful to examine the physical aspects of these devices in some detail. Emission control systems are tailored to the individual needs of each engine-chassis combination; therefore, an exhaustive treatment of all available systems is not possible. However, there is considerable similarity between the devices used by different manufacturers for the same purpose; representative examples can be selected. Because of the extreme public interest in emission controls, most manufacturers have included comprehensive descriptions of control systems for individual vehicles in the service manuals; these documents are excellent sources of detailed information on control systems and devices.

In the following paragraphs, typical examples of the various devices will be described. It should be noted that the names of some components may vary among manufacturers, but with a rudimentary background, the devices may be recognized according to function from descriptions in the service manuals.

Positive Crankcase Ventilation

The positive crankcase ventilation (PCV) system is among the oldest of emission control devices; installation was common in the middle of the last decade. Prior to that time, it was common practice to vent the crankcase to the atmosphere, and forced displacement of hydrocarbon laden crankcase gases was achieved by menas of a draft tube which protruded into the airstream below the vehicle. With the advent of the PCV system, ventilation air is supplied to the crankcase from the air cleaner, and the resulting mixture is routed to the carburetor or intake manifold. The manifold vacuum provides the driving force for the flow. The contaminated gases are thus subjected to the combustion process in the engine cylinders before being released to the atmosphere. A check valve (the PCV valve) is installed in the system to prevent the reverse flow of combustible mixture into the crankcase during conditions such as backfire or engine cranking. A typical system is shown in Figure 49. The nature of the PCV system is such that little effect on fuel economy should be observed. Losses in the system should be offset by the addition of combustible material to the carburetor inlet.

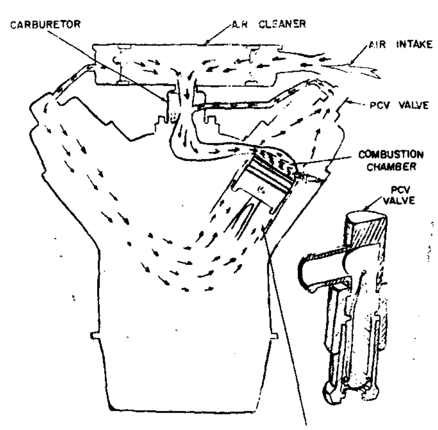


FIGURE 49. CLOSED CRANKCASE VENTILATION SYSTEM

Lean Mixture Operation

Considerable effect on emissions, particularly of hydrocarbons and carbon monoxide, can be achieved by carefully controlling the air/fuel ratio supplied to the engine. In the past, performance was the main measure of mixture adjustment; there was considerable variation in carburetor setting, even among identical vehicle models. When emission control achieved a high priority, it was found that more precise carburetor calibrations were required. Current production carburetors are individually tested on a flow bench; critical adjustments are made and seals are applied to discourage adjustment by unqualified personnel.

In general, lean mixture operation has a beneficial effect on fuel economy, hydrocarbon emissions, and carbon monoxide emissions. A detrimental effect on emissions of oxides of nitrogen can be expected.

An emission control system now being investigated by several groups for possible future use would employ a sensor to detect the oxygen content of the exhaust stream. The output of the sensor would be used to control the air/fuel ratio.

It should be observed that lean mixture operation places additional requirements on the vehicle ignition system. Since lean mixtures are more difficult to ignite and have a lower flame speed, the spark timing, spark duration, and spark intensity are all affected.

Retarded Spark

Emissions of both hydrocarbons and oxides of nitrogen can be controlled by retarding the spark with respect to values common to pre-emission control engines. Traditionally, carburetors have been designed to provide rich mixtures at idle and wide open throttle for performance and driveability; lean mixtures are supplied at part throttle for economy. Flame propagation characteristics are quite different for rich and lean mixtures. It is customary to supply the spark at an earlier time for the slower burning lean mixtures in the economy cruise mode. This spark advance is obtained by changing the orientation of the contact points in the distributor with respect to a top dead center location. An input proportional to engine speed is supplied by rotating weights in the distributor, and an input proportional to engine load is obtained from the manifold vacuum through a diaphragm attached to the distributor. Each engine, therefore, has an "advance curve," or a relationship between the point in the cycle at which the spark occurs, engine speed, and load. In pre-emission control applications, manufacturers' specifications called for spark advance settings as high as 10 deg at idle; the speed and load advance mechanisms might cause ignition as much as 30 deg in advance of top dead center in an economy cruise condition.

It has been determined that emission control can be effected by retarding the spark or changing the spark advance characteristics that have been determined over a period of years, with performance as the objective. The idle setting has been reduced, by most manufacturers, to the vicinity of top dead center, and the "advance curve" has been modified by the addition of control devices. These devices function during various operating modes of the vehicle where emission control is found to be a problem. Typical examples are described in the following paragraphs.

Electronic Ignition

The conventional ignition system employs a mechanical switch (the contact points) to supply the signal for spark initiation. A more precise and reliable system can be assembled using noncontacting magnetic omponents; one manufacturer has adopted this technique, and the other major companies can be expected to follow in the near future. The two systems are illustrated in Figures 50 and 51. The electrical signal supplied by the magnetic sensors can be subjected to various forms of electronic logic with input from other sensors of engine operating condition. The resulting modified signal is used to direct the timing of the ignition spark.

Distributor Solenoid

In order to provide adequate spark advance for good starting capability with carburetor settings appropriate to good idle emissions, some vehicles are equipped with electrical solenoids to allow

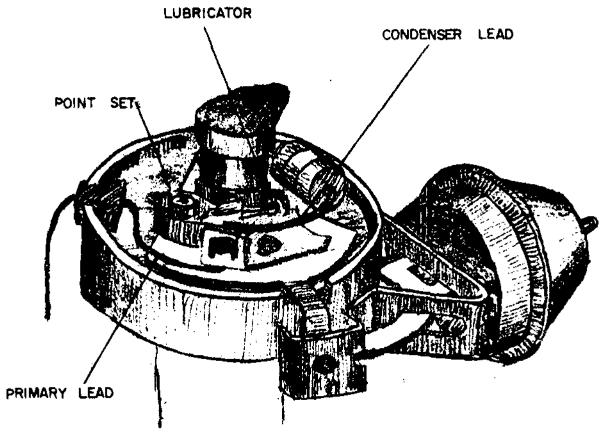


FIGURE 50. CONVENTIONAL DISTRIBUTOR

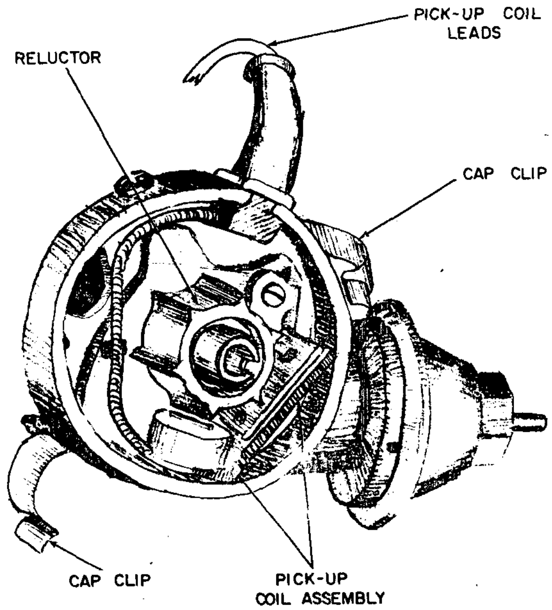


FIGURE 51. ELECTRONIC IGNITION DISTRIBUTOR

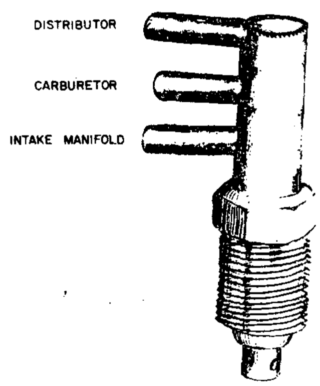


FIGURE 52. THERMAL VACUUM SWITCH (T.V.S.)

additional spark advance during engine cranking. The solenoid may be located within the vacuum advance diaphragm unit attached to the distributor.

Vacuum Delay Valve

In some situations, such as the transition from idle to part throttle, it is desirable to delay the spark advance characteristics of the ignition system. A small orifice is placed in the vacuum line to accomplish this result. Most such valves are also temperature sensitive; spark advance control becomes a function of ambient temperature.

Thermal Vacuum Switch

The distributor vacuum advance is also controlled by engine water temperature on most vehicles. The retarded spark settings usually used at idle for emission control can result in excessive coolant temperatures; when this possibility exists, the thermal vacuum switch allows the spark to advance. The switch is typically inserted in the cooling water jacket as shown in Figure 52. On some vehicles, a switch is located on the vehicle transmission which controls spark advance as a function of drive gear. Thus, the spark may be allowed to advance when the vehicle is in high gear, but spark retard is maintained during first and second gear operation.

Idle Stop Solenoid

During recent model years, idle speeds have increased progressively. If the ignition is turned off with the engine at idle, there is a tendency toward "after running" or "dieseling"; that is, the engine continues operation with erratic and arbitrary ignition. The idle stop solenoid is used to hold the throttle linkage at the desired idle condition while the ignition is on; when the ignition is turned off, the linkage assumes a low idle position and the engine stops. The solenoid is located in the vicinity of the carburetor; the plunger contacts the throttle linkage.

Choke Heating Element

On vehicles equipped with automatic chokes, a system is included to decrease the extent of choke operation as the engine warms. On some models, the choke mechanisms have been equipped with auxiliary electric heaters to hasten the removal of choking function after the engine is started at high ambient temperatures. The choke heaters vary in configuration; they are generally located near the automatic choke linkage at the side of the carburetor.

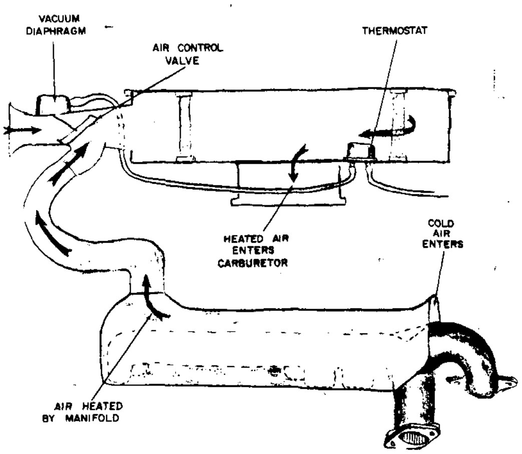


FIGURE 53. HEATED INLET AIR SYSTEM

Heated Inlet Air System

It has been found that hydrocarbon emissions may be decreased by supplying air to the engine at a temperature of about 100°F. Toward this end, a diverter valve has been installed in the snorkel of the air cleaner housing; air is gathered from the vicinity of the exhaust manifold when the underhood temperature is less than a prescribed desirable value. The amount of heated air varies with temperature. A typical system is shown in Figure 53.

Exhaust Gas Recirculation (EGR)

Exhaust gas recirculation is employed on late model vehicles as a means of reducing emissions of oxides of nitrogen. In principle, burned gases are removed from the exhaust system and supplied to the engine air inlet. In practice, this function is performed by a control valve, usually located at the carburetor, which controls the quantity of recirculated gas. The control signal for the EGR valve is typically a vacuum, but the source of the vacuum varies among engine categories. The vacuum signal may or may not be amplified, and ambient temperature compensation may or may not be included. A typical system configuration is shown in Figure 54.

Air Injection

On some vehicles, a system is included for supplying air to the exhaust manifold to assist with the oxidation of unburned hydrocarbons and carbon monoxide. The system consists of a belt-driven air pump, a regulating valve, and a manifold for distribution of the air supply. The regulating valve senses manifold vacuum, and serves both as a check valve to protect the pump from hot gases and as a bypass valve to prevent backfire on deceleration. A diagram of the air injection system is shown in Figure 55.

Evaporative Emission Control

Federal requirements now limit the quantity of hydrocarbon emissions from sources other than the exhaust. Fuel tanks and carburetors on new cars are vented to the atmosphere through a filter, usually a canister containing activated charcoal. When the engine is not operating, fuel vapors are adsorbed by the material in the canister. When the engine is operating, the hydrocarbons are removed from the canister material by heat from the engine compartment; the evolved vapors are conducted to the carburetor and thence to the combustion chamber. A diagram of the system is shown in Figure 56.

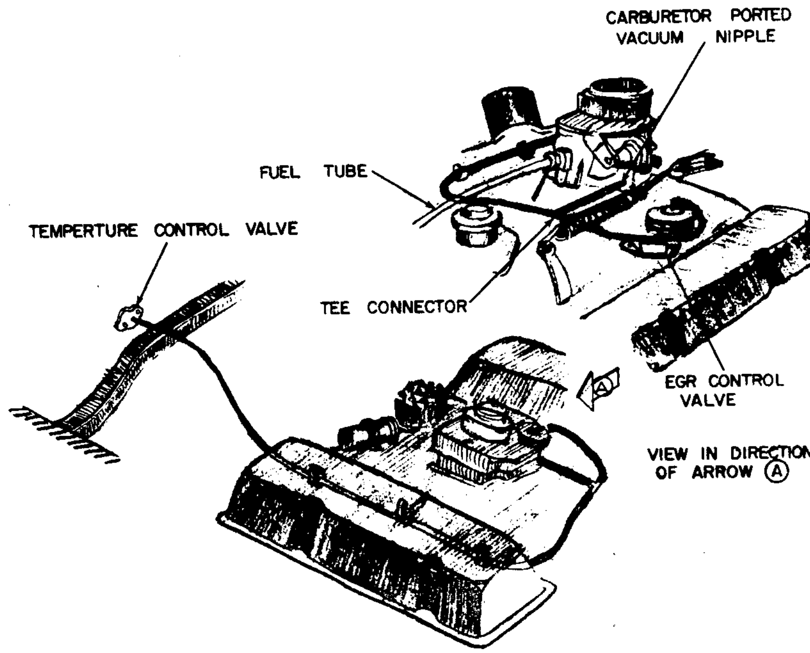


FIGURE 54. EGR SYSTEM

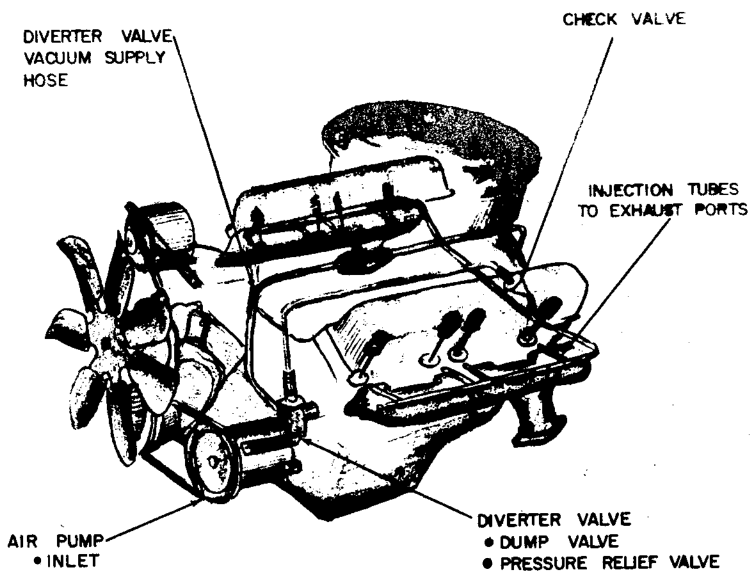


FIGURE 55. AIR INJECTION SYSTEM (8-CYLINDER ENGINE)

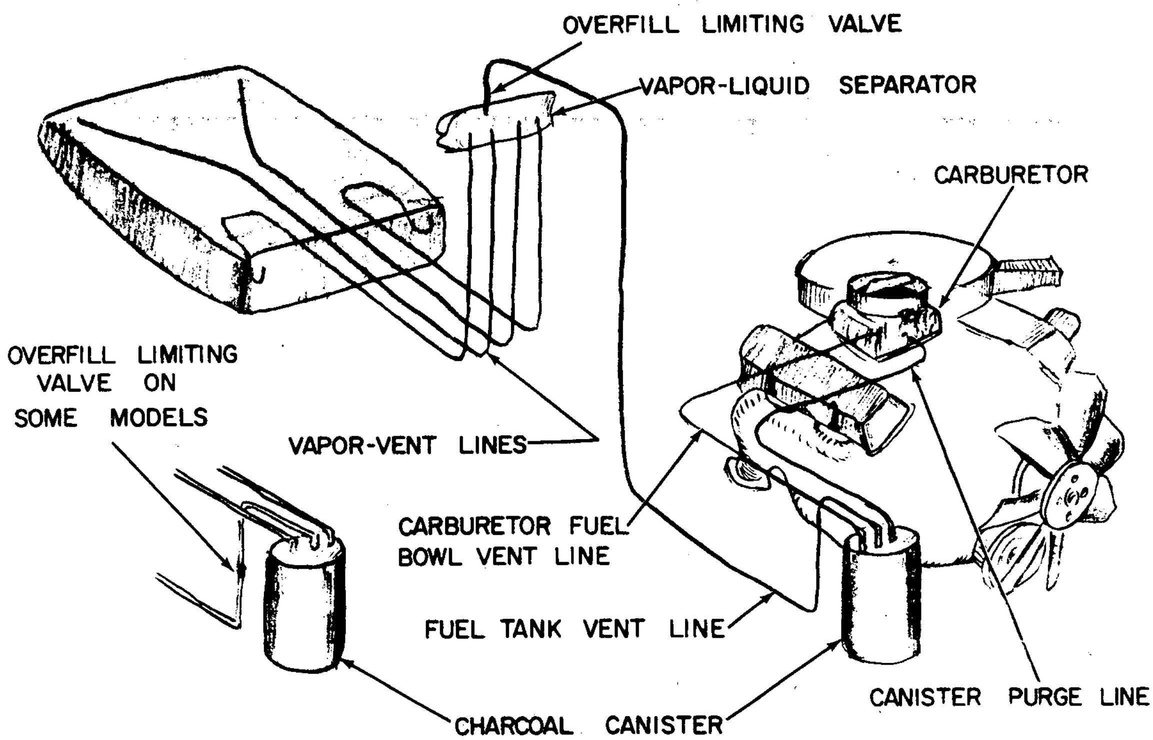


FIGURE 56. EVAPORATION CONTROL SYSTEM

Exhaust Reactors

The devices described above are used to limit the emissions from an engine. Another category of systems, exhaust reactors, can be used to control concentrations of contaminant substances in the exhaust system.

Thermal Reactors

Thermal reactors can best be described as insulated exhaust manifolds. Hot gases leaving the exhaust ports of the engine are allowed to dwell in the manifold, and excess air is supplied with an air injection system. Combustible materials, therefore, have an opportunity to oxidize before leaving the exhaust. Thermal reactors must be located at the exhaust port of the engine, and they are characterized by operating temperatures on the order of 2000°F. The only significant production of thermal reactors to date has been in connection with the Mazda rotary engine.

Catalytic Reactors

Another type of reactor, which operates on a catalytic conversion principle, may be used in the exhaust system. Exhaust gases which are capable of reacting are exposed to a suitable catalyst, and the reaction proceeds at an advanced rate. Much of the effort expended on reactor research has been directed toward the development of a completely suitable catalyst. Many substances are available, but the best performance has been obtained from noble metals such as platinum and palladium. It should be noted that catalysts can serve several purposes, and the same material can serve as a catalyst either for reducing oxides of nitrogen in an oxygen free atmosphere or for oxidizing hydrocarbons and carbon monoxide in an oxygen rich atmosphere. Over a specific, very narrow range of the air/fuel ratio, a single catalyst bed can be used to diminish the concentrations of all three contaminant substances.

The previous discussion, although not exhaustive, should serve as an outline for those interested in pursuing the specific emission control devices used on a particular vehicle model. A numerical relationship for predicting the degree of emission control that could be expected from each device would be desirable, but such a treatment would be enormous in scope even if sufficient data were available. There is considerable interaction between the devices used in any emission control strategy. The contribution of an individual device to contaminant control depends upon the specific engine design, the other devices used, and the tuning of the entire system. Typically, a control strategy for a particular model is determined from tests involving several devices used with that engine, and the most appropriate combination is selected and submitted for certification.

10. TURBOCHARGED SPARK-IGNITED, CARBURETED ENGINE

Concept

In order to provide what is considered to be adequate power for acceleration, present-day automobile engines have the capability for much higher power output than is necessary for normal driving. Consequently, "normal" driving (without high power accelerations) requires the engine to operate at relatively low torque levels.

On the other hand, the spark-ignition carbureted engine has its minimum brake specific fuel consumption at high bmep (torque). This characteristic is shown in Figure 57 and is due to the lower percentage of friction loss as power is increased at constant speed. Therefore, the conventional automobile engine operates under normal conditions at relatively high brake specific fuel rates.

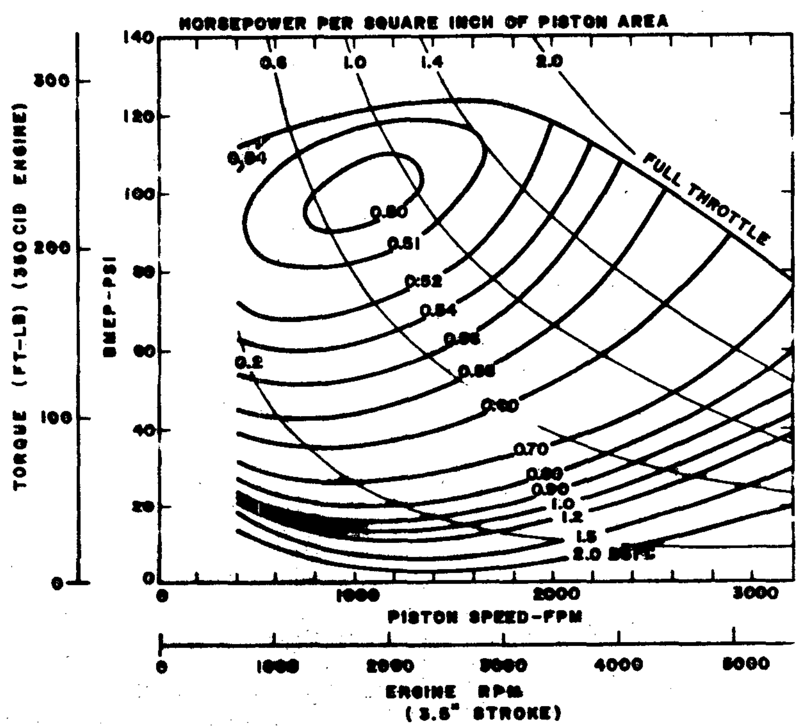


FIGURE 57. PERFORMANCE MAP OF CARBURETED SPARK IGNITION ENGINE

The use of a smaller engine would, of course, alleviate the fuel consumption problems for normal driving (since the operating bmep would be increased), but there would be no capability for the acceleration levels previously obtained with the larger engine. Maximum vehicle speed would also be limited.

A possible solution lies in the use of a smaller engine with a turbocharger. The reduction in engine displacement requires that normal driving be done at a higher percentage of full power in areas of lower fuel consumption. The turbocharger provides sufficient power boost for acceleration and high vehicle speeds.

On the other hand, higher air inlet temperature (due to the pressure boost) increases the knock tendency of the engine. This is exemplified in the necessity for high-octane gasoline and water-alcohol injection in turbocharged aircraft engines, and the use of alcohol fuel (with a hich octane number) in turbocharged racing engines.

The problem of exhaust emissions is by no means eliminated by the use of turbocharging. Since fuel-air ratios must remain substantially the same in both turbocharged and naturally-aspirated (NA) engines, and other details of the combustion chamber, exhaust system, valving and spark advance are basically the same, one would expect that HC and CO emissions would be approximately equal in both engines. The increased peak temperature in the combustion chamber would be expected to raise NO_X levels to some extent. (The turbocharged engine has no inherent tendency to provide EGR automatically, since the ratio of exhaust pressure to intake pressure is very nearly unity.)

The above discussion assumes that the compression ratio of the engine is not changed when turbocharging is added, nor is the pressurized inlet air-cooled by a heat exchanger (aftercooler). As will be shown later, the horsepower output of the larger NA engine can be obtained in the smaller turbocharged engine, *without knock*, by several methods:

- Reduced engine compression ratio—This change, of course,increases the brake specific fuel consumption of the engine due to loss of engine efficiency.

- Aftercooling, with slightly reduced compression ratio—Here the efficiency loss is minimized by air-cooling, but since the aftercooler cannot be 100 percent efficient (and, therefore, the inlet air temperature in the turbocharged engine will be higher than that of the NA engine), some reduction in the compression ratio is necessary.

- Increased fuel octane number—It is considered that this step is not acceptable, due to impending Federal limitation on the use of TEL and the far-reaching effect on the refining industry of a requirement for large quantities of high octane fuel.

- Injection of water alcohol—This mixture, injected at the high bmep's where knock occurs, has a powerful knock-reducing tendency.

- Spark retard—Increased torque output can be obtained from a given engine by retarding the spark and increasing the manifold pressure to the point of incipient knock.

- Rich fuel-air mixtures—It is possible to use fuel as an antidetonant in order to obtain a knock-free operation.

Performance Evaluation Procedure

To evaluate this concept, a conventional 350-cu in. displacement (naturally-aspirated) engine was used as a reference. Power required for normal driving and maximum acceleration of a 4300-lb automobile was determined from test data, and the engine speeds at which this power was needed were fixed.

Next, the size of the turbocharged engine was chosen. The smaller the displacement of this engine, the higher the bmep required and the more difficulty anticipated from engine knock. On the other hand, as the displacement of the turbocharged engine increased toward that of the NA engine, the less the anticipated improvement in fuel economy.

The engine picked was a V-8 arrangement with a stroke-to-bore ratio of 1.0. Initially, the compression ratio was maintained at 8:1 to prevent any losses in thermal efficiency to this cause.

Pressure boost, bmep, and bsfc were calculated from the following general relations:

$$\text{bmep} \propto \rho \eta_v F h_v \eta_t \eta_m \tag{1}$$

where

ρ – inlet density of fuel/air mixture

η_v – volumetric efficiency

F – fuel/air ratio

h_v – fuel heating value

η_t – thermal efficiency

η_m – mechanical efficiency

$$\eta_m = \frac{\text{bmep}}{\text{bmep} + \text{fmep}} \tag{2}$$

where fmep = friction mean effective pressure

$$\text{bsfc} \propto \frac{1}{\eta_t \eta_m} \tag{3}$$

then,

$$\frac{\text{bmep}_1}{\text{bmep}_2} = \frac{\rho_1 \eta_{v_1} F_1 h_{v_1} \eta_{t_1} \eta_{m_1}}{\rho_2 \eta_{v_2} F_2 h_{v_2} \eta_{t_2} \eta_{m_2}}$$

$$\frac{\text{bsfc}_1}{\text{bsfc}_2} = \frac{\eta_{t_2} \eta_{m_2}}{\eta_{t_1} \eta_{m_1}}$$

Volumetric efficiency is only slightly changed by increased inlet air pressure, so $\eta_{v_1} \approx \eta_{v_2}$. Further, $F_1 = F_2$, $h_{v_1} = h_{v_2}$ (same fuel), and the thermal efficiency is only affected a negligible amount by boosting the inlet pressure if the compression ratio is unchanged. Therefore,

$$\frac{\text{bmep}_1}{\text{bmep}_2} = \frac{\rho_1}{\rho_2} \frac{\eta_{m_1}}{\eta_{m_2}} \tag{4}$$

$$\frac{\text{bsfc}_1}{\text{bsfc}_2} = \frac{\eta_{m_2}}{\eta_{m_1}} \tag{5}$$

for the case where the compression ratio is unchanged.

The control system chosen for the turbocharger involves a "waste gate" which is a valve in the exhaust pipe upstream of the turbocharger turbine that, when open, diverts exhaust gas around the turbine and limits the turbocompressor speed. The motion of the waste gate is controlled by manifold vacuum, so that low vacuum (high engine output) closes the waste gate and increases inlet boost. A secondary control limits the boost pressure to a chosen maximum to prevent excessive bmep and possible engine destruction.

At this point, a reference "fuel map" for a naturally-aspirated engine of the same displacement as the turbocharged engine was chosen. The maximum bmep-piston speed curve was calculated based upon Equation (4), the boost control characteristics, and the reference "fuel map." The lines of constant bsfc for the turbocharged engine were calculated from Equation (5). In each case, the two subscripts in these equations referred to the NA engine and the turbocharged engine, respectively.

As was mentioned earlier, a primary concern with this technique must be the increased knock tendency of the turbocharged engine. To evaluate this, the technique described by Chen[26] was used, combined with experimental data from Barber[27]. In brief, this technique uses the theory that engine knock is primarily a function of the temperature of the end gas (last gas to burn), and that a satisfactory estimate of this temperature can be obtained by calculating the adiabatic compression temperature of the gas when compressed from inlet pressure to the maximum combustion pressure. Maximum combustion pressures were estimated from Taylor.[21]

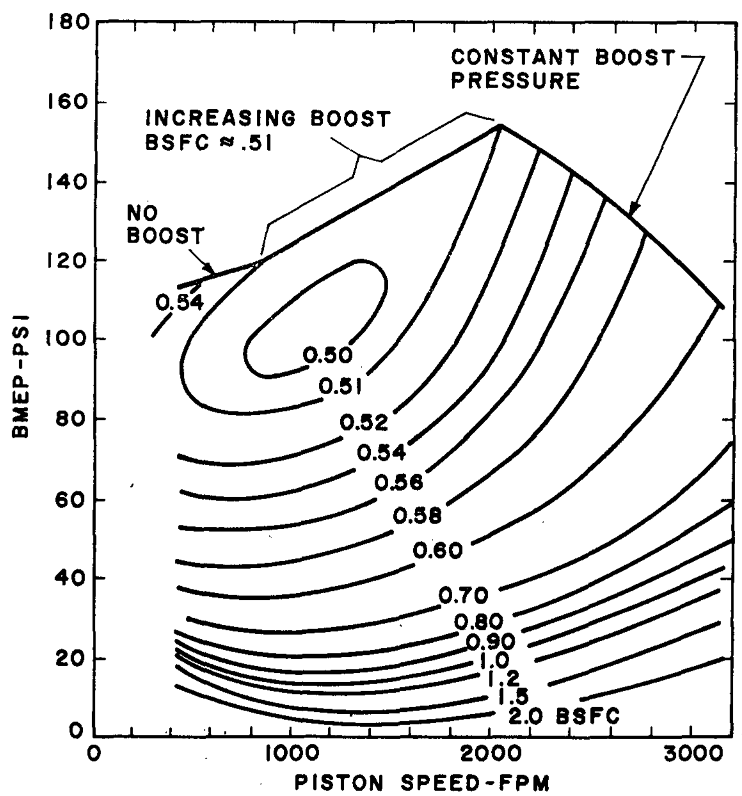


FIGURE 58. TURBOCHARGED CARBURETED ENGINE, 250 CID, CR = 8.0, WATER-ALCOHOL INJECTION

Results

Case I: 250 CID Turbocharged Engine, No Aftercooler, 8:1 Compression Ratio

The computed fuel map for this engine is shown in Figure 58. No boost pressure is acquired until a piston speed of 900 fpm is reached due to low turbocompressor speed. Bmep then increases linearly until the maximum bmep is reached at a piston speed of 2040 fpm. This bmep provides the same power as is attained in the 350 CID NA engine. As piston speed increases thereafter, boost is constant at a pressure ratio of 1.45 (a density ratio of 1.25) across the compressor. The specific fuel consumption of the engine at bmep lower than about 100 psi is unaffected by the turbocharger, since the waste gate is open and boost is essentially zero.

This fuel map was used to compute fuel economy using the procedure described in Chapter 3 of this report.

[26] Chen, T. N. and R. N. Alford, "Combustion of Large Gas Engines," ASME Paper 71-DGP-6, 1971.
[27] Barber, E. M., "Knock-Limited Performance of Several Automobile Engines," SAE Transactions, July 1948.

The adiabatically-compressed temperature of the end gas at the maximum bmep point is about 1720°R. This is higher than the knock-limiting temperature of 1480°R (calculated from the 250 CID NA engine at maximum bmep). The chosen turbocharged engine is without doubt above the knock-limited power at the maximum condition, which merely serves to verify the experience of aircraft and racing engine manufacturers.

The solution for this case is the use of water-alcohol injection in an amount of about 0.5 lb mixture/lb fuel. Injection is into the inlet manifold and is necessary only when the engine operates in the boosted range (above about 115 bmep). This technique was used in limited production by Oldsmobile in 1962 to 1963 in a 215 CID turbocharged, V-8 automobile engine put on the general market. The technique has evident service and maintenance problems, since failure of the injection system can be catastrophic to the engine under certain operating regimes. However, the technical feasibility of the water-injected arrangement is not in doubt.

The fuel economy (miles/gal) of the 250 CID turbocharged engine was calculated to be 17 percent greater than that of the reference 350 CID naturally-aspirated engine. Since this comparison is based upon both engines having uncontrolled emissions, it was necessary to estimate the effect of the required emission controls upon the fuel consumption comparison. It is desired to compare the turbocharged engine with a 0.41–3.4–2.0 emission level against the naturally-aspirated engine with a 1973 emission level. This was done using the following equation:

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

where

A – fuel economy of 250 CID turbocharged engine meeting emissions standards of 0.41–3.4–2.0 (miles/gal)

B – fuel economy of 350 CID naturally-aspirated engine meeting 1973 emission standards (miles/gal)

C – fuel economy of 250 CID turbocharged engine, uncontrolled emissions (miles/gal)

D – fuel economy of 350 CID naturally-aspirated engine, uncontrolled emissions (miles/gal)

From previous calculations,

$$\frac{C}{D} = 1.17$$

From published information and private communications, it is estimated that the ratio D/B = 1.09.

From the discussions in the following section, it is apparent that the problems of reducing emissions in the turbocharged engine will not be much different than with the conventional engine. From a number of sources, it is estimated that the loss in fuel economy from the uncontrolled conventional engine to the same engine meeting the 0.41–3.2–2.0 standards is 15 percent. Using this value for the turbocharged comparison, A/C = 1/1.15. Therefore,

$$\frac{A}{B} = (1.17)\left(\frac{1}{1.15}\right)(1.09) = 1.11$$

Case II: 280 CID Turbocharged Engine, Aftercooled, 7.2:1 Compression Ratio

Since water-alcohol injection to prevent knock is an undesirable addition to the engine, the use of an aftercooler was investigated. The use of engine coolant in the aftercooler was found to be not feasible, due to the relatively small difference between coolant temperature and inlet air temperature. An air-to-air aftercooler was chosen, using ambient air for the coolant. Aftercooler effectiveness (E) was taken as 0.7 where

$$E = \frac{T_b - T_c}{T_b - T_w}$$

T_b – air temperature before aftercooler

T_c – air temperature after aftercooler

T_w – coolant "in" temperature

The use of an aftercooler did not reduce end gas temperature enough to preclude knock, so the compression ratio was reduced so that knock was incipient at the maximum engine bmep. The new compression ratio was 7.2 (reduced from 8). The choice of engine displacement was then affected by two parameters–the boost pressure and the efficiency loss due to decrease of compression ratio. Calculations showed that with the same boost pressure as Case I, the displacement must increase to 280 CID to provide the required maximum power.

Figure 59 is the resulting performance map for this arrangement. This engine is not knock limited at the maximum bmep shown on the map with regular grade fuel and does not require water injection.

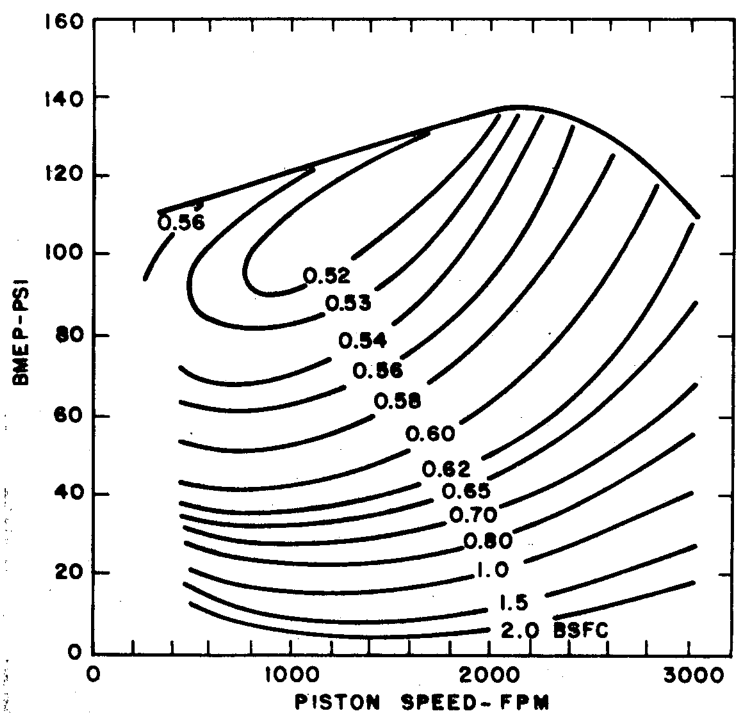


FIGURE 59. TURBOCHARGED CARBURETED ENGINE, 280 CID, CR = 7.2, AFTERCOOLED

Fuel economy was calculated for this engine using the standard procedure. Results showed a 10-percent improvement over the reference 350 CID, NA engine.

Using the reasoning already discussed,

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

where

A – fuel economy of 280 CID turbocharged engine meeting the 0.41–3.4–2.0 emission standard

B – fuel economy of 350 CID naturally-aspirated engine meeting 1973 emission standards

C – fuel economy of 280 CID turbocharged engine, uncontrolled emissions

D – fuel economy of 350 CID naturally-aspirated engine, uncontrolled emissions

As calculated above, C/D = 1.10. From the previous section, D/B = 1.09, and A/C = 1/1.15, so

$$\frac{A}{B} = (1.10)\left(\frac{1}{1.15}\right)(1.09) = 1.04$$

Case III: Turbocharged Engine with Compression Ratio Reduced to Prevent Knock, No Aftercooling

In an attempt to eliminate the aftercooler without using water or water-alcohol injection to prevent knock, the compression ratio of the engine was reduced to the point where no knock would occur with regular grade gasoline with a boost pressure ratio of 1.45. This compression ratio was 5.

The loss of thermal efficiency and power was so severe due to this change in compression ratio that the displacement had to be set at 340 CID to return the engine to the 350 CID, naturally-aspirated power output. Obviously no fuel economy gain can be expected with this arrangement, in fact, it is certain that a loss in fuel economy will occur since the displacement is hardly reduced from the naturally-aspirated engine, while the compression ratio has been significantly reduced.

This is hardly unexpected, since it again confirms experience and experimental data. When no change in octane number is allowed and no aftercooling or other knock control methods are used, the use of turbocharging is not conducive to performance improvement in spark-ignition engines.

Exhaust Emissions

As mentioned earlier in this section, it would not be expected that turbocharging would have any major effects upon engine exhaust emissions, when two engines of equal maximum output are considered—one turbocharged and one naturally-aspirated. Fuel/air ratios and exhaust mass flow rate are essentially the same in both engines; combustion chamber geometry is similar; and size effects are small. The only exception is that in the turbocharged engine without aftercooling, and with a compression ratio the same as the NA engine, peak combustion temperatures will be higher and a slight increase in NO_X would be expected.

Schweikert and Johnson[28] arrived at somewhat similar conclusions after testing a turbocharged 307 CID engine without aftercooling, both with and without EGR. The engine compression ratio was 8.5-unchanged from the standard engine. Knock was not encountered because the load did not exceed 60 bmep. Conclusions were that without EGR, emissions are about equal in turbocharged and NA engines with the same maximum power output. With EGR in both engines, CO is unchanged, HC is increased slightly in the turbocharged engine, and NO_X is increased about 20 percent in the turbocharged engine. These latter results are not altogether consistent with theoretical considerations and lead one to expect that other factors (poor fuel distribution, poor EGR distribution, back pressure effects not considered in this work, etc.) contribute to the rather large increase in NO_X.

In the turbocharged engine of Case I (unchanged compression ratio, use of water-alcohol injection), NO_X should be less than, or at the most, equal to that of an NA engine of same maximum power. Water injection is a powerful means for reduction of NO_X.

In the turbocharged engine of Case II (reduced compression ratio, aftercooled), all emissions should be very nearly equal to that for an NA engine with the same maximum power output, since maximum combustion temperatures are limited to those of the NA engine.

The conclusion to be drawn is that the turbocharged engine will require the same emission controls as the NA engine to meet the same emission standards. Therefore, either of the turbocharged engine versions will require EGR and oxidation reactors (either thermal or catalytic), in addition to close control of fuel/air ratio and starting and warmup improvements.

Evaluation

Engine Description

In this section, two engines will be compared to the conventional carbureted engine of equal power. Both are turbocharged, and the first has a displacement of 250 cu in., uses water-alcohol injection for knock suppression, and has a compression ratio of 8. It will be designated as the 250 CID-TC engine. The second has 280 CID, employs air aftercooler, and has a compression ratio of 7.2. It will be designated as the 280 CID-TC-A engine.

Fuel Economy

As compared to the fuel economy of the 350 CID conventional engine with 1973 emission levels, the fuel economies of the two engines under consideration, both meeting a 0.41–3.4–2.0 emission standard, are as follows:

250 CID-TC: 11 percent better than the 350 CID conventional

280 CID-TC-A: 4 percent better than the 350 CID conventional

Performance

Each of the turbocharged engines have the same maximum power as the 350 CID naturally-aspirated engine, so vehicle top speeds are the same for all. However, the inertia of the turbocharger

[28] Schweikert, J. F. and J. H. Johnson, "A Turbocharged Spark Ignition Engine with Low Exhaust Emissions and Improved Fuel Economy," SAE Paper 730633, 1973.

wheel makes it likely that the acceleration performance of the turbocharged engines will be inferior to the conventional engine.

Emissions

For reasons already discussed, the problem of reducing emissions is probably not much affected either way by the addition of a turbocharger. The fuel economy estimates takes this factor into account.

Noise

The addition of a turbocharger increases high frequency engine noise. Much of this noise can be attenuated by sound absorption material in the engine compartment, but overall noise levels of the turbocharged engine will be higher.

Weight and Size

Based upon considerations of engine displacement, turbocharger weight, and noise attenuation materials, the following estimates are made:

250 CID-TC:	75 percent of conventional 350 CID engine weight, same box volume
280 CID-TC-A:	90 percent of conventional 350 CID engine weight 110 percent of conventional 350 CID engine box volume

Reliability and Maintainability

The addition of a turbocharger and the necessary turbocharger controls will have a small detrimental affect upon engine reliability. The further addition of an aftercooler and associated blower will further reduce reliability.

A more serious problem may be the possible failure of the knock-suppressant systems. In the 250 CID-TC version, provisions should be made to limit engine power output upon failure of the water-alcohol injection unit in order to prevent severe engine knock. In the 280 CID-TC-A engine, very high ambient temperatures or deterioration of aftercooler effectiveness (due to fouling, for instance) could put the engine into a knock regime leading to engine destruction. Power limiting means based upon a maximum inlet air temperature may be necessary.

These additional devices, combined with other advanced engine systems, certainly decrease the turbocharged engine reliability and maintainability.

Safety

No significant changes in safety considerations are foreseen as a result of turbocharging.

Fuel

The present analysis is based upon using the same fuel as is now being used by conventional engines.

Engine Related Modifications

For the 250 CID-TC engine, no significant alterations in other vehicle systems are foreseen.

The 280 CID-TC-A engine will require a blower for the aftercooler. A method to accomplish this without adding electric motors and additional generator capacity has been used for diesel engine aftercoolers. A portion of the hot compressed air coming from the compressor is bled off and used to drive a very small air turbine. This turbine is coupled to a direct-drive blower that draws ambient air over the cooling surfaces in the aftercooler. A bleed rate of 10 to 20 percent is apparently adequate to accomplish this and makes it necessary to provide a slightly larger capacity turbocharger. This technique will doubtlessly require additional development work, but it appears to be a good approach.

Cost

For the 250 CID-TC engine, the incremental cost factors involved are:

- smaller engine
- higher loading on engine
- addition of turbocharger
- addition of water-alcohol injection

The reduction in material cost due to the engine displacement decrease is probably nullified by the necessity for higher quality exhaust valves and added structural strength to permit operation at generally higher bmep and consequently higher temperatures. It is estimated that the incremental cost increase of the 250 CID-TC engine over the 350 CID carbureted engine is $75 to $150.

For the 280 CID-TC-A engine, the incremental cost factors are:

- smaller engine
- higher engine loading
- addition of turbocharger
- addition of aftercooler
- addition of aftercooler blower

The aftercooler is expected to be a significant cost item. It is estimated that the incremental cost increase of the 280 CID-TC-A engine is $150 to $250.

Consumer Acceptance

Consumer acceptance will be detrimentally affected by increased acquisition cost, engine noise, possible acceleration deficiencies, and a more difficult maintainability. Improved fuel economy will partially offset this, as will possibly the "image" associated with turbocharged engines. On the balance, it is expected that consumer acceptance will be more difficult to obtain with the turbocharged engine than with the conventional carbureted engine.

Demonstration by 1976

It is feasible to demonstrate a vehicle with a turbocharged engine by 1976. No new technical problems must be solved. Emission control is expected to be the most difficult problem, but the solution will be similar in nature to those already used in the conventional automobile engines.

Time Required for Implementation

It is excepted that limited production of a turbocharged engine in a vehicle could be accomplished by the 1980 model year. Suppliers at present do not produce turbochargers adequate to meet the demand of an anticipated one million units, and only development impetus in this area would promote expansion of facilities.

11. VARIABLE DISPLACEMENT ENGINE

Concept

From previous discussions it will be clear that, at least from a fuel economy standpoint, it is desirable to use a small displacement engine at low power requirements in order to operate the engine in regimes of lower bsfc. To maintain vehicle performance, it is necessary to increase engine power by some extraordinary means at conditions of high power requirement. One can premise means to continuously vary engine displacement to achieve this result, but the mechanical complexity of such systems is forbidding.

In an attempt to evaluate a less complex and more familiar arrangement, the following engine was synthesized. Assume that a 350 CID, eight-cylinder engine is modified as follows:

- Every other cylinder in the firing order is provided with a modified valve train, and these four cylinders are given an intake manifold separate from the other four cylinders. The valve train is modified so that, upon an external signal, the valves are not actuated by the cam shaft and, consequently, remain closed. The various means used to accomplish this include a cam shaft that is slideable along its axis to disengage the followers from the cam surfaces; a spacer between cam and cam follower that can be mechanically removed; a hydraulic piston in the push rod that can be collapsed by removal of oil pressure to the piston; and so forth.

- Separate carburetors for each set of four cylinders.

- A control system that actuates carburetor thorttle plate in response to load and activates or deactivates the cylinders with the modified valve train.

In operation, the four unmodified cylinders would operate throughout the engine load spectrum. At low power requirements, the modified cylinders would be inoperative with both intake and exhaust valves closed, and the power requirements would be carried entirely by the four operating cylinders. As the imep of the operating cylinders approached their maximum, the four modified cylinders would be brought into operation and the engine would then operate in the conventional manner.

There are obvious questions and unresolved problems with this system involving increased torque fluctuations on four-cylinder operation including smoothness of changeover from four- to eight-cylinder operation and increased complexity. The consequences and relative severity of these will be considered later.

Fuel Economy Evaluation Procedure

To evaluate this concept, a fuel map was constructed. The baseline fuel map for an uncontrolled emission engine was used as the basis for the new map.

To begin with, it will be clear that if the friction of each cylinder is the same whether firing or not, and if the bhp of the engine firing on four cylinders is the same as the engine firing on eight cylinders, then the ihp of each of the four-firing cylinders is double that of the eight-firing cylinders. The mechanical efficiency of the two engines will be equal, and the only gains in bsfc must be due only to the lower percentage heat losses and more favorable fuel/air ratio resulting from operation at a higher ihp for the four cylinders.

However, for the case considered, the friction of the engine with four-firing cylinders is reduced; therefore, gains in bsfc come from an improvement in mechanical efficiency as well as decreased percentage heat losses and leaner fuel/air ratio.

The calculation procedure to obtain the new map was as follows: At a given piston speed an engine bmep was chosen from the baseline map. For the engine with four firing cylinders, bmep was defined as

$$bmep_{eng} = \frac{(bhp_{eng})(792{,}000)}{(N) \quad (D_{eng})}$$

where N is engine speed and D_{eng} is the displacement of all eight cylinders. Valve and pumping friction were calculated[29], and these factors were subtracted from the friction of the four inactive cylinders to calculate a new engine fmep. The mechanical friction of both the engine with four-firing cylinders and the original engine was then calculated. The imep of each of the four-firing cylinders was calculated and the ratio of the thermal efficiency of these cylinders compared to the eight-cylinder engine was computed using data from the baseline map. The bsfc of the four-firing-cylinder engine was then calculated, knowing the bsfc for eight-firing cylinders from the baseline fuel map:

$$\frac{(bsfc)_4}{(bsfc)_8} = \frac{(\eta_m)_8}{(\eta_m)_4} \frac{(\eta_t)_8}{(\eta_t)_4}$$

where η_m is mechanical efficiency and η_t is thermal efficiency.

When the required imep for each of the four-firing cylinders exceeded the original maximum imep of the engine, the engine was reverted to an eight-cylinder operation, and the bsfc of the original fuel map was retained unchanged. The resulting fuel map is shown in Figure 60. Below the line marked "max. bmep - 4 cylinders," the engine operates on four cylinders as already described. Above this line, the engine operates in a normal eight-cylinder fashion. The specific fuel consumption is considerably reduced in the four-cylinder operating mode, but most strongly at low speeds and bmep where the reduction of pumping losses have a large influence on mechanical efficiency.

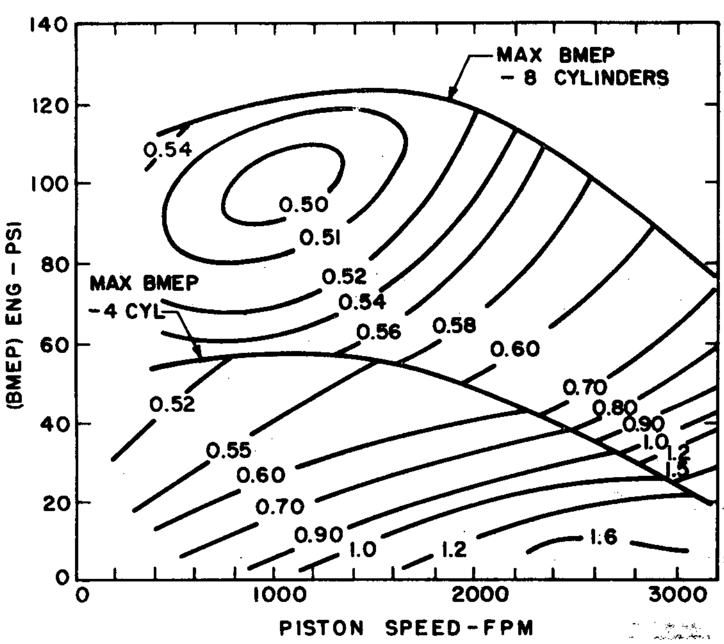


FIGURE 60. VARIABLE DISPLACEMENT ENGINE

Emissions

The basic change in engine operation involved in this scheme is operation of the four-firing

[29]Bishop, "Effect of Design Variables on Friction and Economy," SAE Paper 812A, Jan. 1964.

cylinders at higher imep than normal for the given speed load conditions. Qualitatively, in road operation this will tend to increase mass emission of hydrocarbons, especially near the maximum output of the four cylinders (compared to the conventional engine at the same bmep). CO will be increased but to a lesser degree, and NO_X will also be increased substantially. Some reduction in HC and CO would be expected at idle conditions because of higher cylinder imep and reduced residual fraction. Further, the transition between the four- and eight-cylinder operation requires that four cylinders be started. This repeated starting requirement will result in increased HC and CO. It has been determined that for the chosen engine displacement only one start-stop cycle of the four modified cylinders is required during the LA-4 emission cycle; nevertheless, this characteristic will add substantially to the exhaust emissions.

Above the maximum bmep for the four-cylinder operation, emissions will not differ from the conventional engine. Therefore, it is very nearly certain that emission control on the described concept will be more difficult than on the present conventional engine.

Evaluation

Fuel Economy

The fuel economy of the variable displacement engine installed in a 4300-lb car was computed from the fuel map of Figure 60. Compared to the baseline economy of a standard 350 CID engine, the variable displacement engine demonstrates a 23-percent improvement. This comparison is made based upon both engines having no emission controls. The desired comparison is the fuel economy of the variable displacement engine meeting 0.41–3.4–2.0 emission standards against the standard engine meeting 1973 emission standards. To estimate this, the following equation was used:

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

where

A – fuel economy of variable displacement engine, 0.4–3.4–2.0 emission standard

B – fuel economy of standard 350 CID engine, 1973 emission standard

C – fuel economy of variable displacement engine, no emission controls

D – fuel economy of standard 350 CID engine, no emission controls

The value of C/D is 1.23, as discussed above. D/B is estimated to be 1.09 based upon available data. The ratio C/A must be greater than D/B for several reasons:

- The 0.41–3.4–2.0 standard is more severe than the 1973 standard.

- The uncontrolled emissions of the variable displacement engine are greater than those of the conventional engine. The expected increase in NO_X is especially pertinent since control of this emittant will require increased EGR and a consequent reduction in fuel economy.

- These effects may require an overall increase in the piston displacement of the variable displacement engine in order to maintain equal maximum power, and this will further decrease economy.

Based on these considerations, it is estimated that the ratio C/A will be about 1.20. Therefore,

$$\frac{A}{B} = (1.23)\left(\frac{1}{1.20}\right)(1.09) = 1.12$$

The requirement for repeated starting and shutdown of four cylinders, as the vehicle accelerates and decelerates through the transition power, will require additional fuel and will also increase CO and HC emissions. The method chosen for the fuel economy calculation does not account for this effect, so an additional fuel economy penalty must be assigned. It is estimated that this penalty is about 5 percent of the factor C/D. Therefore, the final fuel economy comparison is:

$$\frac{A}{B} = (0.95)(1.12) = 1.07$$

Performance

The maximum power output of the variable displacement engine is the same as that of the conventional engine, so top vehicle speeds are equal. The problem of transition from the four-cylinder operation to an eight-cylinder operation is difficult, since, at the transition, the valve gear of the four-modified cylinders must be activated, the throttle opened, and the throttle for the four-unmodified cylinders adjusted to a new position so that the overall engine power is held constant. It seems inevitable that the transition will involve a power discontinuity that may unfavorably affect acceleration performance.

Emissions

As already discussed, exhaust emissions of the variable displacement engine will be greater than those of the conventional engine due to:

- Higher average imep of four of the engine cylinders.
- Transition from four to eight cylinders, requiring additional startups.

Noise and Vibration

The exhaust noise of the variable displacement engine will be slightly higher than that of the conventional engine, due to the higher cylinder pressures at low loads. Vehicle vibration will also be increased, although it is not expected that this problem will be especially troublesome, except at idle.

Weight and Size

Inlet manifold design changes, the addition of a carburetor, and the valve control mechanisms will have insignificant effects on size and weight.

Reliability and Maintainability

The addition of a complex valve gear and transition controls will degrade the overall engine reliability and make adequate maintenance more difficult.

Other Modifications to Engine and Vehicle

The total coolant load of the engine should not be substantially affected by the variable displacement engine. However, the distribution of the engine heat will be substantially altered in that four of the cylinders will contribute more heat to the coolant (on a time-averaged basis) than the other four. Modifications to the cooling passages of the four more active cylinders may be necessary.

For the same reason, valve life of the four active cylinders may suffer; therefore, heavy duty valves may be required.

The problem of transition controls has already been briefly discussed and is a major difficulty.

Cost

Factors that influence the incremental acquisition cost of the variable displacement engine as compared to the conventional engine are:

- Valve gear "deactivators" for four cylinders;
- Additional carburetor;
- Divided intake manifold;
- Carburetor and valve controls to provide transition between four- and eight-cylinder operation; and
- Possibly higher quality exhaust valves for four cylinders.

It is estimated that the cost of these items will increase the acquisition cost of the vehicle from $125 to $175.

Consumer Acceptance

Factors that may change the degree of consumer acceptance from that now enjoyed by the conventional engine-powered vehicle are:

- Slightly more noise,
- Engine vibration at idle,
- Power discontinuity at transition between four and eight cylinders,
- Slightly reduced reliability and maintainability,
- Added acquisition cost, and
- Improved fuel economy.

On balance, it is expected that consumer acceptance would be significantly less for this engine than for the conventional engine.

Demonstration by 1976

Vehicles with the variable displacement engine could be demonstrated by 1976.

Time Required for Implementation

It is believed that the variable displacement engine can be produced in limited quantities for the 1980 model year.

12. ENGINE WITH REDUCED FRICTION

General

Methods to reduce engine friction have been discussed in Section 7 of this report. To summarize these, the following changes have been found effective in reducing the friction of engines:

- Light reciprocating parts,
- Large clearances between piston and cylinder,
- Short pistons,
- Reduced number of piston rings, and
- Smaller diameter journal bearings or increased bushing-to-journal clearance.

As was earlier discussed, some of these changes result in increased engine noise. The removal of an additional ring from, say, a three-ring piston will increase blowby by 10 to 15 percent[30]. This will affect PCV valve design and may present other difficulties such as oil carryover requiring further engine detail design changes. Oil degradation rates will increase, and, in some cases, piston oil control may be made more difficult. It must be emphasized that the engine modifications listed above will have repercussions on engine reliability, noise, life, oil consumption, oil life, acquisition cost, and operating cost. Insufficient data are available to us to evaluate these factors precisely.

Nevertheless, production engines have been built in the past incorporating all or most of the listed modifications. These engines were the reciprocating aircraft engines, and some of them demonstrated remarkably low friction values.[21] Figure 61 shows measured friction mep for both automobile engines and a V-12 aircraft engine. The difference in fmep between the two types of engines is striking.

One further method for reducing friction in a conventional engine is by the modification of the lubricant. References 31 and 32 present test results on automotive engines showing a friction reduction through the use of about 1 percent by weight of molybdenum disulphide in the crankcase oil. These tests indicate that, with no other charges, the fmep can be reduced by approximately 5 percent.

It is believed that a reduction in engine friction is possible with some sacrifices in increased engine noise, increased cost, and perhaps reduced engine reliability. Quantitative data are insufficient to evaluate these factors accurately. In view of this, the best approach is to take what seems to be a feasible friction reduction based upon the information provided above and to compute the improvement in fuel economy that would be realized by this change in friction.

If fmep is reduced to maintain the same bmep, the throttle opening must be reduced. The inlet air density is, thereby, reduced and throttling losses increase which tend to raise the fmep. This effect

[30] K.R. Kamman, et al., "Two-Ring Piston Development," SAE Paper 690750, 1969.
[31] Ethyl Corporation, "The Effect of Molybdenum Disulphide in the Crankcase Oil on Engine Performance," Report No. RS-222, Feb. 1963.
[32] Climax Molybdenum Co. of Michigan, "Motor Oil Tests with Climax MoS_2 Suspensions," Report No. RP-29-69-2, Feb. 1971.

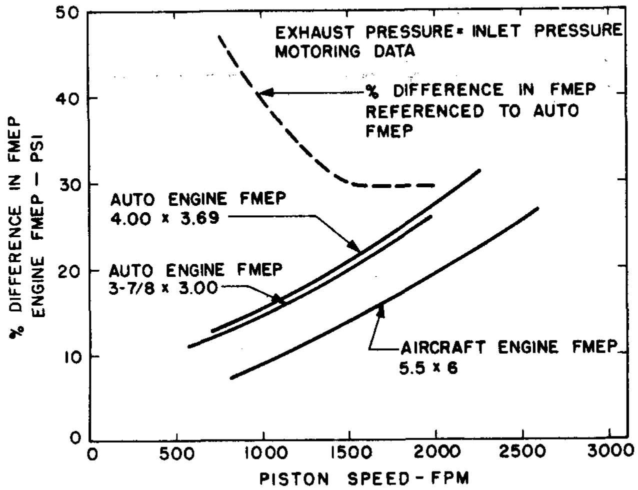


FIGURE 61. MOTORING FRICTION OF CARBURETED ENGINES

is most pronounced at low loads so that the measured percentage reduction in friction at full throttle becomes less as the throttle is closed and the load is reduced. When variations in the fuel/air ratio due to changes in throttle position are considered (combined with the resulting change in thermal efficiency), the picture becomes somewhat involved. For purposes of estimating a feasible friction reduction, it is assumed that the fmep can be reduced by 20 percent at full load, reducing to a 10 percent reduction at zero load by the schedule shown in Figure 62.

Fuel Economy Analysis Procedure

For the purposes of this analysis, it is assumed that the bsfc is a function only of the inverse of the mechanical efficiency. Using the percent reduction in friction shown in Figure 62, the reference fuel map for the 350 CID carbureted engine without emission controls was modified to account for the reduced friction. The resulting map is shown in Figure 63.

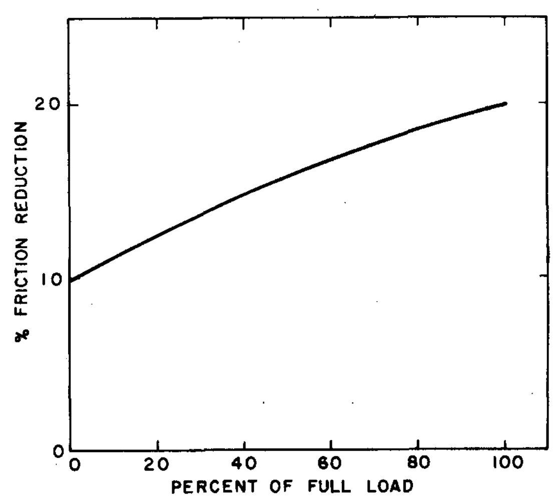


FIGURE 62. ESTIMATED POSSIBLE FRICTION REDUCTION

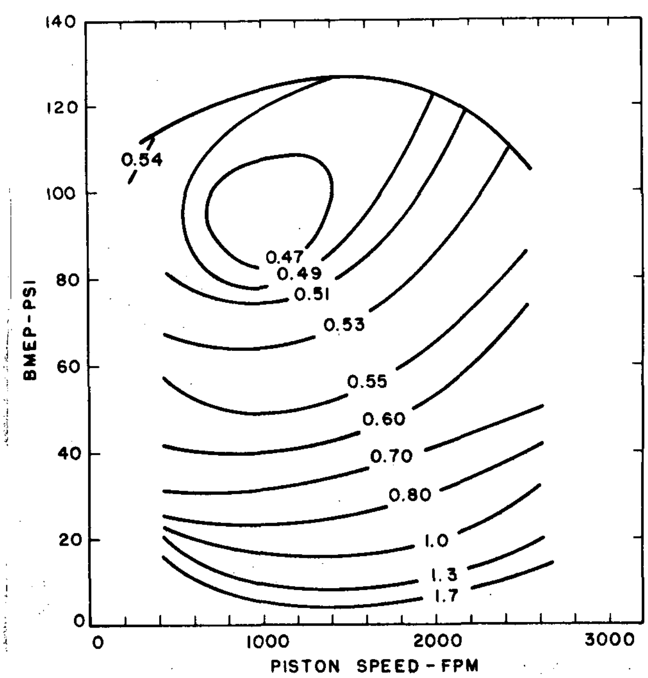


FIGURE 63. ENGINE WITH REDUCED FRICTION 350 CID

The fuel economy improvement was calculated using the standard procedure, and the result was a 0-percent improvement in economy. This is a comparison with both engines (reference and low friction) uncontrolled for emissions. The desired comparison is obtained as follows:

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

where

A – fuel economy of a low-friction engine meeting 0.41 to 3.4 to 2.0 emission standards

B – fuel economy of conventional engine meeting 1973 emission standards

C – fuel economy of a low-friction engine, uncontrolled emissions

D – fuel economy of conventional engine, uncontrolled emissions

From the previous equation, C/D = 1.0. The reduction in friction probably has a relatively minor effect on the methods used to meet the 0.41 to 3.4 to 2.0 standards. It has been estimated (see Section 8) that the effect on fuel economy of meeting these emission standards is a 15-percent reduction; therefore, A/C is 1/1.15. D/B has been estimated to be 1.09. Therefore,

$$\frac{A}{B} = (1.0)\left(\frac{1}{1.15}\right)(1.09) = 0.95$$

This disappointing result indicates that

- The driving cycle used for these calculations of fuel economy causes the engine to operate under conditions where throttling losses are a considerable part of total engine friction, and reductions in mechancal friction has a relatively small effect.
- Friction reduction, considerably greater than those assumed, will be necessary to significantly affect fuel economy; however, greater friction reductions require disproportionately greater sacrifices in noise level, blowby, cost, and reliability.
- Friction reduction in smaller engines running at a higher bmep and speed than the 350 CID engine analyzed herein will be more rewarding.

It is concluded that these results do not justify further consideration of friction reduction methods under the restraint criteria of this effort.

13. LEAN MIXTURE ENGINES

General

"Lean mixture engines" will be taken in this report to include all spark-ignited, carbureted engines that operate on fuel-air mixtures leaner than those normally used in carbureted engines. A great many methods have been proposed to obtain lean mixture operation including high energy ignition sources, multiple spark-ignition sources, turbulence promoters in the combustion chamber, heated intake air, fuel vaporizing or dispersing devices, and so forth. The diversity of these methods, combined with conflicting claims for improved fuel economy and a paucity of data, has led us to the following rationale for analysis.

At each engine speed and load there exists a best economy fuel-air ratio. Mixtures leaner than this best economy mixture cause a reduction in power output, and the corresponding decrease in mechanical efficiency causes an increase in bsfc. Therefore, if it is assumed that an engine can always be operated at the best economy fuel-air ratio, and if a fuel map for this type of operation is prepared, the fuel economy obtained from the map will represent an estimate of the best economy possible from the use of lean mixtures in a carbureted engine.

Fuel Economy Analysis

Consider two engines, one with normal pre-emission carburetion and one with carburetion arranged to provide operation at the best economy fuel/air ratio. For these two engines running at the same speed and throttle setting:

$$\frac{\text{bmep}_2}{\text{bmep}_1} = \frac{(F_1 \eta_{t_1} / F_2 \eta_{t_2}) - (\text{fmep}/\text{imep}_1)}{1 - \text{fmep}/\text{imep}_1}$$

where

F_1 – Fuel-air ratio, normal carburetion

F_2 – Fuel-air ratio for best economy

$$\frac{\text{bsfc}_2}{\text{bsfc}_1} = \left(\frac{F_2}{F_1}\right) \left(\frac{\text{bmep}_1}{\text{bmep}_2}\right)$$

Since the throttle setting and engine speed are the same for the two engines, $\text{fmep}_1 = \text{fmep}_2 = \text{fmep}$. From a fuel map of the conventional engine, values of bmep_1 and bsfc_1 can be taken (at a given speed). The fuel/air ratio for a conventional pre-emission engine is shown in Figure 64[33]. A characteristic friction curve was chosen for the engine.[34] Figure 65[21] shows the product $F\eta_t$ as a function of the fuel/air ratio. From these data, F_1, $F_1 \eta_{t_1}$ and fmep were taken, and the imep_1 was calculated from the sum of bmep_1 and fmep_1. Then a number of values for F_2 were chosen, and the bmep_2 and bsfc_2 were calculated for each value of F_2. A typical result is shown in Figure 66. Note that the minimum values of bsfc differ little from the normally used values, and that the normal and best economy fuel/air ratio values differ by the greatest margin at the higher loads. The resulting fuel map of

[33]Fawkes, et al, "The Mixture Requirements of an Internal Combustion Engine at Various Speeds and Loads" Thesis, MIT, 1941.
[34]Edson and Taylor, "The Limits of Engine Performance," Vol. 7, SAE Special Publication *Progress in Technology* (TP7), 1964.

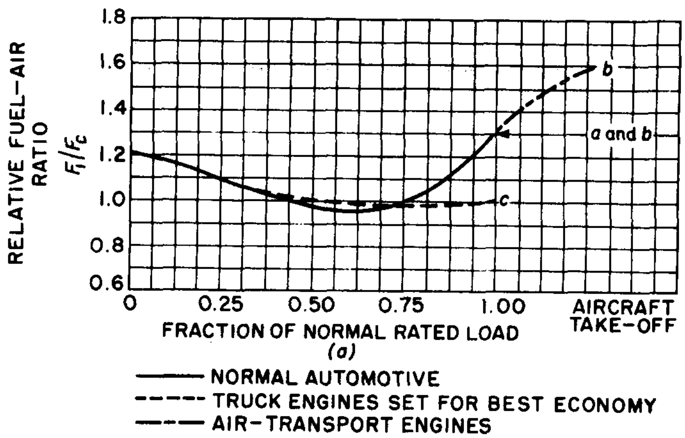


FIGURE 64. NORMAL FUEL/AIR RATIOS FOR AUTOMOBILE ENGINES

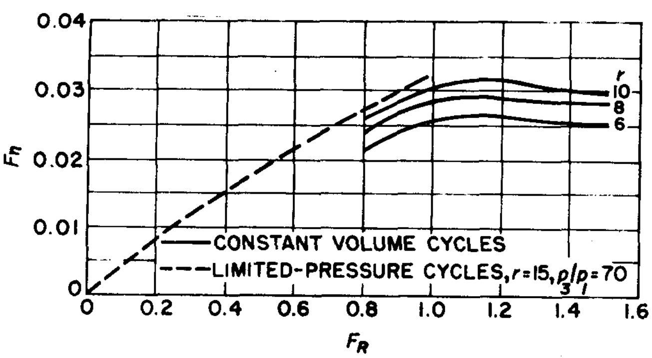


FIGURE 65. PRODUCT $F\eta$ FOR FUEL/AIR CYCLES

Figure 67 shows the result of the complete analysis. Each point on the map represents engine operation at the best-economy fuel/air ratio, except near maximum bmep where the relative fuel/air ratio was increased to 1.2 to obtain the rated power output of the engine.

Note that this method takes into account the reduction in pumping losses due to wider throttle openings at leaner mixtures. Also, no consideration is given to the actual physical problem of igniting lean mixtures; it is assumed that the best economy mixture can be ignited. Actually, only in very few operating regimes is the best economy mixture low enough so that any difficulty would be expected using normal spark ignition.

It has been shown experimentally[35] that the method described above leads to a somewhat less-than-optimum engine performance, and that the method where fuel rate is held constant and engine air flow (or throttle position) is allowed to vary at constant speed leads to a leaner, more efficient fuel/air ratio. However, the latter method is not amenable to calculation, since the minimum bsfc is set by the lean limit of ignition, and this lean limit is very dependent upon engine operating conditions and is difficult to predict with any accuracy. Further, experiments show that the difference in optimum bsfc determined by the two methods is small and probably well within the accuracy of our calculations. Therefore, the less accurate, but calculable technique is used here to approximate the more accurate technique.

Evaluation

Using the fuel economy calculation procedure standard in this report, the 350 CID lean-mixture engine shows an 8-percent increase in fuel economy over the 350 CID conventionally carbureted engine. This comparison is based upon both engines being uncontrolled for exhaust emissions. To correct for this factor, the following equation is used:

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

[35] Schweitzer, P.H., "Control of Exhaust Pollution Through a Mixture-Optimizer," SAE Paper No. 720254, Jan. 1972.

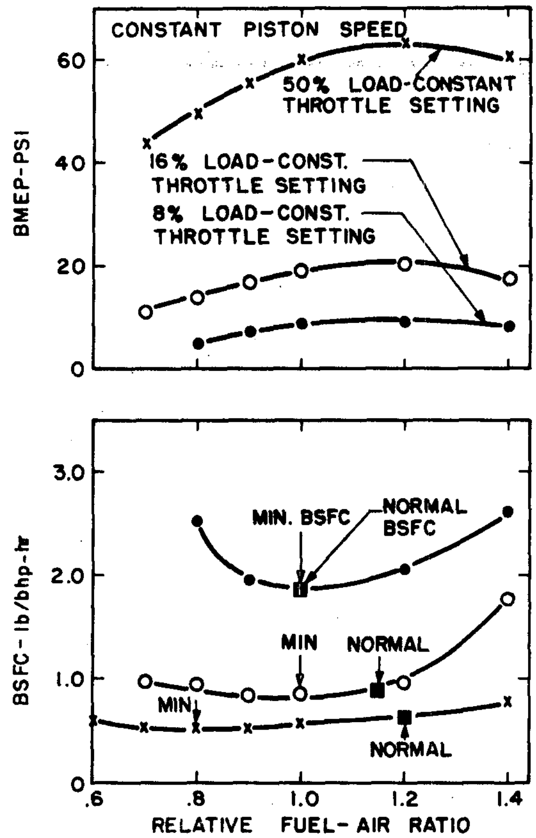


FIGURE 66. EFFECT OF FUEL-AIR RATIO VARIATION ON CARBURETED ENGINE PERFORMANCE

where

A – fuel economy of a lean-mixture engine meeting 0.41 to 3.4 to 2.0 emission standards

B – fuel economy of a conventional engine meeting 1973 emission standards

C – fuel economy of a lean-mixture engine without emission controls

D – fuel economy of a conventional engine without emission controls.

The previously discussed results show that C/D = 1.08, and the ratio D/B has been estimated as 1.09. The comparison of the economy of the lean-mixture engine with and without emission controls requires consideration of the method used to achieve lean mixture operation.

The Honda CVCC engine (a lean-mixture engine within our definition) apparently suffers only small losses in economy in achieving acceptable emissions, due to its divided combustion chamber and slow burning of the lean mixture. It is estimated that to reach the required standard of 0.41–3.4–2.0, the factor A/C is about 0.95. For this engine, then, A/B = (1.08) (0.95) (1.09) = 1.12 or approximately a 12-percent improvement in fuel economy *if* the overall fuel/air ratio of the Honda CVCC is held closely to the best economy value and other losses are about the same as a conventional engine. In actual practice, it would be expected that the actual gain would be less than this value, since always holding closely to the best economy fuel/air ratio is probably not attained in practice.

The Honda CVCC appears to be unique in its ability to burn lean mixtures without excessive NO_X formation. As is discussed elsewhere in this report, in an engine with a homogeneous mixture of fuel and air, maximum NO_X production occurs at a fuel/air ratio somewhat leaner than stoichiometric. Since normal engine operation is for the most part at mixtures richer than this, leaning the mixture will tend to raise NO_X emissions. In the absence of other NO_X control mechanisms, higher NO_X will require greater rates of EGR, and the benefits of leaner mixtures may be lost altogether. For homogeneous mixture engines (such as those relying on higher spark energy, turbulence, multiple spark, or extraordinary fuel vaporization methods to operate at lean mixtures), the ratio A/C may vary between 0.85 and 0.90. For this case, A/B is in the range 1.00 to 1.06. When it is considered that this range for A/B is optimistic, in the sense that it was assumed that *all* engine operation is at a best economy fuel/air ratio, it is probable that no benefit in fuel economy will be derived by these techniques.

In order to further confirm these results, a detailed analysis was done for a particular lean-mixture engine concept. The results of this analysis are presented in the next section.

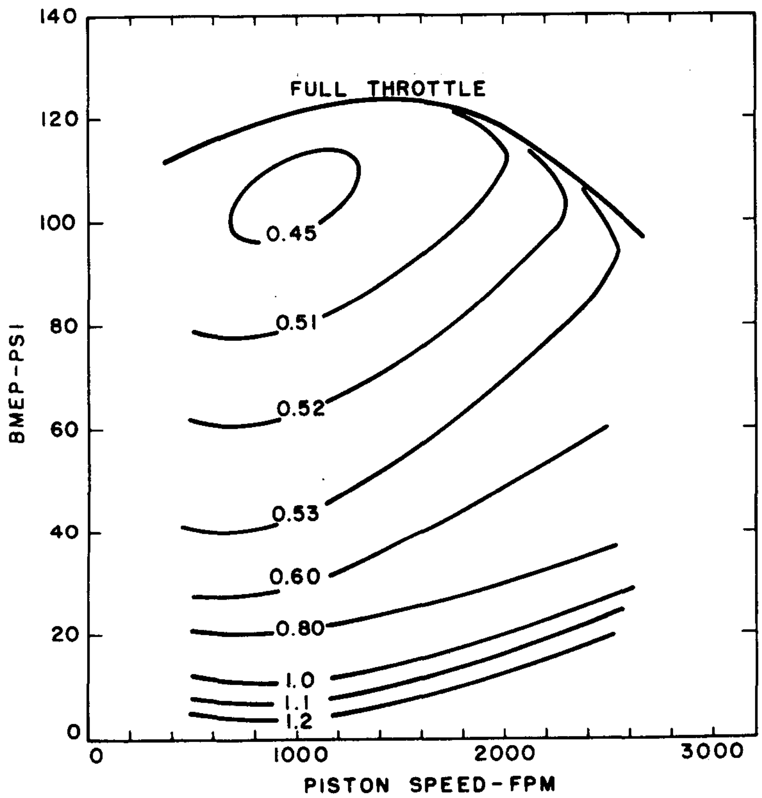


FIGURE 67. LEAN-MIXTURE ENGINE 350 CID

Intake Valve Throttling

The concept of intake valve throttling has been used to allow stable operation of a *carbureted, spark ignition* engine at extremely lean air/fuel ratios. The basic feature of this design is the relocation of the throttling mechanism from the carburetor to the cylinder intake valve. The air supply to the engine is throttled, but the control is achieved by an adjustable valve at each cylinder rather than by a single valve which serves several cylinders.

The basic alteration required is the installation of a mechanism to vary the lift of the intake valve while the engine is operating. The adjustable lift controls the size of the intake aperture, and with proper adjustment the combustion process can be affected. For any circumstance involving flow through an orifice, there exists a minimum orifice size for which the velocity of the fluid at the orifice is equal to the local *velocity of sound.* This *critical* orifice size limits the flow through the orifice; the flow rate cannot be changed if the upstream pressure is fixed. If the intake orifice is adjusted to critical size for a given engine condition, then the sonic flow at the valve subjects the fluid to very large shear forces. Application of these forces results in atomization of the liquid fuel and intense, small scale turbulence in the intake charge. The turbulence promotes mixing, and extremely rapid burning occurs when the charge is ignited. Thorough mixing and rapid combustion allow successful use of lean mixtures.

The intake valve lift must be adjusted to produce sonic velocity in the intake orifice for each engine air requirement (load). Fuel may be supplied either as a premixed charge, as from a carburetor, or by an intake port fuel injection system.

It should be noted that there are salient differences between intake valve throttled engines and stratified charge engines. With intake valve throttling, the extreme flow conditions at the intake orifice promote charge homogeniety and small scale turbulence. In the stratified charge engine, large scale turbulence, or swirl, is sought, and the combustion process takes advantage of the fact that the fuel-air charge is highly inhomogeneous. Instead of representing a difference in principle, therefore, the intake valve throttled engine uses the variable lift valve as an accessory to approach maximum combustion efficiency with a conventional, premixed charge engine.

Intake valve throttled engines have been tested in single cylinder, multicylinder and vehicle mounted configurations[36]. During the multicylinder and vehicle testing reported in the reference cited, intake port fuel injection was used. Satisfactory operation was achieved at extremely lean fuel/air ratios without vehicle surge, but fuel economy was found to be a strong function of spark advance (Fig. 68).

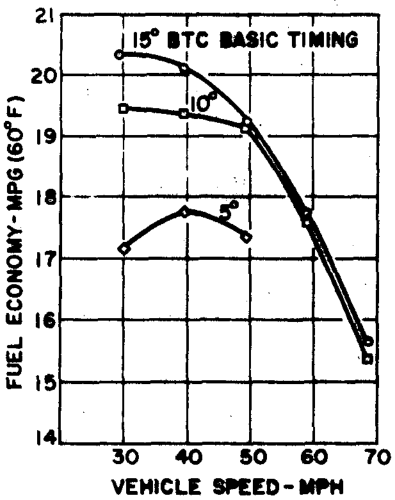


FIGURE 68. FUEL ECONOMY AS A FUNCTION OF SPARK ADVANCE

Evaluation—Intake Valve Throttling

Identification of Improvement

Intake valve throttling has been recommended as a means of conserving fuel. The available data are primarily contained in Reference[36], which is a report of tests conducted with a single cylinder engine, a multicylinder engine, and an engine installed in a vehicle. Although the reported information indicates that the intake valve throttling mechanism may allow lean mixtures to be burned, the fuel economy benefits remain to be confirmed.

Fuel Consumption Reduction

The fuel consumption data, shown in Figure 68, indicate a sensitivity to ignition timing but do not provide a comparison to a reference vehicle. A complete engine map was not provided, although partial maps in the form of curves of brake mean effective pressure versus brake specific fuel consumption were presented as shown in Figures 69 and 70. It can be observed from these curves that the conventional and intake valve throttled engines exhibit virtually identical fuel consumption at values of brake mean effective pressure in excess of 20 16/sq inch. Since these curves are cited as typical of engine performance at other speeds, it seems that a significant economy advantage is not obtained. Figures 69 and 70 are composed of a number of "fishook" curves, i.e., curves of bsfc versus bmep at constant throttle opening with varying fuel/air ratio. It can be seen from

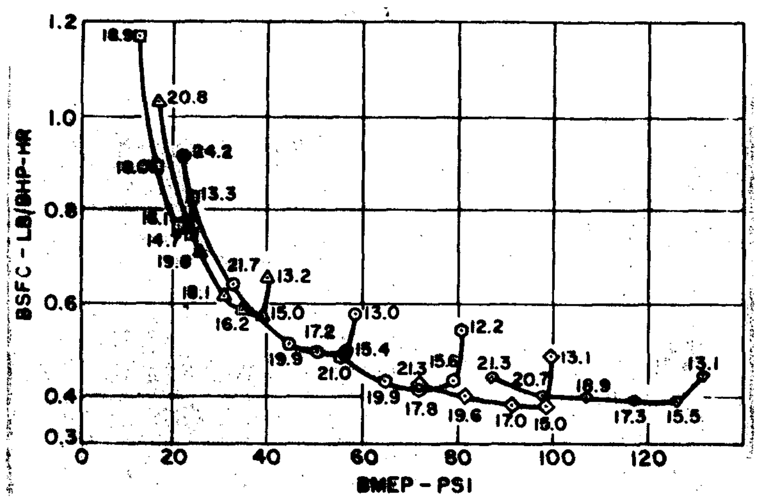


FIGURE 69. SPECIFIC FUEL CONSUMPTION WITH CONVENTIONAL THROTTLING 1200 RPM, INDOLENE 30 FUEL

[36]Stivender, Donald L., "Intake Valve Throttling (IVT)—A Sonic Throttling Intake Valve Engine", SAE Paper 680399, 1968.

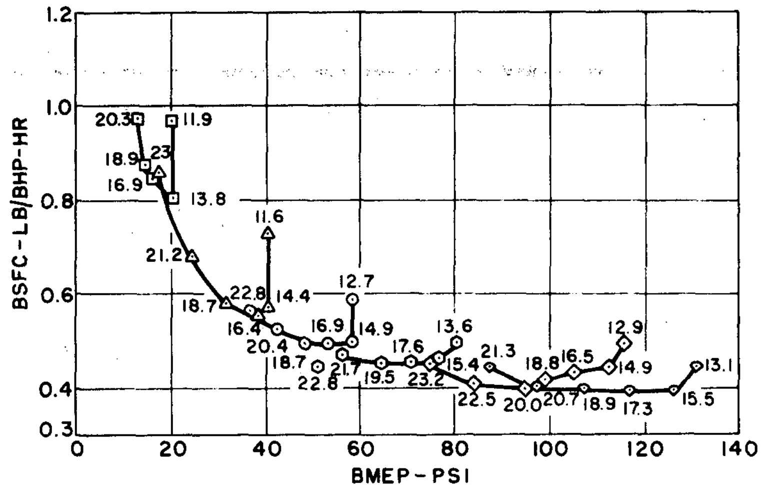


FIGURE 70. SPECIFIC FUEL CONSUMPTION WITH VALVE THROTTLING 1200 RPM, INDOLENE 30 FUEL

comparison of the two figures that the best fuel/air ratios at the various loads are much the same for the two engines.

It should be noted that this system is particularly amenable to fuel shutoff during deceleration. If compatibility with the emission control systems can be assured, then the valve actuating mechanisms can be set for zero lift when deceleration conditions prevail. However, this capability is also available with the intake port fuel injection system which would almost certainly be required on an intake valve throttled engine.

It is possible that more comprehensive data on the intake valve throttling process exist in the files of the automobile manufacturers. If this is the case, and a complete engine map can be obtained, then a numberical evaluation of the concept using the established procedure would be possible. The more complete body of data, with evaluation, might allow a more favorable impression of the fuel economy benefits associated with intake valve throttling.

State of Development

The procedure has been demonstrated with single cylinder and multicylinder laboratory engines, and some road testing has been performed. Routine development work would be required prior to a production commitment.

Demonstration

The required engine modifications have been operated on at least one single cylinder test engine, one multicylinder test engine, and one vehicle mounted engine.

Reliability

The intake valve throttling process requires that a variable adjustment capability be added to moving parts of the engine. This greatly increases the complexity of the valve system, and some reliability problems are likely. In addition, an intake port fuel injection system would probably be used; such systems are somewhat more complex than the carburetors that they replace.

Cost

Excluding the cost of the intake port fuel injection system, which is analyzed as an entity elsewhere in this report, the cost of the intake valve throttling accessories is an estimated $25 to the consumer. It should be noted, however, that although the proposed engine is throttled, no manifold vacuum is produced. Engine load signals to the ignition system and the transmission must be provided by another means, and a separate pump would be required for vacuum operated accessories such as power brakes.

Safety

Since the engine is basically throttled, compression braking should not be affected. There should be no conflict between the proposed intake valve throttled design and the 1973 Safety Standards.

Emissions

Although some emissions measuring equipment was connected to the test engine described in Reference 36, no specific values were reported. It was observed that emissions were similar to those from conventional engines in both character and quantity, except that carbon monoxide concentrations could be quite low if conditions were favorable. However, an engine operating at very lean mixtures, without misfire, should exhibit low values for all contaminants. Comprehensive exhaust emission tests will be necessary before definite conclusions can be reached.

Noise

The additional mechanical components required for the intake valve throttled engine may contribute to the engine noise level, but the effect is probably not significant. In addition, there may be noise associated with the flow through critical nozzles at the valves. This is typically an extremely noisy situation, and some noise control effort may be necessary.

Performance

Performance should not be affected by the installation of the intake valve throttling mechanisms; at or near wide open throttle the engine would return to the conventional operating condition.

Time Required for Implementation

Since the technique has been demonstrated, the additional testing and development should be possible prior to the production commitment deadline for the 1980 model year.

Consumer Acceptance

Since the proposed alterations simply relocate the point at which inlet air is throttled, the average consumer should not detect any difference in engine operation.

14. INTAKE PORT FUEL INJECTION

Throughout this document, reference has been made primarily to carbureted gasoline engines of the type supplied with the reference vehicles. In the case of diesel and stratified charge engines, to be discussed later in this report, fuel is injected directly into the cylinder under high pressure. Another fuel supply system, commonly referred to as port injection, utilizes a low pressure pump which supplies fuel to nozzles located in the intake manifold in the vicinity of each intake valve. Generally, the air supply to the engine is throttled in a manner similar to that for a carbureted engine; in fact, some systems employ standard carburetors as air metering devices. A signal from the air metering device is used to regulate the fuel supply.

Several systems using low pressure port injection have been marketed on production vehicles. The early Rochester system, used on some Corvette automobiles during the 1960's, allowed a continuous flow of fuel through the nozzles.[37] Later systems, such as the Bendix-Robert Bosch system used on Volkswagen automobiles, are described as "timed" systems; but individual cylinders do not receive an appropriately timed fuel quantity. The "timing" with these systems consists of a timed fuel flow to each cylinder bank; systems planned for the future can provide for timed fuel flow to each cylinder.

Another feature of future systems is overall control of the air/fuel ratio using engine emissions as an input to the system in addition to the traditional speed and load signals. The systems employ a three-way catalyst in the exhaust system; such a catalyst is capable of reducing the quantities of unburned hydrocarbons, carbon monoxide, and oxides of nitrogen as long as the engine air/fuel ratio is maintained within narrow limits. An oxygen sensor located in the exhaust system supplies a signal to the fuel injector for air/fuel ratio control; it is claimed that adequate control is virtually impossible with production carburetors.[38]

In general, intake port fuel injection systems appear to offer several advantages by comparison with conventional carburetors. For a carbureted engine, a considerable quantity of liquid fuel must be maintained on the walls of the intake manifold as a result of the vapor-liquid equilibrium characteristics of the fuel. This liquid fuel coating on the manifold walls causes a lag in throttle response on acceleration and increased emissions of hydrocarbons and carbon monoxide during deceleration. If the amount of liquid fuel on the manifold walls can be reduced, as it would be with port injection, then driveability should be improved and emissions of hydrocarbons and carbon monoxide should be reduced. Furthermore, faster warmup of the engine should be achieved because fuel is supplied directly to the hot intake valve and vaporization capability is increased.

Evaluation—Intake Port Fuel Injection

There is some evidence that intake port fuel injection can be used to an advantage on production vehicles. Throughout this evaluation, the system considered will be of the electronic control type, with fuel individually metered and timed to each cylinder.

Reduction in Fuel Consumption

The primary advantage that intake port fuel injection can offer in the area of fuel economy is precise control of the air/fuel mixture. A carburetor, once it has been assembled and calibrated, is

[37]Dolza, J., E. Kehoe, D. Stoltman, and Z. Duntov, "*The GM Fuel Injection System,*" SAE Transactions, V. 65, p. 739, 1957.
[38]Rivard, J.G., "Closed Loop Electronic Fuel Injection Control of the Internal Combustion Engine," SAE Paper 730005, 1973.

not amenable to mixture alteration except for the idle mixture, and the use of a float bowl prevents rapid response to changing engine requirements. Some aircraft carburetors incorporate a capability for mixture change, but the control system is not generally sensitive to engine speed and load. The timed fuel injection system supplies appropriate quantities of fuel to each cylinder, and the electronic control unit can be adjusted to modify the air/fuel ratio. The system could be regarded as a pressurized carburetor with outlets at each cylinder and a sophisticated control system. Carbureted engines traditionally exhibit difficulty with uniform fuel distribution to all cylinders; this problem is diminished with fuel injection because the fuel is supplied to each cylinder rather than to a central location. Another feature of the fuel injection system which can be used advantageously is the possibility of fuel shutoff during deceleration. There is some indication that excessive quantities of fuel would be required to restart the engine, and possible detrimental effects on emission control equipment will be treated later, but the possible gain of about 5 percent in fuel economy provides sufficient motivation for further study.

Claims for reduced fuel consumption through the use of timed fuel injection may be found in the literature, but none may be regarded as completely definitive[38,39]. A thorough test program using standardized procedures will be required before conclusions can be drawn.

Fuel Consumption—Urban Cycle and Road Load

The fuel economy calculation procedure outlined elsewhere in this study was performed using the engine map presented in Reference 39 for a 283 cu in. engine equipped with fuel injection (see Figure 71). The calculation procedure resulted in an estimated 6-percent improvement in average fuel economy (urban plus road load) by comparison with the reference vehicle. This result should not be regarded as conclusive, since there were weight and displacement differences between the two vehicles. An engine map for the carbureted version of the vehicle was also presented in Reference 39, but it is not clear that the carburetor and the engine were well matched even though the configuration was stock. There is no accessible direct comparison between fuel injection and carburetion for the same modern vehicle.

Calculations were also performed for the reference vehicle based on the assumption that the fuel flow to the engine could be stopped entirely during deceleration. An electronic engine speed restart sensor would also be required. The result was an increase of 5 percent in fuel economy for the reference vehicle for the combined urban and road load calculation procedure.

Stage of Development

Electronic fuel injection systems have been under development for several years, and numerous European production vehicles have been equipped with modified versions of the system considered here. The Cosworth-Vega, to be introduced in limited production in 1974, will be equipped with the two-group injection.

Some development will be required on the feedback control, three-way catalyst system. Although such systems have been demonstrated, the oxygen sensors are notoriously short lived, and considerable effort is being devoted to development in this area.

[39] Freeman, J.H., Jr. and R.C. Stahman, "Vehicle Performance and Exhaust Emission, Carburetion Versus Timed Fuel Injection," SAE Paper 650863, 1965.

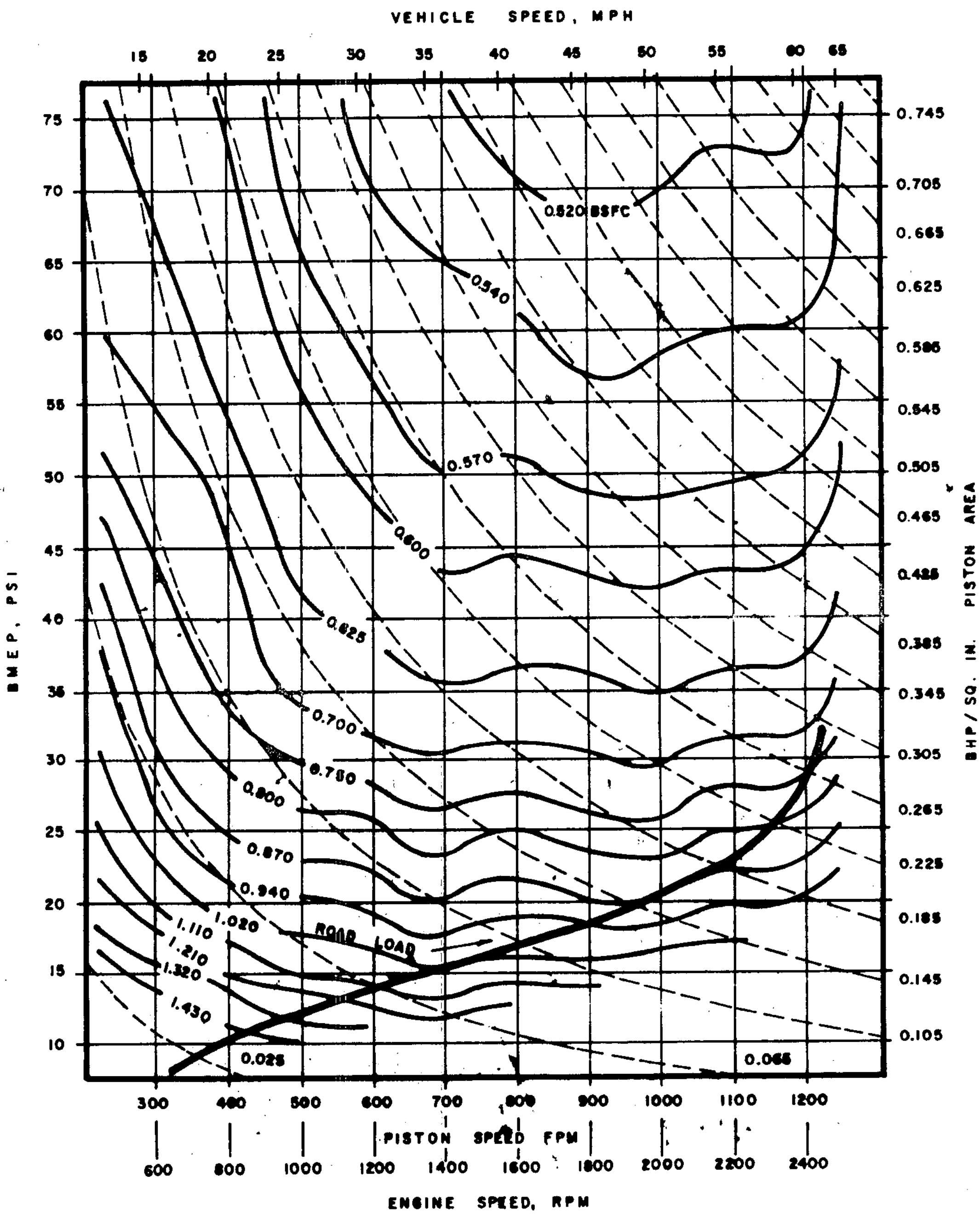


FIGURE 71. PARTIAL PERFORMANCE MAP OF VEHICLE C EQUIPPED WITH COMPLETE GASOLINE INJECTION SYSTEM (Automatic Metering Control Provides an Essentially Constant Air/Fuel Ratio of 14.5:1 at All Speeds and Loads.)

Demonstration by 1976

Electronic timed injection systems have been demonstrated. The remaining development problems will, in all probability, succumb within sufficient time to allow demonstration by 1976 of a system suitable for the useful life of a vehicle.

Reliability

The reliability of the timed fuel injection systems should be no worse than that of a conventional carburetor once the oxygen sensor durability problems are solved. In the area of pumps and injection nozzles, a wealth of diesel engine information is available and fuel handling and filtering procedures will be quite similar to those for diesel service. The control systems and sensors are well within the state-of-the-art, and their durability has been demonstrated on production vehicles.

It should also be noted that the electronic control systems are much more amenable to diagnosis by sophisticated electronic procedures than are carburetors. This factor is significant in the light of current trends toward automatic engine diagnosis.

Cost

The cost to the consumer of a basic timed injection system, without catalysts or oxygen sensor and with the two-group injection, has been estimated at $75 for the relatively small production quantities treated in this study. This cost is expected to drop with high volume production.

Any discussion of cost must include the effect of elimination of the carburetor. Carburetors, and the additional devices that accompany them, are becoming increasingly complex; the assumption that fuel injection systems are too expensive must be reexamined in the light of contemporary systems.

Safety

There should be no safety problems with the fuel injection systems that do not exist with carburetors. Although the fuel in the system is subjected to moderate pressure ($\sim$ three atmospheres), the injection nozzles provide a positive shutoff. Control of carburetor evaporative emissions is unnecessary.

Emissions

There appears to be general agreement that fuel injection systems allow better emission control, primarily because of enhanced ability to control air/fuel ratio. Also, since a warmup condition can be approached more quickly, the emissions following a cold start can be reduced. Prototype systems using a three-way catalyst with feedback (oxygen sensor) control, ignition timing control, and exhaust gas recirculation have met the most stringent Federal Standards with the exception of the requirement for 50,000 mile durability[38]. While this prototype system is presently a high risk item, and less complex systems are specified for the designs synthesized in this report, the example does serve to illustrate the potential of fuel injection systems. Other tests have also indicated that emissions can be controlled more readily with fuel injection; one of the main reasons cited in the uniformity of fuel distribution.

The use of fuel shutoff during deceleration was mentioned previously as a possible economy measure. In emission controlled engines, the possibility exists that reactors could cool during periods of fuel shutoff; a warmup period would be required after the vehicle returned to an acceleration or cruise mode. Emissions in this circumstance could conceivably be increased by a substantial amount, and all the fuel economy benefits of deceleration shutoff would not be available.

Noise

The noise levels of fuel injected engines should not be significantly different from those of carbureted engines.

Performance

The fuel injection system, because of the possibility of precise air/fuel ratio control, offers the possibility of good economy with minimum performance degradation. Performance of a vehicle with a fuel injected engine should be better than that of an equivalent vehicle with a carbureted engine for the same economy and emission levels.

Time Required for Implementation

Since prototype systems of the closed loop, feedback control type have been demonstrated, and since systems without feedback control are now in production, there should be no difficulty in providing fuel injection systems for 10 percent of the 1980 production volume. Accelerated implementation should be possible, particularly with the experience to be gained from the Cosworth-Vega production.

Consumer Acceptance

Fuel injection systems have been in demand for years from the "enthusiast" segment of the driving public; acceptance by this group should be immediate. Most drivers have no interest in the details of vehicle operation, but they do compare new vehicles with those that they have owned previously. If the fuel injection system can allow improved performance by comparison with a similar, carbureted vehicle, then there should be no problem with acceptance.

15. STRATIFIED CHARGE ENGINES—SPARK IGNITION

The concept of a stratified charge engine has received considerable attention in most discussions of automobile economy. Since this engine concept differs from the familiar design used in conventional automotive engines, some explanation of the operating principles and pertinent terminology is appropriate.

Modern automobile engines are almost universally of the *carbureted, spark ignition* type. Fuel and air are mixed in the *carburetor*, and this mixture is conducted to the engine *cylinders* through the *intake manifold*. The process which occurs in the carburetor results in a mixture of air and fuel droplets. The fuel vaporization begins at the carburetor and continues throughout the passages of the intake manifold. When the air/fuel mixture is delivered to the cylinder, approximately 60 percent of the fuel is in the vapor state[1]. It is important to note that liquid fuel is present in the intake manifold; the conditions of liquid-vapor equilibrium prohibit the operation of a "dry" manifold unless all of the fuel is in the vapor state. The liquid deposited on the walls of the intake manifold will, in subsequent discussion, be connected with excessive hydrocarbon emissions during deceleration.

The mixture of air, liquid fuel, and fuel vapor is allowed to enter the cylinder at the appropriate time during the cycle through the *intake valve*. The valve usually opens when the piston is near the bottom of its stroke, and fuel vaporization continues during the compression process. In addition, a stirring motion is provided to the mixture by passage through the valve. After compression, at the time when the sparkplug fires, there is a homogeneous mixture of air and fuel throughout the combustion chamber; if the mixture is chemically correct, then combustion proceeds across the cylinder. It should be observed that there is a thin region adjacent to the cylinder walls into which the flame cannot propagate; the mixture in this *quench volume* is chemically altered but not consumed.

In order to provide a quantity of fuel appropriate to the load, the conventional engine is equipped with a *throttle*. The throttle is basically a valve which regulates the amount of air supplied to the engine; increasing airflow increases the pressure drop in the carburetor venturi, and more fuel is supplied.

The primary difference between conventional and stratified charge engines may be recognized by comparing the mixtures present in the cylinder at the time of ignition. In the above discussion, it was observed that the conventional engine utilizes a homogeneous mixture of fuel and air; however, in the stratified charge engine, the carburetor is eliminated. Air alone enters the combustion chamber through the intake manifold and is compressed. A short time before ignition, an amount of liquid fuel appropriate to the load is sprayed into the combustion chamber. Instantaneously, therefore, there is a region of the combustion chamber which is occupied almost entirely by fuel; the remaining volume is occupied by air. This idealized geometric configuration provides the origin of the term "stratified" in the description of the engine concept.

The above description is obviously oversimplified. There is actually a pronounced motion of the air in the combustion chamber, and usually this motion has the form of a swirl or vortex induced by the flow through the valve. Since the injection of the fuel requires some time, the fuel-rich region of the combustion chamber may more accurately be described as a cone, with the apex at the injection nozzle, which follows the swirling flow. Ignition occurs at the sparkplug, and successful combustion requires that fuel and air both be present at the plug when the spark occurs. Furthermore, since liquid fuel will not burn, some vaporization of the fuel must occur prior to ignition. The vaporization of fuel actually commences as soon as fuel enters the chamber and continues throughout the combustion process.

It may be observed that the combustion process in a stratified charge engine is quite similar to that in a diesel engine. This is in fact the case, except that the compression ratios are much lower in the stratified charge engine (10:1 or less), and combustion is initiated by a spark rather than by the high temperature which results from compression in the diesel engine. The stratified charge engine concept allows diesel efficiencies to be approached with a spark-ignition, multifuel engine. These are the main characteristics which motivate the proponents of the stratified charge concept.

Experiments with stratified charge engines have been conducted since 1920, and there are several modern engine designs that are in the stratified charge category for which a considerable body of test data are available. It will be worthwhile to devote some attention to the details of these experimental systems. A listing of the various types is shown in Table 18.

Considerable attention has been devoted in the popular press to the Compound Vortex Controlled Combustion (CVCC) engine developed by Honda. The engine is equipped with a carburetor and precombustion chambers designed to allow burning of lean mixtures; a diagram is shown in Figure 73[40]. The Honda design is quite similar to an earlier Russian design[41] and to a retrofit device investigated by an American firm early in the 1960's[42]. Since the engine employs a carburetor rather than fuel injection, load control is achieved by variation in the position of a throttle valve located in the air inlet. This is the same load control method used by conventional carbureted engines, and, therefore, the CVCC design does not realize the full potential of the stratified charge concept. Furthermore, the vehicles which have been tested were adjusted to meet 1975 emission standards, and an economy penalty of 10 to 20 percent, by comparison with 1973 certification vehicles, was observed (Reference 40). There may be some evidence of reduced power output. Engine displacement for the CVCC engine is substantially greater than that of the conventional engine for the same vehicle (71 CID vs 119 CID). A report of more recent conversion of a Chevrolet 350 CID engine to CVCC operation is encouraging; 160 horsepower was achieved at 3700 rpm for the stock engine, and 4000 rpm for the CVCC conversion[43]. The same report also suggests fuel economy values competitive with those of other vehicles in the same class during operation on the LA-4 cycle. Emission test results for the 350 CID conversion were also favorable. Configuration similar to the Honda design has been investigated by Fiat[44] and by Volkswagen[45]. In the latter case, experiments were also conducted using a fuel injection system to provide enrichment. During the Volkswagen experiments, it was also observed that performance and emissions at light loads could be improved through the use of a throttle.

Several stratified charge engines have been designed which employ precombustion chambers but minimize the throttling of intake air. An example of this type is the Broderson Engine, which has been examined at intervals over the past 30 yr[40]. In this design, shown in Figure 73, a carbureted (but unthrottled) fuel/air mixture is supplied to the main chamber; fuel injection is used to provide the rich mixture required for ignition in the precombustion chamber. The reported results indicate

[40]Bascunana, J., "Divided Combustion Chamber Gasoline Engines—A Review for Emissions and Efficiency," APCA, 66th Annual Meeting, 1973.
[41]Gussak, L.A., *Izvestiga ANNSSSR, Energertika i Transport*, Nr 4, (1965).
[42]Rhodes, K.H., "Project Stratofire," SAE Paper 660094, 1966.
[43]Anon, "An Evaluation of a 350 CID Compound Vortex Controlled Combustion (CVCC) Powered Chevrolet Impala," Environmental Protection Agency Report 74-13 DWP, October 1973.
[44]Tienert, R.M., "Automotive News," 4 June 1973.
[45]Heitland, H., "A Status Report on the Pre-Chamber Injection Volkswagen Stratified Charge Engine," First Symposium on Low Pollution Power System Development," October 1973.

TABLE 18. INTERNAL COMBUSTION ENGINE SYSTEMS–NON-HOMOGENEOUS FUEL/AIR MIXTURES

Category	Example	Fuel admitted by	Stratified by	Mixing before ignition	Mixing after ignition	Load controlled by
Open chamber–compression ignition	D.I. diesel	Injection	Low fuel volatility and injection details	Interaction of fuel injection with air motion (swirl)	Continued effects of injection, thermal-centrifugal effects	Fuel-input
	MAN system	Injection	Similar to diesel; in addition fuel is injected parallel to combustion chamber wall with high swirl	Similar to diesel, but to a much lesser degree due to injection technique	Thermal-centrifugal	Fuel-input
Open chamber–spark ignition	Texaco Ford Witzky/SwRI Hesselman	Injection	Controlled air motion and precise fuel injection	Interaction of fuel injection with air motion (swirl)	Continued effects of injection, thermal-centrifugal effects	Fuel-input, sometimes also air-throttled
Divided chamber–compression ignition	I.D.I. diesel	Injection	Fuel injection into pre-combustion chamber	Fairly complete mixing in pre-combustion chamber due to high swirl	Expansion of burning contents of pre-combustion chamber into the main chamber	Fuel-input
Divided chamber–spark ignition	Broderson eng. SwRI "pumpless gas"	Injection	Fuel injection into pre-combustion chamber	Fairly complete mixing in pre-combustion chamber due to swirl & turbulence	Expansion of pre-chamber contents, inflow/outflow due to piston motion	Fuel-input
	Honda CVCC	Carburetion	Not stratified, richer mixture in pre-chamber to serve as high energy ignition source for main chamber	Premixed	Expansion of pre-chamber contents into main chamber	Throttling of fuel and air
	SwRI 71 series conversion	Injection	Not stratified in pre-chamber. Stratified in main chamber by swirl & density difference between gas and air	Complete mixing in pre-chamber due to turbulence	Expansion of pre-chamber contents into main chamber	Fuel-input

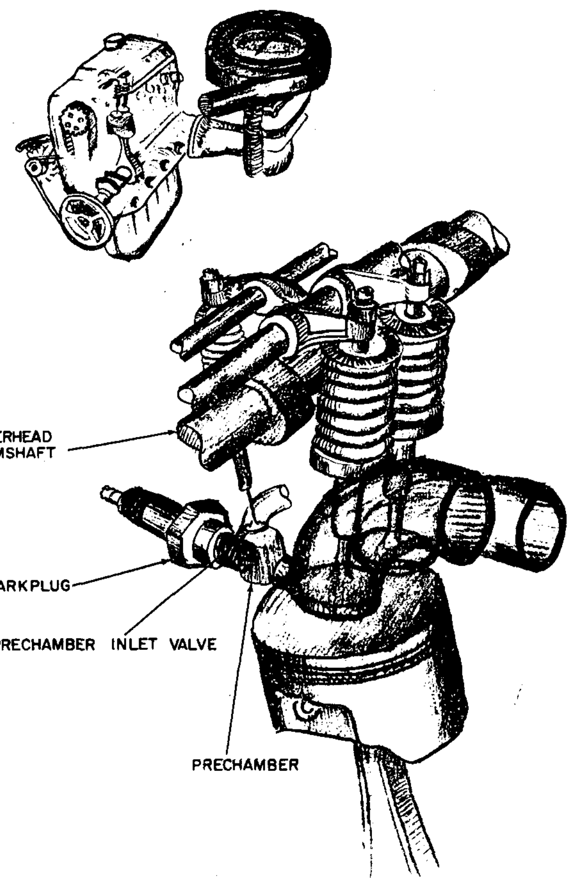


FIGURE 72. HONDA CIVIC CVCC ENGINE

a high thermal efficiency over the entire load range during operation at two speeds; more data obtained during variable speed operation would be helpful.

The stratified charge engine concept, which has received the most comprehensive attention, is the open chamber design typified by the Texaco (TCCS), Ford (FCP, PROCO), and Witzky/SwRI engine[46-58]. These designs differ primarily in geometric detail; all employ direct fuel injection, and all rely on motion of the air charge induced by flow through the valve to produce charge stratification. Figure 74 depicts a typical open chamber stratified charge design. Engines of this type are usually operated either without throttling of the intake air or with throttling at very low (idle) loads only. It is apparent that successful ignition depends upon the presence of an appropriate mixture at the sparkplug when the spark occurs. The assurance of this condition over the entire operating range has been one of the primary problems in the development of the open chamber engine. Performance data obtained from engine tests indicate favorable fuel economy, and typical engine map is shown in Figure 75. Fuel maps of this type are

[46]Coppoc, W.J., E. Mitchell and M. Alperstein, "A Stratified Charge Multifuel Engine Meets 1976 U.S. Standards," 38th mid-year meeting of the API Division of Refining, May, 1973.
[47]Davis, C.W., E.M. Barber, and E. Mitchell, "Fuel Injection and Positive Ignition–A Basis for Improved Efficiency and Economy," SAE Paper 190A, SAE Summer Meeting, 1960.
[48]Mitchell, E., J.M. Cobb, and R.A. Frost, "Design and Evaluation of a Stratified Charge Multifuel Military Engine," SAE Paper 680042, 1968.
[49]Mitchell, E., M. Alperstein, J.M. Cobb, and C.H. Faist, "A Stratified Charge Multifuel Military Engine–A Progress Report," SAE Paper 720051, 1972.
[50]Cobb, J.M. and E. Mitchell, "Performance Development and Evaluation of the Multifuel Texaco Combustion Process Model 2A042 Military Standard Engine," Final Technical Report to U.S. Army Mobility Command, Contract No. DA-44-009-AMC-991(T) Mod 1, 1967.
[51]Witzky, J.E., "Stratification and Air Pollution," Institution of Mechanical Engineers Paper C136/71, 1971.
[52]Hussman, A.W., F. Kahoun, and R.A. Taylor, "Charge Stratification by Fuel Injection into Swirling Air," Presented at 1962 SAE Combined National Fuels and Lubricants, Power Plant, and Transportation Meeting, 1962.
[53]Witzky, J.E., and J.M. Clark, Jr., "A Study of the Swirl Stratified Combustion Principle," SAE Paper 660092, 1966.
[54]Willis, D.A. and W.E. Meyer, "Investigation of a Stratified Charge Engine Employing the Air Swirl Induced Stratification Principle," Pennsylvania State University Interim Report to U.S. Army Mobility Command on Contract No. DA-36-034-ORD-3638T, 1964.
[55]Witzky, J.E., "Stratified Charge Engines," ASME Paper 63-MD-42, 1963.
[56]Bishop, I.N., and Aladar Simko, "A New Concept of Stratified Charge Combustion–The Ford Combustion Process (FCP)," SAE Paper 680041, 1968.
[57]Bishop, et al, U.S. Patent 3,315,640,1967.
[58]Simko, A.M., Choma & L.L. Repko, "Exhaust Emission Control by the Ford Programmed Combustion Process," SAE Paper 720052, 1972.

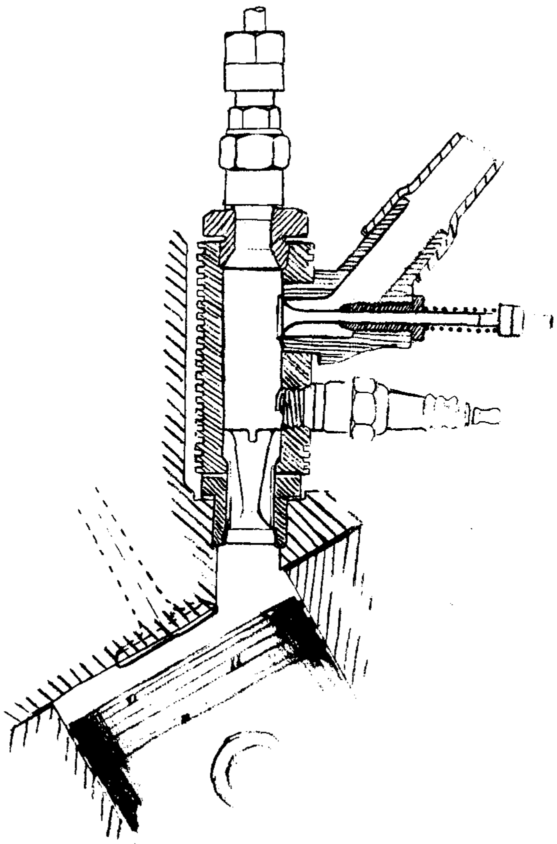
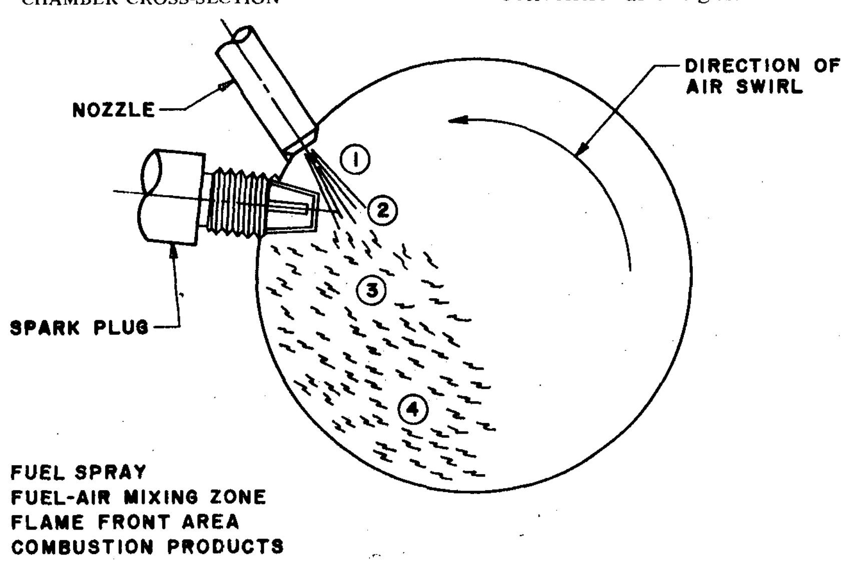


FIGURE 73. BRODERSON ENGINE COMBUSTION CHAMBER CROSS-SECTION

apparently not available for emission controlled engines, although the features of the combustion process indicate that low emissions are characteristic of the stratified charge design. For example, combustion occurs in a richer than stoichiometric mixture, but the overall mixture in the chamber is lean. The rich mixture in the combustion zone tends to inhibit formation of NO_x, while the excess air tends to promote complete oxidation of hydrocarbons and carbon monoxide. Furthermore, the fuel is consumed as it enters the combustion chamber, and the hydrocarbon concentration in the quench volume is much smaller than that which is considered typical of an engine operating with a homogeneous charge.

In summary, the stratified charge engine has several desirable features which warrant further investigation and development. Foremost among these advantages is a significant improvement in thermal efficiency. In addition, the stratified charge concept offers a decrease in the complexity of emission control procedures and, in some versions, a multifuel capacity not available with conventional designs.



1 FUEL SPRAY
2 FUEL-AIR MIXING ZONE
3 FLAME FRONT AREA
4 COMBUSTION PRODUCTS

FIGURE 74. TEXACO CONTROLLED-COMBUSTION SYSTEM

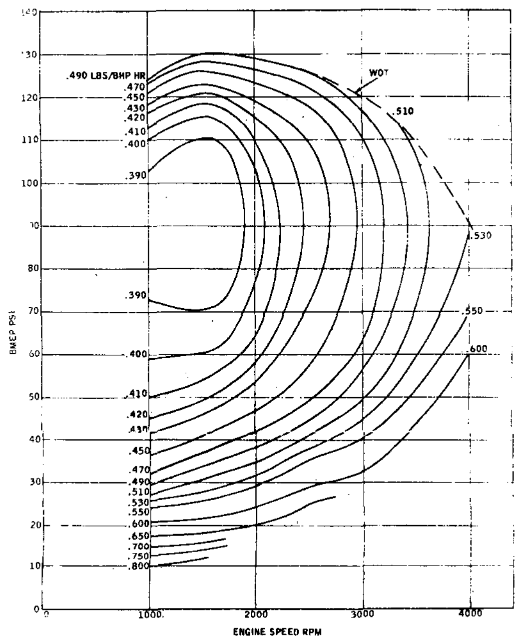


FIGURE 75. BSFC ISLAND CURVE (430 CID FCP ENGINE)

Stratified Charge Engines—Evaluation

Identification of Improvements

Stratified charge engines, particularly the open chamber variety, offer a substantial decrease in fuel consumption by comparison with conventional engines having the same displacement. Specifically, the available test data indicate that thermal efficiencies characteristic of diesel engines can be attained at lower compression ratios with spark-ignition, stratified charge engines. Table 19, from Reference 46 shows the results of some tests.

Fuel Consumption Reduction

The fuel consumption for several engine sizes and two vehicle sizes was estimated, according to the procedure outlined previously in this document, for engines having an open chamber design. The source of fuel consumption data was the FCP engine map (Figure 75), and for engine sizes other than 430 CID, the abscissa was interpreted in terms of piston speed. The results of the calibration are compared to the results for the baseline vehicle in Table 20. The fuel consumption at idle was assumed to be equal to that of the reference vehicle rather than reduced to values characteristic of diesel engines. It may be observed from the table that a pronounced decrease in fuel consumption occurs upon reduction of engine displacement, even at constant vehicle weight. This feature is a consequence of the shape of the contours on the engine map. As shown in Figure 75, the brake specific fuel consumption contours in the lower left quadrant have a small positive slope, in contrast to the same lines on the conventional spark-ignition engine map (Figure 3). Other maps for open chamber stratified charge engines are similar to Figure 75; this line shape is assumed to be characteristic of the engine design.

TABLE 19. OVER-THE-ROAD FUEL ECONOMY–M-151 VEHICLE

Course	Avg speed, mph (km/h)	Fuel economy, mpg (l/100 km) Standard gasoline	Naturally aspirated TCCS	Percent improvement
1[a]	21(33.8)	12.5(18.8)	21.6(10.9)	72.8
2	25(40.2)	16.6(14.2)	24.8(9.5)	49.4
3	33(53.1)	17.5(13.4)	24.4(9.6)	39.4

(a)Includes high portion of idle time.

TABLE 20. FUEL ECONOMY IMPROVEMENTS OF OPEN CHAMBER STRATIFIED CHARGE ENGINE

Engine displacement	Inertia weight	Uncontrolled emissions			Emissions effects considered
		Urban cycle (%)	Road load (%)	Composite (%)	Composite (%)
430	4500	12.6	23	17	—
350	4500	27	31	29	34
300	4500	45	46	45.4	—
300	4000	51	52	52	—

The effects of emission controls on fuel economy were taken into account as follows:

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{D}{B}\right)\left(\frac{A}{C}\right)$$

where

A – fuel economy of stratified charge meeting 0.41-3.4-2.0 emissions

B fuel economy of conventional engine meeting 1973 standards

C fuel economy of stratified charge, no emission controls

D fuel economy of conventional engine, no emission controls.

Since all previous comparisons have been on the basis of equal performance, only the 350 CID engine was analyzed for emission effects. For this engine, from Table 20, C/D = 1.29. The ratio D/B has been assumed to be 1.09. The ratio A/C is primarily affected by the NO_X requirement. The subsequent discussion on emission control in the stratified charge engine and Figure 76 indicates that the economy loss for 2.0 g/miles of NO_X is roughly 3 to 5 percent. It is assumed that A/C = 0.95. Therefore,

$$A/B = (1.29)(1.09)(0.95) = 1.34$$

This result is shown in Table 20.

It should be noted that the tests reported in Reference 43 indicate that the fuel economy for the Honda CVCC design, in the 350 CID configuration, is competitive with that of other vehicles in the same class.

Stage of Development

The stratified charge engine, in one of its various configurations, has been tested at intervals since 1920 when Ricardo performed the first experiments. The concept has apparently been resurrected

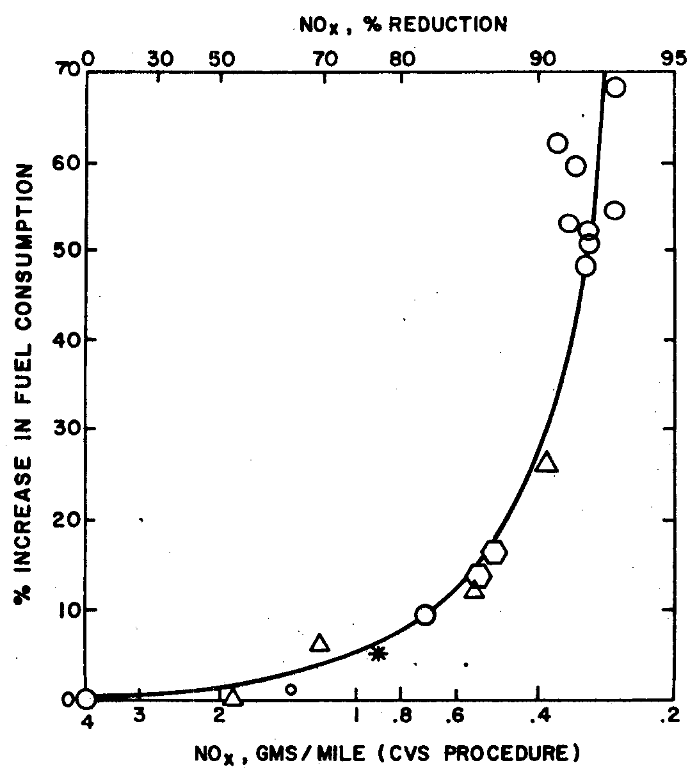


FIGURE 76. EXPERIMENTAL STRATIFIED CHARGE ENGINES

at times when change was imminent within the industry, probably as one of several alternatives available at the time. The fact that the decision was always made against final development and production can probably be attributed to increased cost of the engine. High efficiency stratified charge engines require fuel injection pumps similar to those on diesel engines, and this cost has been regarded as prohibitive for automobile engines. At the present time, however, required performance in the areas of economy and emissions may allow the evaluation to swing in the other direction. It should also be pointed out that conventional engines have evolved over a period of half a century; the engines of today bear only a family resemblance to those of the 1920's, and no candidate engine system has had the benefit of the practice obtained from production runs amounting to millions of engines. The most recent reports [46-58] indicate that considerable progress has been made in stratified charge engine design within the last 5 yr; hence, further development should be encouraged.

Demonstrable by 1976

In addition to the Honda tests for carbureted engines, open chamber stratified charge engines have been demonstrated in vehicles capable of road operation[46,56]. Furthermore, at least one program is presently underway which involves conversion of a fleet of military vehicles to stratified charge operation. Results will be available by 1976.

Reliability

The basic engine block of a stratified charge engine is identical to that of a conventional engine, however, the engine heads are different. The open chamber engine requires the addition of a fuel injection nozzle, and the divided chamber engine generally requires an extra valve. Reliability for the carbureted stratified charge engine, except for the extra valve, should be equivalent to that of a conventional engine. In the case of an open chamber engine, a fuel injection pump is added but the carburetor is eliminated. Considerable experience has been obtained from the manufacture of diesel fuel injection pumps. The only obstacle to very high reliability is the cost trade-off which will occur when injection pumps are placed in passenger car service. Fuel processing for a fuel injection system must be performed carefully; filtration techniques common to diesel service must be employed. After the procedural details are specified for production engines, the system reliability should be equivalent to that of present engines.

Cost

In the case of the open chamber stratified charge engine, the cost of the basic engine should not exceed that of conventional engines. However, the cost of the fuel injection system is offset only by the cost of the present carburetor. Fuel injection pumps and nozzles are notoriously expensive because of the precision which must be maintained in their manufacture, but they have never been produced in quantities appropriate to passenger car service. One proponent of stratified charge engines has devoted considerable attention to cost reduction, and further reduction is possible in planning for production[56].

The completion cost situation cannot be considered without some attention to the consequences of conversion to a different concept of engine operation. The emission control philosophy, which has prevailed until the present time, has employed standard reciprocating engines with minor design changes. The performance of these engines has been modified with accessories in order to meet emission standards. Thus, a 1973 vehicle typically has a conventional engine with modified ignition, fuel, and exhaust systems. By all indications, this trend will continue throughout most of this decade; the quantity of added control equipment will simply be increased to accommodate the prevailing emission standards. If, however, a change in basic engine design is assumed, and if the new engine has inherently low emission properties, then the amount of accessory control equipment is reduced. The reduction in cost as a result of unnecessary control equipment may be applied to the additional cost of engine conversion.

Another additional cost factor may be encountered for some stratified charge engines. The open chamber engine operates without an air throttle; it produces no manifold vacuum. If designers of various auxiliary systems, such as climate control, persist in the use of vacuum for the performance of tasks such as valve opening, it will then be necessary to equip the engine with a vacuum pump. In every case except the engine load signal to the automatic transmission and the ignition system, however, there is presently an equivalent method of performing routine control functions.

The net additional initial cost for an open chamber stratified charge engine should not exceed $150 to 200 per vehicle. After experience has been obtained by service and aftermarket personnel, maintenance and repair costs should be equivalent to those for the conventional carbureted engine.

Safety

The safety aspects of the engine are identical for both conventional and stratified charge engines. The stratified charge system could be regarded as somewhat safer if its capability for burning fuels other than gasoline was utilized. A heavier fuel, such as diesel oil, is much less hazardous than gasoline in the event of an accidental spill.

Another safety feature revolves around the basic engine operating principle. Because the open chamber engine is unthrottled, "compression braking," or use of the engine to slow the rate of descent on a grade, is not available. This problem would be overcome by the use of an idle-only throttle, and, in fact, this feature has been used in several of the candidate engines for idle fuel control. If a completely unthrottled engine is ultimately used, then the problem of retardation on grades would be transferred to the vehicle braking system.

Emissions Standards

Most of the development work on stratified charge engines was performed prior to the current interest in vehicle emissions; therefore, the quantity of emission test data obtained with modern instrumentation is not large. Although certification testing is not complete, the fact that the Honda CVCC engine performed well during EPA tests received wide attention in the popular press, and some comprehensive data are available[43]. For open chamber engines, which generally exhibit better economy, some emission data are available in References 46 and 58. As mentioned above, the open chamber engine is an inherently low emission design by virtue of the nature of the combustion process. It should also be noted that the direct fuel injection and positive ignition allow good cold starts without the need for fuel enrichment during warmup, and fuel cutoff during deceleration is readily obtainable. The authors of Reference 46 have reported the results of emission tests in which a 141 CID military engine, turbocharged and converted to open chamber stratified charge operation, was compared to its unmodified counterpart. These results are reproduced in Table 21. It should be noted that neither vehicle was tuned for optimum emission control. Also included in Table 21 are test results obtained after the addition of an oxidation catalyst to the vehicle exhaust system. An effort to achieve maximum emission control through the use of catalytic reactors, exhaust recirculation, and optimum tuning resulted in compliance with the most stringent Federal regulations, as shown in Table 22; however, a severe fuel penalty was incurred, as shown in

TABLE 21. EXHAUST EMISSIONS–TURBOCHARGED TCCS M-151

CVS 3000-lb Inertia	NAPCA Data LA4-S3 Cycle		Cold Start Fuel–Gasoline
Emissions	Std M-151 (gm/mi)	TCCS M-151(gm/mi)	
		w/o catalyst	with catalyst
HC	6.40	3.85–4.58	1.74–1.97
CO	76.22	9.08–9.62	2.29–2.69
NO_X(a)	3.39	1.74–1.92	1.81–2.23

(a)As NO_2 determined by fuel cell type instrumentation.

TABLE 22. EXHAUST EMISSION LEVELS–CONTROLLED TCCS M-151

1975 CVS C/H
2750-lb Inertia

Lead-Free Gasoline

Laboratory	Mass emissions (gm/mi)		
	HC	CO	NO_X (a)
Texaco	0.25	1.17	0.33
Texaco	0.28	0.62	0.32
Texaco	0.30	0.79	0.29
EPA	0.40	0.26	0.30
EPA	0.33	0.15	0.31
EPA	0.37	0.30	0.31
1976 Standards	0.41	3.4	0.40

(a)NO_X by chemiluminescent analyzer.

TABLE 23. EFFECT OF EMISSIONS CONTROL ON FUEL ECONOMY–M-151 TCCS

1975 CVS CIH
2750-lb Inertia

Lead-Free Gasoline

Test site and engine Configuration	Mass emissions (gm/mi)			Fuel economy (mpg)
	HC	CO	NO_X	
Texaco–std. Configuration(a)	0.36	0.61	0.31	16.2
Texaco–reduced EGR(b)	0.48	0.57	0.45	17.6
EPA–std. Configuration(a)	0.37	0.24	0.31	15.8
EPA–reduced EGR(b)	0.50	0.14	0.70	21.9

(a)Average of three determinations.
(b)One determination.

Note: Fuel economy of standard, uncontrolled, carbureted vehicle emitting at levels shown in Table 21: 13–14 mpg.

Table 23. In addition, the table indicates the results of a reduction in the rate of exhaust recirculation; some of the economy decrease can be recovered at the expense of slightly higher emissions. The extent to which emission controls affect fuel economy is graphically illustrated in Figure 76, which depicts increase in fuel econsumption compared to extent of NO_X control.

In another series of experiments, other investigators concluded that an open chamber stratified charge engine could satisfy the most stringent emission requirements, but the fuel economy benefits of the stratified charge operation were diminished in the process[58].

The data associated with the above discussion indicate that hydrocarbon emissions present the most severe control problem when the 0.4-3.4-2.0 standards are considered. Qualification of the system under these standards will probably require light EGR and substantial catalytic treatment of the exhaust. It has been found that throttling devices can be used to assist in the maintenance of appropriate catalyst temperatures; however, the presence of a throttle diminishes the fuel economy benefits. A series of catalytic reactor elements, arranged parallel and connected to an appropriate exhaust system, should allow the attainment of a proper catalyst temperature over the entire load range of the engine.

The open chamber stratified charge engine does not offer the entire solution for both economy and emission control, but the basic concept appears to have advantages from both viewpoints. Given the benefits of further development and the experience gained with emission control of other spark-ignition engines, the open chamber design appears to be a viable candidate for a low emission, low fuel consumption replacement for the conventional engine.

The stratified charge engine is subject to the same odor considerations as in the diesel engine.

Noise

Some investigators, particularly the author of Reference 40, have reported extreme combustion noise during the operation of some stratified charge engines. However, the noise seems to be sensitive to the geometric arrangement and dimensions of the combustion chamber, and reduction to acceptable levels is possible with proper design.

Performance

In general, it has been observed that the performance of open chamber stratified charge engines is improved by comparison with baseline vehicles. The apparent decrease in performance of the Honda carbureted, divided chamber engine can be attributed to the throttling of the intake air and the efforts of the manufacturer to comply with emission standards without using additional control devices.

Time Required for Implementation

As mentioned previously, a fleet of military vehicles is presently being converted to open chamber stratified charge operation; the testing program will continue through 1977. If the tests are entirely successful, it is possible that some production engines could be available by 1980. A need for further development determined as a result of this program, however, would delay the commitment to produce the engine. The probability of engine availability by 1980 would be greatly enhanced by concurrent development programs beginning in 1974.

The carbureted, divided chamber engine may enter production as early as 1975. However, advantages of this engine in the area of fuel economy have not been clearly demonstrated.

Consumer Acceptance

As demonstrated by the results from Reference 46, an open chamber stratified charge engine should be virtually indistinguishable from a conventional engine from the viewpoint of the operator. There have been problems, during the development programs, with combustion harshness, but it is thought that these problems can be minimized as experience is gained. The transient response is regarded as excellent and the driveability as very good[46]. The implications of an unthrottled engine in the area of safety have been discussed; some driver education may be necessary if completely unthrottled engines are used. The engine is capable of providing power for accessories using any of the schemes currently available or proposed. As previously mentioned, an accessory vacuum pump may be required in order to supply traditionally vacuum power devices.

16. DIESEL ENGINE

General

In discussing the general characteristics of diesel engines, it is convenient to consider the differences between the diesel and the spark-ignited carbureted engine and the consequences of these differences.

The major characteristic of the diesel is its method of obtaining fuel ignition. The fuel is ignited by compression of the fuel-air charge to a temperature where self-ignition of the fuel occurs. This phenomenon is not only controlled by temperature but by the duration of time the fuel is maintained at temperature, so that both short time -high temperature and long time -low temperature will cause self-ignition. The required engine speed sets the time duration available for this process, and the engine compression ratio is used to control the temperature.

It is not necessary in this report to consider the complex interrelationships of fuel cetane number, ignition delay, injection duration, injection timing, and engine compression ratio. It is sufficient to point out that years of development have led to the present method of operation in which a diesel fuel with a relatively short ignition delay (high cetane number) is used, combined with fuel injection into the combustion chamber shortly before ignition is desired, using compression ratios considerably higher than carbureted engines.

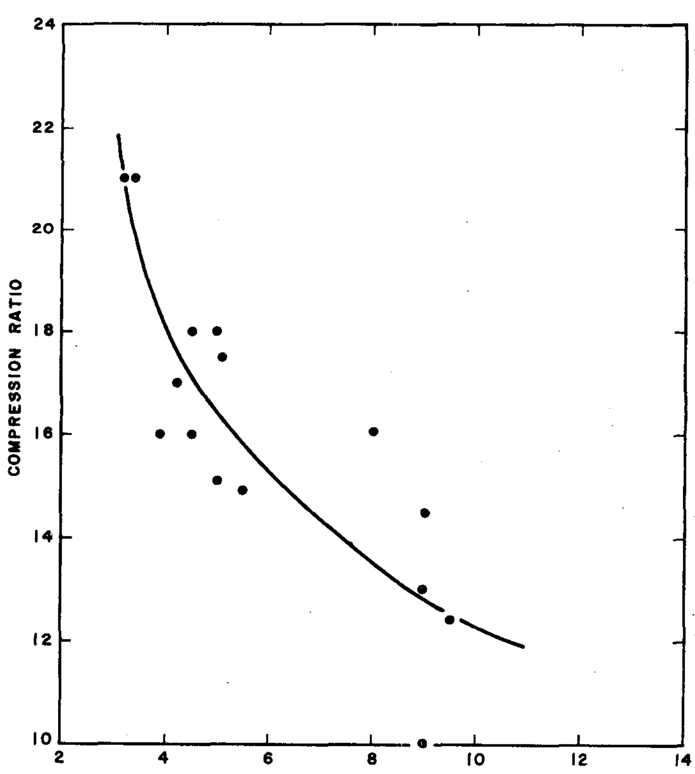


FIGURE 77. CHARACTERISTICS OF PRODUCTION DIESEL ENGINES

Thus, the diesel engine is without a spark-ignition system, and ignition timing is controlled by fuel injection timing. Figure 77 shows compression ratios of production diesel engines versus engine bore diameter. The trend of increasing compression ratios with decreasing bore is a strong one and is important to this discussion. Compression ratios of diesel engines are set, in practice, by the requirement for cold-starting. The trend of compression ratio with bore size is due to two primary factors:

- In similar engines, the percentage heat loss is greater in the engine with the smaller bore, hence air temperatures are lower than those in larger bore engines.
- The use of prechambers is common with the small diesel engines, and heat losses are greater in prechamber engines than in other diesel designs.

- Very large bore engines are frequently operated within heated enclosures, so there are no requirements for low air temperature starting.

The high compression ratios found in small diesel engines are generally undesirable, since the rather small increase in thermal efficiency obtained from a compression ratio increase from say, 16 to 21 is small compared to the added friction losses, increased starter motor power requirement, and higher stresses imposed by the increased compression ratio.

From a fuel economy viewpoint, it may be desirable to utilize a method to vary compression ratio so that cold starts could be achieved at a high compression ratio, and the compression ratio could then be reduced to a lower value for normal operation. Means to do this add complexity and cost to the engine, and the real economic gain obtained by this strategem is probably very marginal.

In the short time period after liquid fuel is injected into the combustion chamber, the individual fuel droplets evaporate and provide a wide range of fuel-air ratios at different points within the combustion chamber. Ignition is believed to occur in regions slightly leaner than stoichiometric, but, regardless of the quantity of fuel injected, there are always local regions where the conditions for ignition are correct. Therefore, there is no need to maintain an overall fuel-air ratio within a specific range, as with the carbureted engine. This allows the diesel to run unthrottled. The engine power is controlled by fuel input and the fuel-air ratio is very nearly proportional to engine power (at a given speed).

It has been pointed out (see Section 7) that thermal efficiency increases as the fuel-air ratio is decreased. Since the carbureted engine runs within a narrow range of fuel-air ratios (in the general vicinity of stoichiometric) and the diesel runs from very low fuel-air ratios up to a fuel-air ratio of about 0.6 of stoichiometric, the thermal efficiency of the diesel is superior to that of the carbureted engine. At low loads, the effect is significantly enhanced by the absence of throttling losses. The higher compression ratio of the diesel also contributes to a gain in thermal efficiency.

TABLE 24. COMPARISON OF WEIGHTS AND VOLUMES–AUTOMOTIVE DIESEL VS. AUTOMOTIVE CARBURETED*
(Cylinder bore 3 in. to 4 in.)

Characteristic	Carbureted	Diesel
Engine weight (lb/in.^{3})		
Displacement	1-1/2–3	3–4
Engine weight (lb/hp)		
Max BHP	2–5	4–9
Engine "box" volume (in.^{3}/in.^{3})		
Displacement	30–50	70–90

*Data from Taylor, "The Internal-Combustion Engine in Theory and Practice," Vol II, MIT Press, 1968.

The increased compression ratio of the diesel leads to higher combustion pressures and requires a heavier and stronger engine construction, as compared to the carbureted engine (See Table 24). Due to the characteristics previously discussed, the diesel engine is a natural candidate for supercharging. Increased inlet air pressures and temperatures reduce ignition delay and improve combustion characteristics. The only limits to the level of supercharging achieved are set by structural considerations (due to increased combustion pressures) and thermal considerations (due to increased power output, and hence, increased heat rejection to the coolant). Supercharging is widely used in diesel engines and is most commonly affected by a turbocharger in which the air compressor is driven by an exhaust gas turbine. Turbocharger technology is well advanced. Turbochargers represent a considerable improvement over the use of a direct engine-driven blower because:

- Higher overall engine efficiencies are obtained because the loss in engine power, due to increased exhaust pressure, is less than the power required to drive a compressor from the engine crankshaft.

- Installation is relatively simple because only inlet and exhaust plumbing must be altered.

- The turbocharger is a mechanically simple device.

Drawbacks of turbochargers, compared with direct drive compressors, are:

- There is usually an acceleration lag due to the inertia of the turbine and compressor wheels.

- The cost of the turbocharger may be higher than a direct-driven blower.

The unique method of fuel delivery and ignition give the diesel its own special exhaust emission characteristics. In a well-designed engine, CO is very low due to the overall lean mixture in the chamber. HC emissions are also lower than those for the carbureted engine and not due to a wall quenching effect but to extremes of fuel-air ratios impairing combustion. NO_X formation is also reduced in the diesel partially due to lower gas temperatures at partial loads. When considering only HC, CO, and NO_X, the diesel presents a very favorable emission prospects.

However, the diesel has its own special emissions problem. At high loads, some of the injected fuel has difficulty in finding sufficient air for complete combustion and carbon particles are formed, which emerge from the exhaust as visible black smoke.

Through combustion mechanisms not fully understood, the diesel also produces a characteristic exhaust odor which is partially due to unburned fuel but also involves a number of other combustion products. Both diesel smoke and diesel odor are widely held to be objectionable, primarily from esthetic considerations,[59] although improvements can be made in both areas.

An effective strategy for reduction of diesel exhaust emissions involves the use of supercharging. At a given power output, supercharging results in a lower fuel-air ratio which reduces HC, CO, odor, and smoke. The increased power output gained from supercharging allows a retardation of injection timing that is very effective in the reduction of NO_X. The net result is reduced emissions at the expense of a lower gain in power than would be expected from the effects of supercharging and a slightly reduced fuel economy from the effects of injection retard.

The diesel engine, due to its combustion process, has a noise level higher than that of the carbureted engine. Subsequent to start of combustion, the diesel has higher rates of pressure rise in the combustion chamber than the carbureted engine, and these relatively high pressure rise rates produce audible combustion knock. This noise is most apparent at engine idle, due to increased ignition delay with higher pressure rise rates and to the absence of other vehicle and engine noises associated with high speed operation.

To summarize, the diesel engine is characterized by the following factors, in comparison to the spark-ignited carbureted engine:

- Improved fuel economy

- Greater weight and size

[59]Springer, K.J., "An Investigation of Diesel Powered Vehicle Odor and Smoke," Final Report, Automotive Research and Development Section, National Center for Air Pollution Control, Dept. of Health, Education and Welfare, 1968.

- Reduced CO, HC, and NO_x emissions
- Increased particulate emissions (smoke)
- Increased exhaust odor
- Increased noise

Fuel Economy Analysis Procedure

Naturally-Aspirated Diesel Engine

The first case considered was that of a naturally-aspirated engine with a maximum power output equal to that of a 350 CID carbureted engine. The first objective was to produce a fuel map for such an engine and to determine engine displacement.

A complete engine fuel map for a 128 CID automotive diesel was available. This engine has a compression ratio of 21 and uses a "Comet" combustion system consisting of a swirl-type prechamber containing about 75 percent of the total combustion volume. In its present configuration, this engine meets the 0.41-3.4-2.0 gaseous emission level. Preliminary calculations showed that the required diesel engine would have a displacement of about 400 cu in., so it was deemed necessary to reduce the percentage friction losses of the required engine below those of the 128 CID engine. A friction power loss for the required engine was estimated based upon a range of friction data for various diesel engines.[21] This assumed engine friction is shown in Figure 78. The reduction of friction of the required engine below that of the 218 CID engine is necessary to properly reflect the probable reduction in the compression ratio of the larger engine due to its larger bore and reduced heat losses.

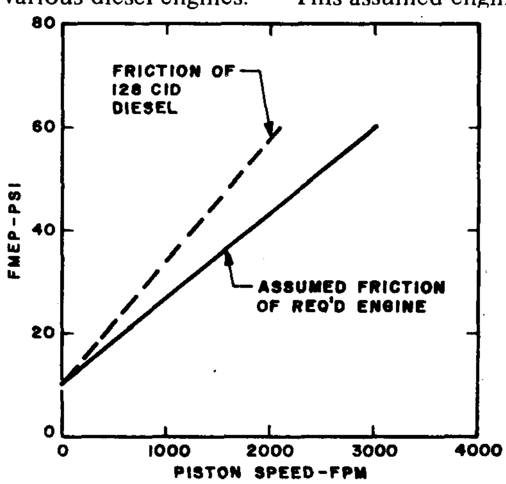


FIGURE 78. DIESEL ENGINE FRICTION

The available map for the 128 CID engine was then modified to account for the friction reduction. A maximum bmep of 115 psi was obtained with this naturally-aspirated engine, which led to a displacement requirement of 378 cu inches. The complete map is shown in Figure 79. Idle fuel flow was computed from SwRI data on similar engines.

The weight of this engine (under the present state-of-art) was estimated to be 1000 to 1100 lb, based on the data of Table 24 and the actual weights of a range of Mercedes-Benz diesel engines of similar design. Engine "box" volume (the product of overall width, depth, and length) is estimated to be from 40 to 60 percent greater than the 350 CID carbureted engine.

Using the standard fuel economy calculation procedure, the 378 CID naturally-aspirated engine shows an improvement of 24 percent in fuel consumption over that of the 350 CID carbureted engine of same maximum power in a vehicle of 4300 lb curb weight. Since the fuel consumption of

[21] Taylor, **The Internal Combustion Engine in Theory and Practice,** Volume I, MIT Press, 1968.

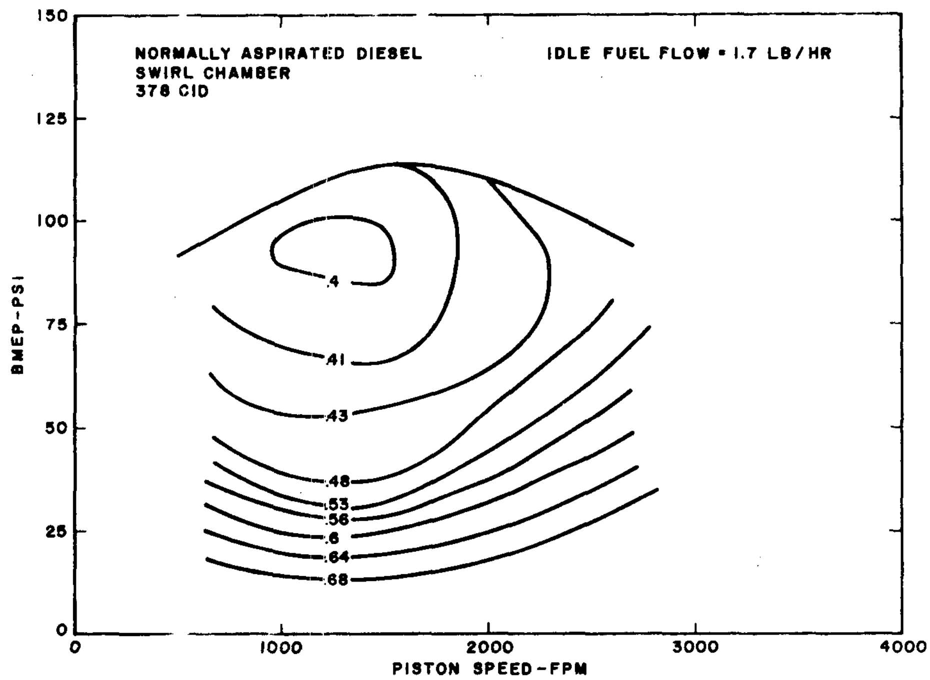


FIGURE 79. PERFORMANCE MAP NORMALLY ASPIRATED DIESEL SWIRL CHAMBER 378 CID

the carbureted engine is that without emission controls, and since the increase of displacement of the 378 CID diesel (over that of the 128 CID engine that meets the 0.41-3.4-2.0 emission standards) alters its emission characteristics, this base fuel economy improvement was modified to account for these factors. This was done using the following equation:

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

where

A – fuel economy of 378 CID diesel meeting the (0.4-3.4-2.0) emission standard.

B – fuel economy of 350 CID carbureted engine meeting 1973 emission standard.

C – fuel economy of 378 CID diesel, as calculated above.

D – fuel economy of 350 CID uncontrolled carbureted engine.

From the above analysis,

$$\frac{C}{D} = 1.24$$

The factor A/C involves the effect of the required change in the degree of emission control in a diesel engine when the engine displacement and vehicle weight is increased, since the 128 CID diesel (in a 2800 lb vehicle) meets the 0.41-3.4-2.0 emission standard and the design of the 378 CID diesel is similar, except for decreased compression ratio. The equation describing the relative power outputs of the two engines is:

$$\frac{P_1}{P_2} = \frac{(DN\eta_v\rho Fh_v\eta_t\eta_m)_1}{(DN\eta_v\rho Fh_v\eta_t\eta_m)_2}$$

For equal η_v, ρ, and h_v and η_t in both engines, the equation reduces to

$$\frac{P_1}{P_2} = \frac{(DNF\eta_m)_1}{(DNF\eta_m)_2}$$

Estimates of the relative power requirements of the two engines in their respective vehicles, the relative mechanical efficiencies due to a reduced compression ratio in the larger engine, and the known displacement and speed ratios (assuming equal piston speeds) yield the estimated average fuel-air ratio required by the larger engine in the same operating cycle to be 20 to 40 percent less than that required by the smaller engine. Thus, assuming good design, the 378 CID engine will also meet the 0.41-3.4-2.0 emission standards, will probably have a lower level of gaseous emissions and odor, and will certainly have reduced smoke levels. The factor A/C is, therefore, not less than 1.0, and, for the sake of conservatism, 1.0 will be used.

As previously discussed, the ratio D/B is estimated to be 1.09. Therefore, the estimated fuel economy improvement of the 378 CID naturally-aspirated diesel in a 4300 lb curb weight vehicle, as compared to a 350 CID carbureted engine in the same vehicle with the same maximum power, is given by

$$\frac{A}{B} = (1.24)(1.0)(1.09) = 1.35$$

The above value is a comparison of miles per gallon. Since diesel fuel and gasoline have different heating values per gallon, the same comparison based on miles/Btu is 1.26.

Turbocharged Diesel Engine

The turbocharging of diesel engines, as explained above, leads to improvement in performance and emissions as well as to lighter and more compact engines. The engine map of the 378 CID naturally-aspirated diesel was modified to represent the performance levels to be expected from turbocharging. First, a maximum turbocharged bmep of 175 psi was chosen. This is a relatively high bmep but is considered to be reasonable in view of the infrequent operation of automobile engines at maximum output. An engine with this bmep requires a displacement of 260 cu in. to match the maximum power of the 350 CID carbureted engine. A boost pressure ratio of about 1.5 is required to achieve this bmep level. It was assumed that the fuel-air ratio in the turbocharged engine would be 0.9 that of the naturally-aspirated diesel engine at full load as a further means to reduce smoke and odor. This provided a 3-percent gain in thermal efficiency at full load compared to the naturally-aspirated engine. The friction of the 260 CID engine was assumed to be equal to that of the 378 CID diesel. From this information, the curve of maximum bmep for the 260 CID turbocharged engine could be calculated from the data of Figure 79 for the 378 naturally-aspirated engine.

At part loads, the transition from naturally-aspirated to turbocharged was made on the following equation:

$$\frac{bmep_1}{bmep_2} = \frac{(\rho \eta_v F h_v \eta_t \eta_m)_1}{(\rho \eta_v F h_v \eta_t \eta_m)_2}$$

The subscripts refer to the 378 CID naturally-aspirated engine and the 260 CID turbocharged engine, respectively. At equal fuel-air ratios, $\eta_{t1} \approx \eta_{t2}$. It was also assumed that $\eta_{v1} = \eta_{v2}$ and $h_{v1} = h_{v2}$.

Then

$$\frac{bmep_1}{bmep_2} = \frac{\rho_1}{\rho_2} \frac{\eta_{m1}}{\eta_{m2}} = \frac{\rho_1}{\rho_2} \left(\frac{1 + fmep_2/bmep_2}{1 + fmep_1/bmep_1} \right)$$

Further,

$$\frac{bsfc_1}{bsfc_2} = \frac{\eta_{m2}}{\eta_{m1}} = \left(\frac{1 + fmep_2/bmep_2}{1 + fmep_1/bmep_1} \right)$$

From these two equations, the performance map for the turbocharged engine could be calculated from that for the naturally-aspirated engine. The result of these calculations is shown in Figure 80.

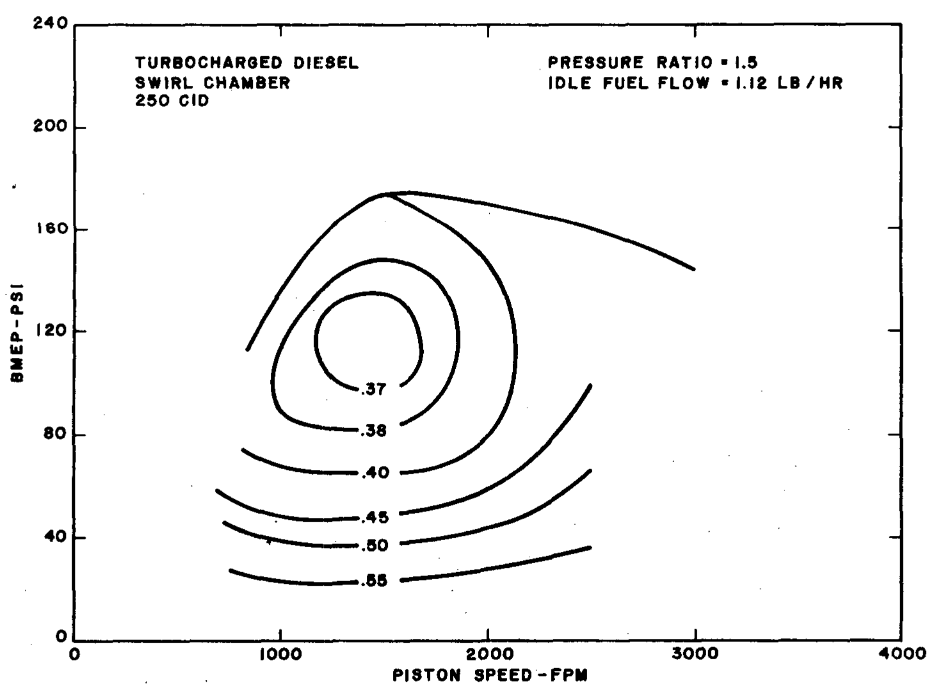


FIGURE 80. PERFORMANCE MAP TURBOCHARGED DIESEL SWIRL CHAMBER 250 CID

Using the standard calculation procedure, the improvement is fuel economy of the 260 CID turbocharged diesel is 55 percent over that of the 350 CID (uncontrolled emission) carbureted engine. In order to obtain a fuel economy comparison based upon the desired level of emission control, the following equation was used, as before:

$$\frac{A}{B} = \left(\frac{C}{D}\right)\left(\frac{A}{C}\right)\left(\frac{D}{B}\right)$$

where

A – fuel economy of 260 CID turbocharged diesel meeting the 0.41-3.4-2.0 emission standard

B – fuel economy of 350 CID carbureted engine meeting 1973 emission standards

C – fuel economy of 260 CID turbocharged diesel, as calculated above

D – fuel economy of 350 CID carbureted engine, uncontrolled emissions

From the preceding analysis, C/D = 1.55

At equal power outputs, the 260 CID turbocharged engine will operate at fuel-air ratios slightly leaner than the 378 CID naturally-aspirated diesel, and hence it would be expected that the gaseous emissions levels will be lower. Since the latter engine is expected to meet the 0.41-3.4-2.0 emission standard, the ratio A/C is set at 1.0. From an earlier discussion, the ratio D/B - 1.09. The expected fuel economy improvement for the 260 CID turbocharged diesel meeting the 0.4-3.4-2.0 emission standard, in comparison to the 350 CID carbureted engine meeting 1973 emission standards is

A/B =(1.55)(1.00)(1.09) = 1.69, based upon miles per gallon of fuel. Since fuels are different (diesel vs gasoline), a comparison on the basis of fuel Btu value is A/B = 1.57.

Evaluation

378 CID Naturally-Aspirated Diesel

General Engine Description—The engine for which this evaluation is made is a 378 CID naturally-aspirated diesel, using a swirl-type precombustion chamber. The compression ratio is in the vicinity of 17, and glowplugs installed in the prechambers are required for cold start. Fuel injection is accomplished by an engine-driven "jerk" pump with individual injectors for each cylinder.

The use of a precombustion chamber is based upon the reduced sensitivity of this type of combustion chamber to the quality of the fuel spray which eases somewhat the precision requirements for fuel pump and injector performance. At this time, evidence available within the industry suggests that a certain level of emissions can be obtained easier with the precombustion chamber engine than with the direct-injection open-chamber arrangement. However, the prechamber engine pays a slight penalty in fuel economy due to increased heat and friction losses, and cylinder head design is somewhat more complicated than the open-chamber engine.

Reduction in Fuel Consumption—The naturally-aspirated diesel will increase the fuel economy of an automobile above that of a 1973 conventional carbureted engine, based upon equal vehicle weight and performance. Using a LA-4 cycle combined with steady road load driving, it is estimated that this improvement is 35 percent on a mpg basis and 26 percent on a Btu basis. This value is based upon the diesel meeting the 0.41-3.4-2.0 emission standard with the carbureted engine meeting the 1973 standards.

Performance—The diesel engine displacement of 378 cu in. has been chosen to ensure that the engine will produce the same maximum power as a 350 CID carbureted engine. In the same weight vehicle, the acceleration and top speed performance of the vehicle will be the same with either engine.

Emissions—The fuel economy and performance of the diesel engine as described here can be obtained while meeting the 0.41-3.4-2.0 emission standards for HC, CO, and NO_X.

It is well within the present state-of-art to limit the exhaust smoke to the barely visible level. At present, there are no standards for maximum mass particulate emission rate for automobiles. However, it should be strongly emphasized that if such standards are set in the future, the diesel engine will lose a good deal of its present attractiveness. Table 25 shows typical diesel engine mass particulate emission rates. Carbureted engines produce 1/2 to 1/3 this amount with leaded gasoline, and about 1/10 to 1/20 this amount with unleaded gasoline. Note that the particulate rates for the diesel were measured under a condition where there was no visible smoke. The diesel emissions are composed mainly of carbon particles formed in the combustion process, and to reduce or eliminate this material from the exhaust is a difficult undertaking.

TABLE 25. PARTICULATE EMISSIONS–MERCEDES 220D DIESEL ENGINE[60]

Sample flow (cfm)	Filter type	Mass (gpm)
4	Fiberglass	0.66
1	Fiberglass	0.80
1	Fiberglass	0.75
1	Anderson + Millipore	0.73
	Average	0.73

[60]Springer, "The Low Emission Car for 1975–Enter the Diesel", Paper No. 739133, 8th Annual Intersociety Energy Conversion Engineering Conference, 1973.

The characteristic diesel odor undoubtedly presents a problem in the wide spread use of automotive diesels. The writers of this report are, however, optimistic about the possibility of significantly reducing odor levels. The solution to this problem is made more difficult by the difficulty of odor measurement and the generally poor understanding of odor formation mechanisms. As understanding improves, it would be expected that gains will be made in this direction. It is not expected, however, that diesel odor will be eliminated, although levels may be reduced to levels acceptable for dense urban concentrations of diesel vehicles.

It is concluded that the naturally-aspirated diesel is satisfactory with respect to HC, CO, NO_X and visible smoke and that the odor levels can be reduced to an acceptable value. Possible future standards on mass particulates may offer considerable difficulty.

Noise—Noise levels of the naturally-aspirated diesel engine are acceptable as far as hearing damage criteria and standards are concerned. Further noise considerations will be discussed under the subsection "Consumer Acceptance."

Weight and Size—The weight of present naturally-aspirated diesels is about twice that of a carbureted engine of equal power, on the average. The size is about 40 to 60 percent greater. This is a strong disadvantage since the additional engine weight plus the weight of necessary added vehicle structure for engine support could be as much as 600 to 900 lb. If this weight penalty must be accepted, it obviously puts the previous assumptions concerning fuel economy (equal vehicle weights for carbureted and diesel engines) into jeopardy, since it is doubtful that the additional engine weight can be offset by an overall reduction in vehicle weight while maintaining equal passenger and luggage space and handling requirements. With this magnitude of weight penalty, it is obvious that a complete vehicle redesign is mandatory with a successful outcome (successful in the sense that all criteria are met) somewhat in doubt.

Weight reductions in the diesel engine are certainly possible. The diesel has never been subjected to the exhaustive design studies for automobile uses that have been applied to the carbureted engine. Such design work will result in weight reduction through better understanding of stress patterns and more efficient use of load carrying structure. Stronger materials can be used with cost penalties. Better control over the initial phase of combustion can reduce peak pressures to some extent. Variable comparison ratio could provide the high compression ratios necessary for starting but reduce the compression ratio used for engine running.

Engine structural failures are usually of the fatigue-type caused by a large number of repeated stress cycles, even though the maximum stress in each cycle is well below the fracture strength of the material. The maximum cycle stress is a function of the engine load. All engine designs are to a large extent empirical in this respect, since a particular engine in different applications will accumulate a different spectrum of stress level-number of cycles, and an accurate analytical analysis of these factors is very difficult. Thus engine structural design is to some extent a matter of evolution requiring trial and error with a good deal of operating experience under consumer conditions. With an (assumed) committment to diesel engines, it would be expected that the weight of the engine could be reduced further for this reason.

It is estimated that with a concerted effort, the weight of the naturally-aspirated diesel engine can be reduced to a value 40 percent greater than a carbureted engine of equal power output. It should be possible to reduce the engine box volume to 20 percent greater than the equal-powered carbureted engine.

However, the naturally-aspirated diesel will remain a relatively large and heavy engine, and this factor must count against it.

Reliability—The diesel engines used in truck and bus service have a good record for long life and reliability. Present-day diesel engines in automobiles have, in general, acceptable reliability and maintainability.

The carburetor and spark-ignition system of the carbureted engine are replaced by a fuel injection system. Although the fuel injection system is a close tolerance, precision device, it apparently shows reliability equal or better than the replaced components.

The absence of the proliferation of emission control devices now required for the carbureted engine should increase the probability that emission levels are maintained at the zero-mileage level.

It is concluded that with a modest amount of familiarization on the part of service technicians, the diesel engine will demonstrate reliability at least equal, and perhaps better, than the conventional carbureted engine.

Safety—The use of diesel fuel, with its reduced volatility, should decrease to some extent the collision fire hazard experience with gasoline.

Fuel—The diesel engine is primarily sensitive to the "ignition delay" of the injected charge of fuel, i.e., the duration between the start of injection and the commencement of combustion. Excessive ignition delay leads to harsh combustion with high pressure rise rates, excessive engine noise, and structural stress. Diesel fuel has a short ignition delay. Gasoline on the other hand, has a long ignition delay and is a very unsatisfactory fuel for conventional diesel engines. For a given fuel, ignition delay can be decreased by an increase in compression ratio. In fact, diesel engines with very high compression ratios have been used as multifuel engines capable of using a wide range of fuels, from diesel fuel to gasoline. The MAN combustion system is another method by which a diesel engine can be made to burn fuels of poor ignition quality. In this system, the fuel is sprayed onto the wall of a cavity in the piston crown and the evaporation of the fuel from the hot piston surface controls burning rates and reduces pressure rise rates.

In general practice to date, the development of multifuel diesel engines has received relatively little attention and diesel fuel is the primary energy source. However, the ability to use a wider range of fuel types may assume new importance, and it is believed that this characteristic of the diesel will be emphasized more in future designs.

Several problems of using fuels of poor ignition quality in diesel engines should be noted. First, of course, is the harsh combustion and the degree to which this is minimized is a measure of the success of the multifuel design. Fuel economy is not affected in a successful design. Since gasoline is less dense than diesel fuel, provisions must be made to increase the fuel pump (volume) delivery rate when switching from diesel fuel to gasoline, otherwise a power loss will be sustained. Gasoline is an extremely poor lubricant, and for this reason some types of conventional diesel fuel pumps fail when pumping gasoline. Fuel pumps with their own lubricating system are a solution to this problem.

Accessories and Controls—The diesel engine, with a compression ratio higher than the carbureted engine, requires a larger starter motor, increased battery capacity, and possibly increased generator capacity.

The close tolerance of the fuel injection pump and injector nozzles demand clean fuel, and adequate fuel filtration must be provided.

The absence of air throttling has some effect on accessories. Automatic transmission shift signals provided by predetermined manifold vacuum levels in carbureted engines must be provided by other means. A possibility is the position of the fuel delivery control rod on the injection pump. The average airflow rate through the engine is greater in the diesel than in the carbureted engine, and the capacity of inlet air cleaners should be slightly increased. Inlet air heating to promote fuel vaporization is not required. Carburetor icing is eliminated.

The precise effect on the required capacity of the cooling system is difficult to assess. Average combustion gas temperature will be decreased due to the lean mixtures of the diesel, but airflow rates through the engine are higher. The first effect tends to reduce heat losses to the coolant, and the second tends to increase these losses. In any event, any change in cooling requirements will be small, and it is sufficient to note that the diesel engine cooling system will be essentially the same as that for the carbureted engine.

Cost—The heavier weight of the diesel will increase the cost of the basic engine over that of the carbureted engine. It is generally conceded that the cost of the diesel fuel system and starting aids are greater than the cost of the carbureted engine's ignition system and carburetor. On the other hand, the cost of emission controls found on the 1973 automobile are, in the main, not applicable to the diesel. The diesel will require additional noise absorption material in the engine compartment. Weight reduction efforts may lead to the use of more expensive materials.

Carbureted engines are presently manufactured in very large quantities, and the cost of these engines is from \$3 to \$5 per rated horsepower. Diesel engines designed for trucks and buses cost perhaps six times this amount, and it is estimated that present-day diesel engines designed for automobile use cost from \$6 to \$15 per rated horsepower. Much of the disparity between diesel and carbureted engines is due to the difference in volume of production.

However, assuming equal production quantities, the diesel cost will be higher due to the factors already mentioned. It is estimated that this increment of cost will be 30 percent. For the engine under consideration, the incremental cost will be from \$160 to \$270.

Consumer Acceptance—Consumer acceptance of the diesel engine is influenced by the following factors:

- Fuel cost
- Vehicle acquisition cost
- Exhaust odor
- Engine noise
- Vehicle handling
- Vehicle performance
- Subjective considerations, such as "image", appearance, vehicle styling as influence by engine size and weight, etc.

The diesel engine has been chosen in this study to provide the same performance as the carbureted engine. It has been assumed that the vehicle can be designed to provide equivalent handling qualities and that passenger space is not changed. It is now assumed that the subjective factors can be made equivalent to those of conventionally-powered automobiles, and it is recognized that these factors are of major importance to consumer acceptance.

Added vehicle acquisition cost is not expected to be a significant detriment to consumer acceptance in view of increasing fuel costs.

Engine noise also will not appreciably reduce consumer acceptance if it is presupposed that the noise levels are controlled by the best presently available techniques. The small conventional automobiles are now relatively noisy compared to full-size autos, and it is expected that the diesel noise level can be maintained at a level comparable with present compacts.

The effect of diesel exhaust odor upon consumer acceptance is difficult to evaluate. We tend to believe that it will not affect an individual's decision to own a diesel automobile unless the concentration of such autos is very high and odor has not been adequately controlled so that diesel odor becomes a widespread nuisance.

With these assumptions, it is concluded that customer acceptance for the naturally-aspirated diesel engine would be good.

Demonstration by 1976—The demonstration of a naturally-aspirated diesel engine in a full-size vehicle with power comparable to the carbureted engine version, by 1976, is feasible. However, it would not be feasible to expect that the problems of engine weight, diesel odor, noise, and engine packaging would be completely solved by this time. Of these, the problems of engine weight and odor are considered to be the most difficult and constitute the factors with the highest risk.

Time Required for Implementation—The primary problems of engine weight and exhaust odor might be expected to be greatly reduced by 2 yr of concerted effort. In that event, production of full-size vehicles powered by naturally-aspirated diesel engines could begin in 1980.

260 CID Turbocharged Diesel

General Engine Description—This engine employs a swirl-type combustion chamber, with a compression ratio of about 18 and glowplug starting aid. Jerk pump injection is used. A turbocharger contributes the major equipment difference between this engine and the 378 CID naturally-aspirated engine previously discussed.

Reduction in Fuel Consumption—Based upon a 4300 lb vehicle weight and 0.41-3.4-2.0 exhaust emission levels, the turbocharged diesel should provide a 69-percent increase in fuel economy over that of a 1973 carbureted engine in a vehicle of equal weight, based upon mpg and a 57-percent improvement based upon fuel heating value.

Performance—The maximum power of the turbocharged diesel is equal to that of a 350 CID carbureted engine. However, the inertia of the turbocharger wheel may reduce the response of the engine and penalize acceleration performance even though the maximum speed of the vehicle is unchanged.

Emissions—Since the naturally-aspirated engine has a demonstrated capability to meet the 0.41-3.4-2.0 standards, and since the effect of turbocharging on emissions is generally favorable, it was assumed that the gaseous emissions of the turbocharged diesel meet the 0.41-3.4-2.0 standard. Exhaust smoke is in the barely visible range. Exhaust odor is less than in the naturally-aspirated diesel, but with the present knowledge of engine design still presents a problem. Mass particulate emissions are also less than the naturally-aspirated diesel, but are still much higher than in carbureted engines. If mass particulate emission standards are set based upon carbureted engine particulate emission levels, the ability of the diesel to meet such standards is very doubtful.

Noise—Due to the increased temperature of the compressed charge at top center, reductions in ignition delay have been experienced after turbocharging with consequent reduced combustion noise. The turbocharger itself is a source of high frequency noise, but this noise is more easily attenuated by sound absorption material than the combustion noise. With suitable noise attenuation design, the turbocharged diesel should have an overall noise level about equal to that of the naturally-aspirated diesel but greater than that of the carbureted engine.

Weight and Size—The weight of the turbocharged diesel, with present design knowledge, will be about 40 percent greater than a carbureted engine of the same maximum power. Engine box volumes will be approximately equal. With design studies, consumer operating experience and perhaps material changes, it should be possible to decrease the weight of the turbocharged diesel to about 10 percent greater than the carbureted engine.

Reliability—Even with the addition of a turbocharger, the high reliability of these devices indicates that the overall reliability of the turbocharged diesel will be about the same as that of the naturally-aspirated version, and equal or better than that of the carbureted engine.

Safety—The addition of a turbocharger has no appreciable affect on the hazards posed by the engine. The use of less volatile diesel fuel should improve overall vehicle safety.

Fuel—All of the fuel considerations made previously for the naturally-aspirated diesel also apply to the turbocharged version.

Accessories and Controls—Again, earlier comments on the naturally-aspirated diesel apply to the turbocharged engine.

Cost—As compared to the naturally-aspirated diesel of the same power, the cost of the turbocharged diesel will benefit from the reduced engine weight and will be penalized by the cost of the turbocharger. It is estimated that the result will be a higher cost for the turbocharged version with an incremental cost increase over the carbureted engine of equal power of from \$200 to \$300.

Consumer Acceptance—The increase in acquisition cost for the turbocharged diesel is expected to be acceptable in view of the fuel cost savings.

As earlier discussed with regard to the naturally-aspirated diesel, engine noise and exhaust odor are not expected to be a serious detriment to consumer acceptance, unless the latter becomes a widespread problem.

The owner of the turbocharged diesel powered vehicle will probably experience reduced acceleration performance due to turbocharger lag. This acceleration degradation is not expected to be serious but may reduce consumer acceptance.

Demonstration by 1976—The demonstration of the turbocharged diesel in a vehicle by 1976 is feasible. Because of the reduced problem of engine size and weight (compared to the naturally-aspirated diesel), the risk of this undertaking is less than that of the naturally-aspirated version. It is also considered to be more worthwhile from a technical viewpoint since the effort would provide a larger amount of now unavailable data.

Time Required for Implementation—Engine weight and exhaust odor probably must be reduced before production could be anticipated. With 2 yr of effort devoted to these activities, and with normal lead time for production, production could begin in 1980.

17. DRIVE TRAINS

Current Practice

Because of the characteristic operation of an internal combustion engine, it must be kept running when the vehicle is brought to a stop and must have a means of amplifying its torque capability at low engine speeds when accelerating. The engine is normally disconnected from the driving wheels when the vehicle is stopped by the clutch or by shearing of fluid in a hydraulic coupling or torque converter. Gear reductions in the transmission and driving axle provide the necessary torque multiplication of the low engine torque at low speeds for accelerating the vehicle.

Except for some developmental types, transmissions in use on passenger cars today are three- or four-speed, manual shift, sliding gear types utilizing a mechanical, friction clutch, or three- or four-speed, planetary gear, automatic shift, hydraulic torque converter or hydraulic coupling types.

Literally hundreds of designs of transmissions have been patented and thousands have been conceived, this being perhaps the most prolific area of automotive innovation and invention. Two of the more recent of these innovative types, representative of many, were considered for this study because they have progressed to the vehicle application stage and have been or will shortly be tested in normal road operating conditions.

One of these is a traction drive (Tracor) in which a wide range of drive ratios can be obtained with continuous rather than stepped variation by varying the radii of the points of contact on power transmitting rollers between the driving and driven elements of the mechanism. Traction drive, "infinitely variable" transmissions are not new, dating back to as early as 1903 in the Cartercar (Reference 62), but early versions fell into disfavor because of slippage and wear of the drive elements when engine power increased to meet user demands for better acceleration and performance. The transmission has been periodically reexamined due to analytical (and demonstrated) fuel economy improvements; however, development problems and costs have always shelved this type of transmission in favor of the present planetary gearset torque converter transmissions. Improved materials and better design of the drive elements in this latest development are credited by the designer with providing low slip, high efficiency, and satisfactory durability in configurations up to 150 hp.

The other (Orshanksy) is also a continuously-variable type of transmission embodying two power transmitting paths, one hydraulic and the other mechanical. The hydraulic elements, positive displacement pump, and motor whose functions are interchangeable, provide a controllable, continuous variation between the stepped ranges of the planetary gear mechanical elements to produce a smooth torque multiplication changeover the full output range.

Both of these developmental types of transmissions have been designed to cover a torque multiplication range from some 5:1 down to 0.5:1, thus providing high starting torque for initial acceleration demands and an ultimate overdrive range for the higher speed cruising range.

The selection of drive training gear ratios for a particular vehicle configuration must provide an optimum balance between the power required to drive the vehicle and the power output characteristics of the engine to be used. Obviously, all three of these factors, vehicle, engine, and drive train gear reduction, are interdependent and all may be considered as variables in the overall performance equation.

[62] Ellis, J. R., "Performance Prediction–A Comparison of Various Methods of Estimating the Performance of a Vehicle," Automobile Engineer, Mar 1958.

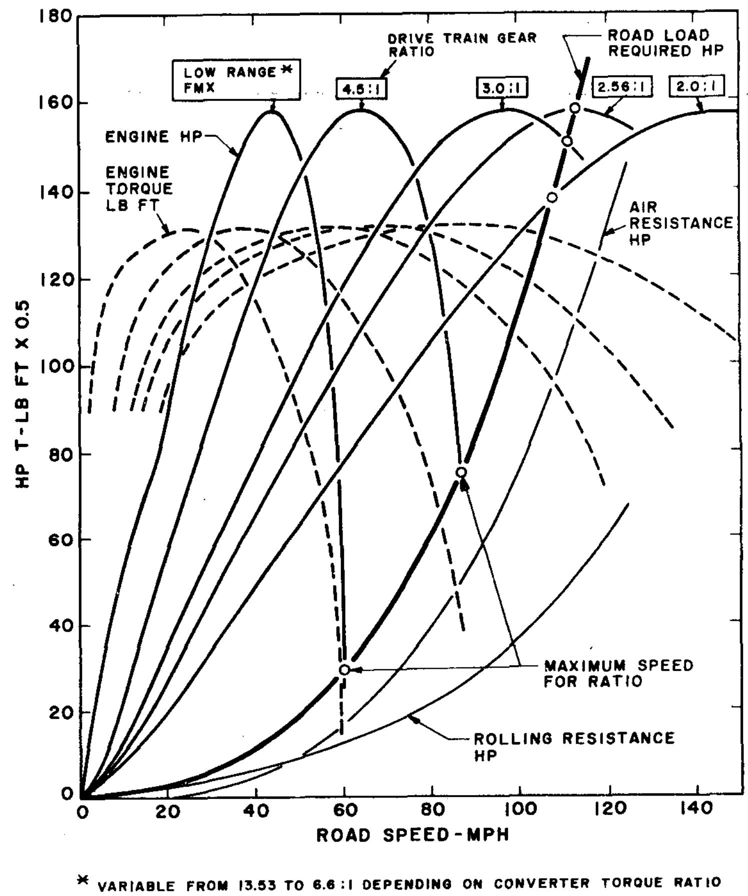
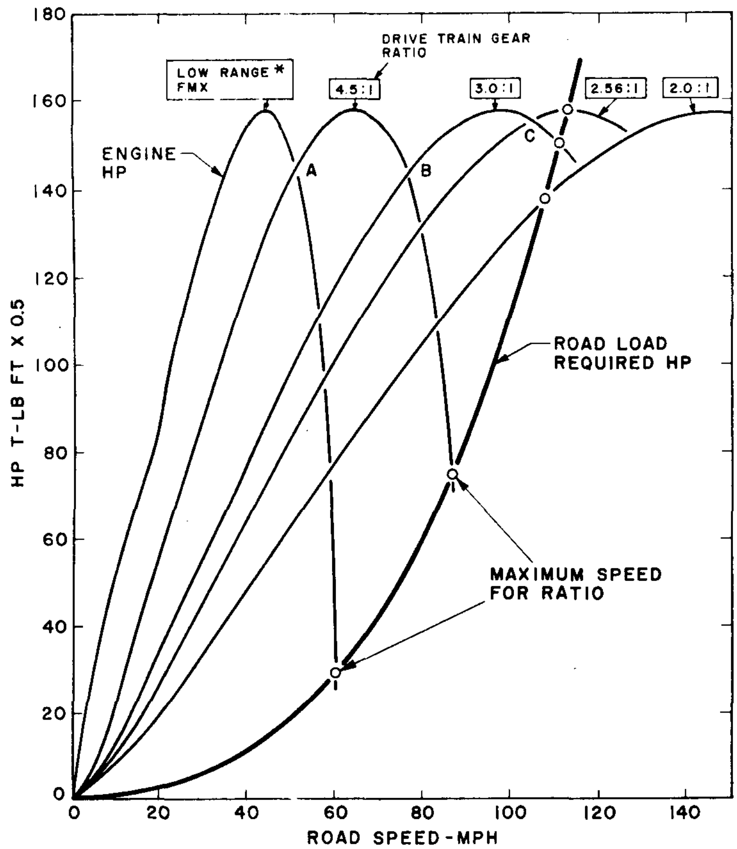


FIGURE 81. VEHICLE-ENGINE PERFORMANCE WITH VARIOUS DRIVE TRAIN GEAR RATIOS

To illustrate this relationship, performance characteristics for several drive train gear ratios for a given engine/vehicle configuration are shown in the attached Figures 81-82. The ratios indicate the overall reduction between the engine and the driving wheels and are achieved by the combined reduction ratios of the transmission, which may also include a torque converter, and the axle gearing. The power required to drive the vehicle under road load conditions, with no grade or ambient wind velocity, is plotted for a standard-size vehicle with two passengers and three-quarters of a tank of fuel. The "wide-open-throttle" engine power and torque characteristics are plotted as separate curves for each selected gear ratio in terms of the vehicle speed. The intersection of each engine power curve with the road load power-required curve defines the maximum vehicle speed obtainable for that gear ratio. Further, at any given speed, the difference between the engine power curve and the road load power-required curve defines the power available for acceleration or grade climbing. Maximum acceleration will be obtained at the peak torque point for each ratio.

Some characteristic performance aspects of these gear combinations should be considered.

(1) Assuming maximum performance in low gear is desired (the automatic transmission is held in low range), the engine power is depicted in the left-hand curve and shows for



* VARIABLE FROM 13.53 TO 6.6:1 DEPENDING ON CONVERTER TORQUE RATIO AT VARIOUS SPEEDS

FIGURE 82. VEHICLE-ENGINE PERFORMANCE WITH VARIOUS DRIVE TRAIN GEAR RATIOS

speeds up to 52 mph., the greatest availability of power for acceleration of any of the gear ratios indicated. The power curve peaks at 44 mph but falls off rapidly thereafter. The vehicle would reach its maximum speed at 60 mph at an engine speed of 5200 rpm. In this case, which represents an actual vehicle configuration, the manufacturer specifies a maximum low-range speed limit of 40 to 52 mph. This is seen to include the peak point of the engine power curve and limits overspeeding of the engine.

(2) The other cases represent arbitrarily selected gear ratios to illustrate the characteristic performance resulting when the peak horsepower point is before, at and beyond the intersection with the road load power-required curve.

- The 4.5:1 ratio curve, e.g., representative of a second-gear transmission ratio of 1.64 with 2.75:1 axle ratio, shows considerable power available for acceleration up to 65 mph and a peak speed of 87 mph at 5100 rpm of the engine. Because of the difficulties of providing satisfactory life, axle ratios below and 2.5:1 are seldom used and should not be considered for normal passenger cars.

- The 3.0:1 ratio curve might represent an axle ratio alone, with the transmission in high gear (1.0:1) or it might be considered as a 1.20:1 third gear in a four-speed transmission with a 2.50:1 axle ratio. This ratio approaches the typical configuration generally considered to be optimum for an engine/vehicle match, wherein the peak vehicle speed is obtained slightly after the peak horsepower point. Good acceleration power is available in the normal driving range of 60 to 18 mph with relatively good acceleration even up to almost 100 mph. Peak speed is reached at 111 mph at an engine speed of 4370 rpm which is not considered excessive as would be those described above for the low gear and 4.5:1 ratio peak speeds.

- The 2.56:1 ratio was selected to bring the maximum speed point to the peak power output point of the engine. It could be considered also in the form of a 0.85:1 overdrive system in conjunction with a 3.00:1 axle ratio. This represents the maximum speed capability for this engine/vehicle combination. It will be noted, however, that although this is only marginally higher (113 versus 111 mph), the power available for acceleration at midrange speeds is considerably lower, hence passing and hill climbing performance would be measurably inferior. Fuel economy, on the other hand, would be improved because of the higher load factor (power available/power required). Used in its overdrive configuration, the 2.56:1 ratio might be selected above 45 to 50 mph with a "kick down" to lower direct drive ratio, such as represented by the 3.0:1 example, for improved passing or hill climbing performance.

- The 2.0:1 ratio system, which could be obtained as a 0.73:1 overdrive with the 2.75:1 axle ratio, indicates an overmatch of the vehicle for the engine power available. The maximum vehicle speed of 108 mph, reached at an engine speed of 2800 rpm, is not very different from the two previous cases, however, passing performance, i.e., power available for acceleration in the normal driving range, is much inferior. Again, as in the above paragraph, in the normal overdrive mode, improved fuel economy would be expected because of the higher power factor. On the other hand, similar or perhaps even better performance might be expected from a considerably smaller engine of similar characteristics, having a peak power output of about 140 hp at 3600 rpm with a 2.8 or 3.0:1 drive train ratio, particularly if coupled with a 0.9:1 overdrive.

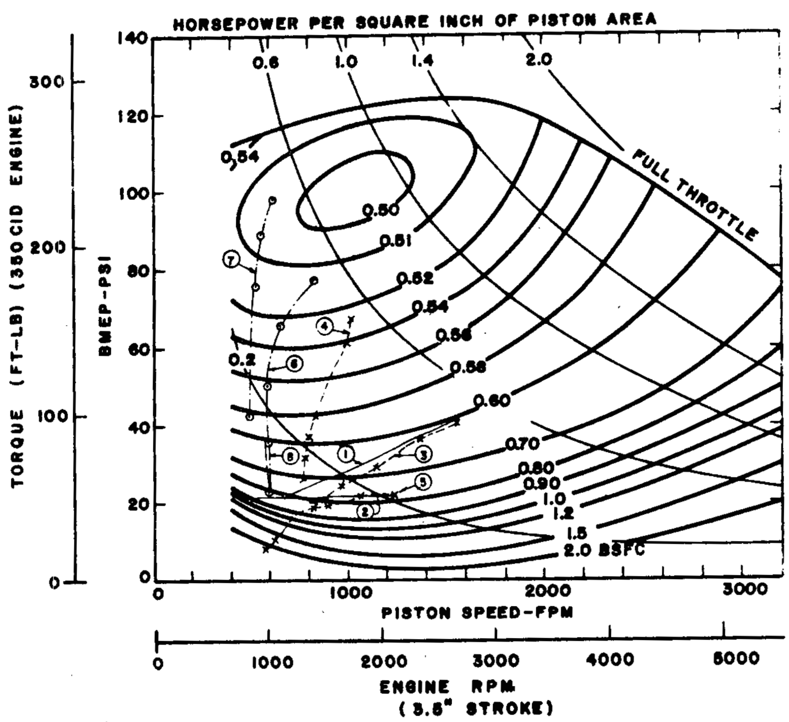
It will be further noted that, if maximum speed capability of 90 to 100 mph is accepted as a design limitation, engines of 95 to 120 hp can provide adequate acceleration, passing and hill climbing performance with drive train ratios of 3.0:3.3:1, even in vehicles of 4500 lb. Because of higher loading factors, even greater fuel economy than in the previously discussed configurations would be expected.

(3) The curves in Figure 82 also illustrate the overall performance through the gears of a transmission, although the ratios here are not precisely the proper progression which would normally be chosen for a given transmission. With such a transmission, with shift points at A and B for a three-speed of A, B, and C for a four-speed configuration, it is seen that acceleration power comprises all of the area between the combined available power curves and the road load power-required curve. In a specific case, the shift points would perhaps be moved somewhat to the left, closer to the peak power point for each gear ratio or the peak torque point, if maximum tractive force is desired.

Evaluation

Most vehicle operation is essentially at constant speeds in which the "road load" performance of the engine with its characteristic fuel consumption is paramount in the total fuel consumption picture.

The current engine/transmission/axle configuration for most standard size vehicles tends to result in a relatively low "power-factor" during most operation in favor of a large reserve of power available for acceleration, particularly for safe passing at higher road speeds. At low rates of acceleration and at road loads this results in engine performance in the higher specific fuel consumption regions as illustrated in Figure 83.



1. ROAD LOAD MANUAL TRANSMISSION
2. LA 4 MANUAL TRANSMISSION
3. ROAD LOAD FMX TRANSMISSION
4. LA 4 FMX TRANSMISSION (ACCELERATING)
5. LA 4 FMX TRANSMISSION (CONSTANT SPEED)
6. ROAD LOAD CVD TRANSMISSION
7. LA 4 CVD TRANSMISSION (ACCELERATING)
8. LA4 CVD TRANSMISSION (CONSTANT SPEED)

FIGURE 83. PERFORMANCE MAP OF CARBURETED SPARK IGNITION ENGINE

As an illustration of the effects on fuel consumption of a variety of transmission concepts, estimates were made for a full-size reference passenger car, having a 350 CID engine and 2.75:1 axle ratio (4500 lb inertia weight–19 hp at 50 mph). Three transmission variations were considered, a three-speed manual shift transmission, an automatic, torque converter transmission, and a development type, continuously variable transmission, as well as other drive train configurations including overdrive and a converter lock-out system for automatic transmissions. Finally, an optimum combination of these configurations, which could be considered as readily available, was made for comparison. (See Table 26.)

TABLE 26. DRIVE TRAIN COMPARIONS

Type	First gear		Second gear		Third gear		Fourth gear	
	Ratio	Overall ratio	Ratio	Overall ratio	Ratio	Overall ratio	Ratio	Overall ratio
Manual transmission	2.5:1	6.88:1	1.5:1	4.125:1	1.0:1	2.75:1	0.7:1	1.92:1
Automatic transmission	2.4:1	13.5[(a)]-6.60:1	1.47:1	4.04:1	1.0:1	2.75:1		
Continuously variable	5.0:1 variable to 0.5:1; overall 13.75:1 to 1.375:1							
Optimum drive train[(b)]	2.6:1	8.3:1	1.6:1	5.1:1	1.0:1	3.2:1	0.7:1	2.24:1

(a)Converter ratio at stall–2.05:1.
(b)302 CID engine, 3.2:1 axle.

The selection of the optimum vehicle drive train components is, in reality, only an initial arbitrary estimate of an optimum configuration based on preliminary calculations of range of available acceleration performance, economy, and speed. In a normal design situation, such preliminary studies would covet a range of engine and transmission characteristics and would ultimately be built up on vehicles and run to determine the limits of road performance and operator acceptance. Such optimization must be considered as an iterative process involving many factors other than speed, acceleration performance, and fuel economy alone.

TABLE 27. ESTIMATED FUEL ECONOMY IMPROVEMENT WITHOUT EMISSION CONSIDERATIONS

Drive train type	LA-4 (%)	Road load (%)	Composite (%)
Manual	13.5	9	11.3
Continuously variable (Tracor CVD)	8.9	41	21.1
Lock-up clutch (on reference automatic)	1	5	3
Overdrive and lock-up (0.70 to 1)	2.3	9	5.3
Optimum design (for economy)	13.3	17	15.1
Hydro mechanical[(a)] (Orshansky)	—	17	—

(a) Data from Reference 63.

Comparative fuel consumption was estimated for the urban Driving Cycle and for Road Load Operation. (Table 27)

The ability of the CVD transmission to maintain engine operation at higher power factor, i.e., to achieve relatively high vehicle speed at low engine speeds and higher torque levels, is evident at the 5.00:1 maximum ratio selected here. The advantage in fuel economy is more pronounced at the selected maximum engine speeds of 850 to 900 rpm than if the engine speed is allowed to increase. This low speed is not recommended, however, without considerable redesign of the engine to provide improved low-speed mixture distribution. Changes in valve overlap, manifolding, and carburetor flow characteristics can be anticipated. These are not insurmountable problems, but they do involve changes in engines which have been designed for high performance at high-speed rather than low-speed.

[63]Orshansky Transmission Corp., "Hydromechanical Passenger Car Transmission," Report No. 403, May 1973.

It also suggests the use of a smaller engine with relatively lower maximum output performance, since the low and medium road speed performance would not be materially affected.

The performance of the manual transmission reflects the better efficiency of this type of drive over the torque converter (baseline) system. It could provide improved fuel economy by decreasing the rear axle ratio further and providing four gear ratios in the transmission or, by what amounts to the same effect, using an overdrive. This, again in combination with a smaller engine which could then operate in its lower speed, higher power factor mode, would provide better fuel economy than shown in the above comparison.

Both the manual and CVD transmissions require the inclusion of a clutch to allow engine operation when the vehicle is stopped. The automatic transmission does not, since it slips readily at engine idle speeds. On the other hand because it still slips to some extent at high engine speeds, lock-up clutches have been considered for the automatic transmission, to be engaged when the converter coupling point (1:1 torque multiplication) is reached. Since this point is normally in the region of 1600 to 2000 rpm (engine), the lock-up clutch shows little improvement in fuel economy even under road load operating conditions, until operating speeds are above 45 to 50 mph. At highway speeds, above 50 mph, the torque converter efficiency would typically be about 90 percent for vehicles with rear axle ratios below 3.0:1. This factor must be combined with another efficiency, on the order of 95 percent, associated with the gears and other driveline components. The use of a lock-up clutch effectively eliminates the inefficiency due to the torque converter, but the other components must still be considered.

It should be noted that the percentage change in overall driveline efficiency may not be translated directly into the context of fuel economy. If the torque converter slip is reduced, then a lower gross engine horsepower is required, and the engine speed is reduced. The brake specific fuel consumption at which the lower horsepower is obtained, and the resulting fuel economy, depends upon the details of the engine map for the particular vehicle. Furthermore, the fuel economy improvement depends upon the specific characteristics of the torque converter and the operating point at which the lock-up clutch engages.

In the example selected for this study, it was assumed that the lock-up clutch engaged at the operating point where the torque ratio was unity. Large improvements in fuel economy, on the order of 15 to 20 percents, can be predicted for a narrow range of speeds immediately above this lock-up point, but the percentage improvement diminishes as the vehicle speed increases. It is probable that the overall improvement could be enhanced by assuming either different torque converter characteristics or a different lock-up point.

It is possible to envision a lock-up clutch that would operate in all gears and would engage at some preset speed ratio during operation. In the limiting case, such a system would approach the conventional manual transmission, but a considerably greater quantity of hardware would be involved. For example, both a clutch and a torque converter would be necessary, and the shifting control system would be required to assume the function and the logic of clutch operation. Such transmissions are used in heavy-duty vehicles, but the adaptation to passenger cars has yet to be effected. Furthermore, if the clutch is used to maximum advantage from a fuel economy standpoint, then serious questions of cost and driveability remain to be resolved.

Other transmission design concepts are of some interest and should be discussed particularly in regard to their potential for improving fuel economy[64,65,66,67]. Among these are variants in torque converter transmissions, four or more speed ratios in transmissions, and fluid couplings.

In general, variants in torque converters affect the torque multiplication performance at low output speeds and are primarily to improve initial acceleration capability of the vehicle[64]. Numerous designs having multistaged turbines and stators can raise the initial torque ratio in the converter and ahead of the transmission mechanical gearing to as high as 3.5:1 as compared with 2.0 to 2.2:1 for the usual passenger car single stage torque converter. Because the torque ratio drops rather quickly to the 1.0:1 level (coupling point) in the range of 1600 to 2000 rpm or less, depending on torque being transmitted, there is little effect on overall fuel consumption since only the acceleration periods are affected. However, such converters generally have lower maximum efficiencies than the less complex single stage converters to that overall fuel consumption will be greater unless converter lock-up devices are employed above the coupling point.

Considerable modification of the torque available for acceleration in starting up from a stopped position can be obtained by changing the hydraulic path diameter of the converter.[68,69,70] This changes the power absorption capacity of the converter pump or impeller, driven by the engine, so that as the diameter is decreased the stall speed of the impeller/turbine combination increases. This increase in engine speed allows the engine to develop greater torque to increase the initial tractive force available for acceleration. As in the multistage turbines, the smaller diameter converter has greater slip and, hence, lower maximum efficiency after reaching the coupling point. A minor reduction in fuel economy will result even though initial acceleration may be better.

The Model T Ford transmission was a mechanical clutch and planetary gear transmission. Modern planetary transmissions could also be operated with a mechanical clutch. Such configurations have recently been used in drag racing (Reference 68) to provide high input torque, "jump" starts. The engine can be run up to high speed to provide a high inertia impact engagement for initial breakaway acceleration. Such operation is extremely hazardous to the drive train, however, and would be completely unacceptable for the normal passenger car driver. Further, unlike the torque converter, no torque multiplication is provided by the clutch. A better solution involves the use of a fluid coupling, and the earliest applications of hydraulic drives were of this type.

Fluid couplings (no torque multiplication) have been used extensively in the past[71], particularly in combination with three- or four-speed planetary transmissions. Their principle advantage is their ability to transmit torque at higher vehicle speeds with relatively small slip, but being able also to disconnect the engine from the drive train (100 percent slip) at low-engine speeds without operator action; i.e., to act as an automatic, hydraulic clutch. Because it is a 1.0:1 torque transmitter, all of the torque multiplication requirements must be incorporated in the transmission and axle mechanical gearing. The transmission gearing and control mechanisms, disc and band clutches become more complex and it was generally for this reason that the fluid coupling transmissions were supplanted by torque converter types incorporating less complex gear trains. Although vehicle operation is

[64] Walker, F. H., "Multiturbine Torque Converters," SAE Paper 359C, 1961.
[65] Greer, J. W. and G. W. Schulz, "A New Ford 3-Speed Automatic Transmission," SAE Paper 660075, Jan 1966.
[66] Fuchs, J., "350 Turbo Hop-Up," Hot Rod, Sep 1972, p 114-5.
[67] Anonymous, "Overdrives for 4-speeds and Automatics," Popular Hotrodding, Jan 1973, p 42-44, 97.
[68] B & M Automotive Products, "1973 Technical Journal and Catalog," 9152 Independence Ave., Chatsworth, Cal. 91311.
[69] Fairbanks Racing Automatics, "Converters by Fairbanks," 336 Elm, Stamford, Conn. 06902.
[70] A-1 Automatic Transmissions, "Catalog," 7239-1/2 Woodley, Van Nuys, Cal. 91406.
[71] Chrysler Corp., "Transmissions," Chrysler Institute of Engineering Graduate School Lecture Notes, August 1957.

more simple and drive train shock loads are damped by the hydraulic coupling, it has little effect on fuel consumption, except possibly a very minor degradation from mechanical clutch or locked drive systems because of slight slippage at cruising speeds.

The use of a fluid coupling or a manual clutch can result in an improvement in idle fuel consumption since the load transmitted during idle (brakes locked) could be considerably reduced. At the idle speeds of present torque converter equipped vehicles, the torque absorption in the converter is about 35 ft-lb. At idle speeds of 700 rpm this represents about 4.7 hp that could be virtually eliminated. The power requirement could also be reduced by decreasing the idle speed; however, emission control of 1973 model automobiles (no aftertreatment) requires that the higher loads be obtained to reduce mass emissions during idle. With the use of catalytic converters, more design flexibility is available in that idle emissions could be allowed to increase and the resulting "dirty" effluent "cleaned up" in the catalyst.

The consumer acceptance of manual clutches is anticipated to be minimal, however, a fluid coupling and a lock-up could be incorporated to substitute for the present converter. On the whole, however, there is considerable latitude available in the design of torque converters. A decrease in the stall torque ratio can be incorporated by a design compromise that will improve the efficiency of the unit as a coupling. On the whole, it appears that redesign of torque converters with some penalty in torque multiplication (loss in performance) can be effected.

Transmissions with greater than three-forward gear ratios are generally used in passenger vehicles with small engines (low power-to-weight ratios) to provide a greater range of torque multiplication for starting acceleration, while avoiding excessive engine speed variations in each individual gear range. They, thus, provide some of the added gear mutliplication by itself, while providing better acceleration performance, will not appreciably affect fuel consumption. If, however, the axle ratio is low, resulting, in effect, in an overdrive mode when the transmission is in direct drive, fuel economics can be obtained without sacrificing passing and hill climbing performance in lower transmission gears.

Typical small car drive train gear ratio configurations, which may be compared with the ratios stated in Table 26 would be:

Gear	MG		VW	
	Transmission	Overall (drive train)	Transmission	Overall (drive train)
First	3.640:1	15.65:1	3.80:1	15.68:1
Second	2.214:1	9.52:1	2.06:1	8.49:1
Third	1.374:1	5.91:1	1.32:1	5.45:1
Fourth	1.000:1	4.30:1	0.89:1	3.67:1
Axle	4.30:1		4.13:1	

Stage of Development

The optimum drive train concept described in the previous section is comprised of engine, transmission, and axle components which are currently available, although perhaps not presently entirely optional as far as the vehicle manufacturer is concerned. The overdrive is marketed as a low production, "after-market" device primarily for drag racing customers to enable them to use their vehicles in normal traffic.

Other drive train concepts than the normal three-speed automatic, torque converter transmission, are less readily obtained. Various diameters of converters to provide stall speed modifications are available from the stock car racing equipment industry. Developmental types of transmissions could be made available within approximately 2 yr in some quantities but would require at least 4 to 5 yr for large scale production for the automotive industry with, perhaps, many problems to be overcome in legal, patent, and contractual areas.

Reliability

Transmission and drive train components now in production are comparatively long lived, highly reliable devices requiring relatively little operator maintenance beyond occasional lubricant level inspection and long-interval lubricant replacement. Some types of automatic, planetary-gear type transmissions are subject to long-interval friction clutch (band) adjustment; and, manual shift, sliding gear transmissions normally require clutch facing replacement at 30 to 50,000 mile intervals.

Other alternative or ancillary elements, such as overdrives or torque converter lock-up clutches, have the same maintenance and reliability characteristics.

Specific details of operating life and reliability of development type transmissions such as the Orshansky hydromechancial and Tracor CVD traction drive are not considered to be substantiated as yet, however, prototype and paper studies conducted by the developers have indicated comparable performance to production transmissions.

It may be stated that the service life of drive train components will be extended if the general public becomes measurably conscious of the problems of fuel economy, since driving practices which achieve best economy avoid excessive acceleration, high speeds, and other high-load operating conditions which adversely affect drive train mortality.

Cost

Costs of concern in transmission and drive train elements relate primarily to initial purchase inasmuch as operating and maintenance costs are essentially insignificant over the life of the vehicle.

The manual shift transmission with a manually operated friction clutch is the least expensive of the configurations studied, however, a detailed breakout of costs of individual components of the complete system as a part of the vehicle base selling price could not be obtained. Detailed costs of the component as replacement parts are obtainable, but these are not representative of initial vehicle costs. It is possible to determine price differentials between the manual transmission costs and costs for other transmission variants. Available costs for overdrives and torque converter transmission variations are not representative of normal passenger vehicle acquisition costs inasmuch as currently available units are in low production, specialty markets. Costs are estimated for development-type units on the basis of their comparative manufacturing complexity in relation to current production types (Table 28).

TABLE 28. COST ESTIMATE FOR DEVELOPMENT TYPE UNITS

Transmission	Estimated Cost in Production
Standard, 3-speed manual	Base
4-speed manual	$200 added
Automated, 3 speed, torque conv.	210-300 added
Overdrive	50 added
Hydro-mechanical, CVD	250 added
Traction CVT	150 added

Safety

The principal influence of the drive train design, with regard to safety, concerns the ability of the vehicle to accelerate and pass expeditiously at highways speeds of

50 to 70 mph. Hence, those drive train designs which provide greater power availability for acceleration in this speed range may be considered safer. Such design, however, may be considered normally to result in increased overall fuel consumption since greater availability of power for acceleration means proportionately less of that power is used for road load (no acceleration) driving. The engine will be operating at a lower power factor, with the characteristically higher specific fuel consumption inherent in such operation.

In the overall goal of improving vehicle safety, much emphasis is placed on those factors which provide occupant protection after an accident has occurred. It is suggested that those factors which assist the driver in avoiding the accident in the first place, such as good acceleration, are even more important in the overall safety picture. It is, thus, extremely important that the ultimate compromise between acceleration and fuel economy not markedly degrade acceleration performance and should not be much below 2 mph/sec for the 50 to 70 mph speed range.

Too low an axle ratio, or the use of an overdrive which results in a too low drive ratio, would reduce this acceleration performance below the requisite level. The overdrive is, however, designed to be disengaged when high acceleration is necessary to avoid such an unsafe passing problem and automatic transmission can be "down shifted" from third to second gear at speeds below 70 mph when high acceleration is desired.

Effects of Transmissions on Engine Exhaust Emission

At a given vehicle operating condition (speed and acceleration), the transmission sets the torque (bmep) and speed of the engine. Further, the means of gear shifting affects transient engine speed and load. Therefore, the transmission has important effects upon the level of exhaust emissions produced by the engine.

In this discussion, changes in emission levels due to transmission effects will be referenced to a conventional three-speed automatic transmission. First, consider a manual shift three-speed transmission. On the operation of this type of transmission, there are more engine speed transients than with the automatic transmission due to the difficulty in coordinating manual operation of clutch, throttle, and shift lever. These speed transients may lead to higher CO and HC emissions because of acceleration-deceleration effects on fuel-air ratio and residual fraction in the combustion chamber. However, if to meet the 0.41-3.4-2.0 emission standard an oxidizing exhaust catalyst is used, it is expected that the possible increase in HC and CO due to manual shifting will be insignificant. For this reason, no strong effects upon exhaust emissions are expected from a change to manual transmission when the emissions are measured on the LA-4 cycle with specified shift points.

The effect upon emission levels from the use of a continuously variable transmission (CVT) are more significant. The purpose of the CVT is to increase the average load at which the engine operates. For this reason it would be expected that levels of NO_X would increase. To meet the 0.41–3.4–2.0 emission standard using a conventional carbureted engine, EGR would necessarily be increased to counteract the increase in NO_X, and this would lead to a fuel consumption increase. A quantitative estimate of the magnitude of this effect is, without actual development test data, little better than an intuitive guess.

The operation of a conventional spark-ignition engine at low speeds and high bmep (~80), as specified by the anticipated operating schedule, will require adjustment of spark timing (retard) to prevent knock, probably readjustment in EGR schedules and, additionally, redesign in carbureted and valve gear system to maximum the performance of the engine transmission system. Data available from independent sources (Ref. 63) indicate small improvements in critical emissions (NO_X) on the order of 10 percent by analysis of the performance schedules of a CVT and a conventional three-speed torque converter transmission with respect to the *same* engine emissions maps. As stated above, design changes for life and performance and emission control are anticipated. In the absence of any good data at all, the authors guess that the ratio of the fuel economy of a vehicle operated with an emission controlled engine and CVT to the fuel economy of the same vehicle with CVT calculated without regard for increased emissions is about 0.90, reflecting our concern for the ability to maintain NO_X control. If emissions of NO_X can be controlled by some device which would allow turning of the engine without regard for the control system, then this factor could be increased to unity.

As a result of emission considerations, it is necessary to modify the economy improvement percentages earlier calculated for new transmission types. Although the fuel map used to obtain the change in fuel consumption is a map for an engine without emission controls, it is assumed that the same percentage improvements will pertain with any engine. However, the change in transmission type will affect the emission level produced by the engine and will require an engine modification (and fuel economy change) if the transmission increases the difficulty of achieving a particular emission standard. To explain the method used to provide fuel economy comparisons accounting for emission controls, the following terms will be defined in the same manner as before:

A – fuel economy of the vehicle with modified drive train meeting the 0.41–3.4–2.0 emission standard

B – fuel economy of the standard vehicle meeting 1973 emission standards

C – fuel economy of the vehicle with modified drive train with uncontrolled emissions

D – fuel economy of the standard vehicle with uncontrolled emissions

E – fuel economy of the standard vehicle meeting the 0.41–3.4–2.0 emission standards

The following equation was then used to compare vehicle fuel economies under different emission standards:

$$\frac{A}{B} = \frac{C}{D} \times \frac{D\,A}{E\,C} \times \frac{E}{D} \times \frac{D}{B}$$

The value of C/D is taken fromTable 27. E/D and D/B are 1/1.15 and 1.09 (see Chapter 8). The ratio (D/E) (A/C) is a measure of the difficulty in making the vehicle with the modified drive train meet the 0.41–3.4–2.0 emission standard, as compared with that of the standard vehicle. Values for this ratio are tabulated below:

Drive train type	Emission factor (D/E) (A/C)
Manual	0.97
Continuously variable	0.90
Lock-up clutch	1.0
Overdrive and lock-up	0.97
Optimum design (for economy)	0.95

A summary of the fuel economy improvements, based upon the various emission standards, are listed below:

From Table 27				
Drive train type	C/D	% Improvement	A/B	% Improvement
Manual	1.113	11.3	1.023	2
Continuously variable	1.211	21.1	1.033	3
Lock-up clutch	1.03	3.0	0.976	−2
Overdrive and lock-up	1.053	5.3	0.968	−3
Optimum design (for economy)	1.153	15.3	1.038	4

Noise

In only one instance in these alternatives for drive train components is there any question of excessive noise generation or any question of essential differences in generated noise levels. This question arises only in the case of the hydromechanical, continuously variable transmission because of the use of positive displacement hydraulic pumps and motors in the liquid drive path. Hydraulic pressures as high as 6000 psi may be generated with attendant noise generation characteristic of such positive displacement hydraulic systems. It is anticipated by the developers that much can be done to attenuate or damp out this vibration and thus eliminate or at least greatly reduce the noise.

It should also be recognized that in the case of the optimum configuration embodying the smaller engine and higher axle ratio, some minor increased noise may result at any given road speed from the higher engine speed.

Performance

As discussed in the introduction to this section, vehicle performance, i.e., acceleration, is determined by the excess of tractive effort available from the engine through the drive train over that force required to keep the vehicle moving at a constant speed. Thus, a transmission which can keep the engine operating as near as possible at its maximum torque point while the vehicle speed increases steadily will produce the maximum acceleration.

Only the truly continuously variable transmissions can meet this requirement precisely. All other types of transmissions can only approximate such a match with the torque converter types next best in performance and four-speed and three-speed, manual shift transmissions in order behind them.

Smaller diameter, high-stall speed torque converters can be adapted to the automatic transmissions to allow the engine to approach its maximum torque capability while the vehicle is standing still, if maximum torque is desired at this point. This is frequently done in drag racing vehicles[68,69,70], however, normal passenger vehicles use considerably lower-stall speed converters with less slip and improved efficiency at the higher vehicle speeds in road load operation, as previously discussed in relation to fuel economy.

Consumer Acceptance

Except for a small, elite class of motoring buffs who want to "shift for themselves," the public prefers automatic transmission systems. The average motorist is neither sufficiently skilled nor does he ever desire or need to use the maximum performance capability of his vehicle, which might be achieved with a manual transmission. Transmission systems for the general public should, therefore, provide the optimum of driver skill and equipment maintenance and servicing.

Although cost considerations may be important from a manufacturer's competitive point of view, they are not particularly important insofar as buyer convenience is concerned. That is, the public willingly pays more for automaticity and avoidance of personal effort.

For these reasons, all of the automatic transmissions, including the development types discussed, can be considered to be essentially equally acceptable. The manual shift transmissions are less acceptable, probably in direct relation to the buyer's concern for economical transportation, unless he is one of the previously mentioned purists.

18. TIRES

Background

Although automobile manufacturers have begun to stress reduction in tire rolling resistance[72], other characteristics have traditionally been considered more important. In the forefront have been treadwear and durability: long mileage to bald; and resistance to tread groove cracking, ply and splice separation, puncture, ozone degradation, atmospheric oxidation, and impact and other road hazards. More recently, antiskid, antihydroplaning, and noise factors and the tire's contribution to vehicle ride, handling, road holding characteristics, and compatibility with vehicle suspension systems have become important. Raw material and manufacturing costs, also, have had strong influence.

Traditionally, the automobile tire has been of bias ply construction in which the carcass is so constructed that the cords of alternate plies of tire fabric cross one another at an angle. A modification makes use of partial plies, or breakers; plies extending only across the tread area, either between full plies (bead to bead), or on top of them, under the tread area. More recently the belted bias play has come into prominence, its distinguishing feature being conventional bias full plies surmounted by circumferential fabric belts under the tread. A still later development is the steel belted radial in which the cords of full plies lie in radial planes through the geometric center of the tire, with a belt ply made of steel wire.

Until World War II, natural rubber was, essentially, the sole elastomer available in sufficient quantities for tire production on the scale required by the automobile population. With that supply cut drastically by the war in the Pacific, the synthetic rubber industry developed, using petroleum as feedstock. Early synthetics suffered from extremely high hysteresis losses and overheating and, probably high rolling resistance, but progress in polymer development and selection, and rubber compounding have combined to yield car tires of essentially 100 percent synthetic with performance nearly equal to that of natural rubber tires, and in some respects, superior to them. Approximately 80 percent of the total rubber used in all manufacturing in 1973 will be synthetic[73].

Current Practice

A majority of original equipment tires on current new cars is of belted bias construction, although radials are standard on luxury and some specialty models. Original equipment tires on the Continental and Capri, for example, are Michelin steel belted radials. Original equipment on the large Mercurys is a domestic steel belted radial. On the remainder of the Mercury line, with the possible exception of the Cougar, original equipment is bias belted. Bias ply, bias belted, and radials are all available in the replacement market, with both the belted bias ply and radial available with either fabric or steel belt[74].

As indicated, although some vehicle manufacturers are emphasizing reduction in tire rolling resistance[72], it has not been a primary goal, and today's production tire represents a balanced combination of optimum qualities. Improvement in one characteristic generally results in degradation of others. Bias, bias belted, and radials are all in production today, and efforts to improve rolling resistance will affect each construction differently, or to a different degree. Some possibilities for improvement are outlined following.

[72] Robert Wilds, Highway Safety Research Institute, Ann Arbor, Michigan 10-9-73.
[73] "Record for 1973: 200 million tires hipped for Autos," Automotive News, 12-3-73.
[74] Telephone Contact, Mickey Smith, Main Lincoln Mercury, San Antonio, Texas, 12-7-73.

Inflation Pressure

Tests and observations document possible improvement in tire rolling resistance with increase in inflation pressure[75,76]. Laboratory investigation indicates that tire rolling resistance decreases with increasing inflation pressure[75,76]. The relationship for a bias ply tire is indicated in Figure 84, for

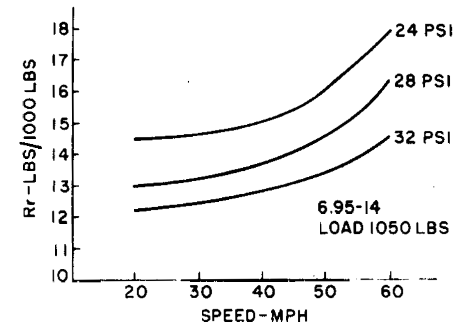


FIGURE 84. BIAS TIRES

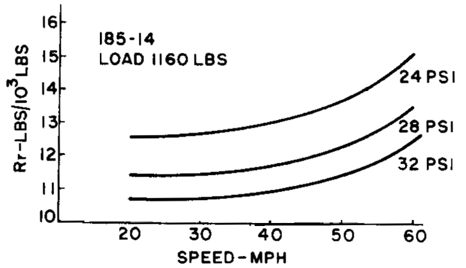


FIGURE 85. RADIAL TIRES

a radial in Figure 85, and for a belted bias tire in Figure 86 for an inflation pressure range of 24 to 32 psi and a speed of 20 to 60 mph for the bias and radial, and for a range of 17 to 60 psi at 30 mph for the belted bias. Figure 87 summarizes the general relationship between inflation pressure and tire rolling resistance, from which can be derived a relationship of about 5 percent change in rolling resistance for each 2 psi change in inflation pressure within the 24 to 32 psi range[77]. On this basis, tire rolling resistance could be decreased as much as 20 percent by increasing inflation pressure from 24 to 32 psi.

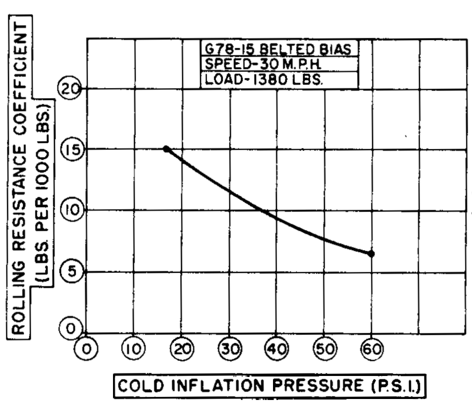


FIGURE 86. BELTED BIAS TIRES

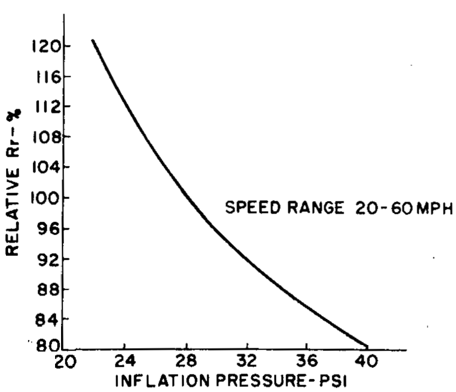


FIGURE 87. GENERAL RELATIONSHIP-INFLATION PRESSURE VS. ROLLING RESISTANCE

Tread Mass

Tire rolling resistance consists of two components, one independent of speed, the other speed dependent which varies as the square of the tire mass per unit tread area[76]. Experimentation has

[75] Elliott, D. R., W. K. Klamp, and W. E. Kramer, "Passenger Tire Power Consumption," Paper No. 710575, SAE Transactions 1971, pp 1885-1898.

[76] Stiehler, R. D., M. N. Steel, G. G. Richy, J. Mandel, and R. H. Hobbs, "Power Loss and Operating Temperature of Tires," Proceedings, International Rubber Conference, November 8-13, 1959. Washington, DC pp 73-83.

[77] Curtiss, W. W., "Low Power Loss Tires," Paper 690108.

shown that by buffing the tread of a new tire to the bottom of the tread grooves, rolling resistance decreases for both bias ply and radial ply tires, Figure 88[78]. For the bias ply, rolling resistance decreases approximately 50 percent at 20 mph and about 70 percent at 70 mph. Rolling resistance of the radial is affected similarly, but not to the same degree, ranging from about 45 percent reduction at 20 mph to about 25 percent at 70 mph. At a given speed, reduction is almost linear with tread removal[77]. For 50 percent worn, rolling resistance of a bias ply tire can be expected to decrease about 25 to 30 percent, and a radial about 10 to 25 percent. Current passenger car tires worn to wear indicator tread depth are approximately 15 percent better for power loss than they were when new[79].

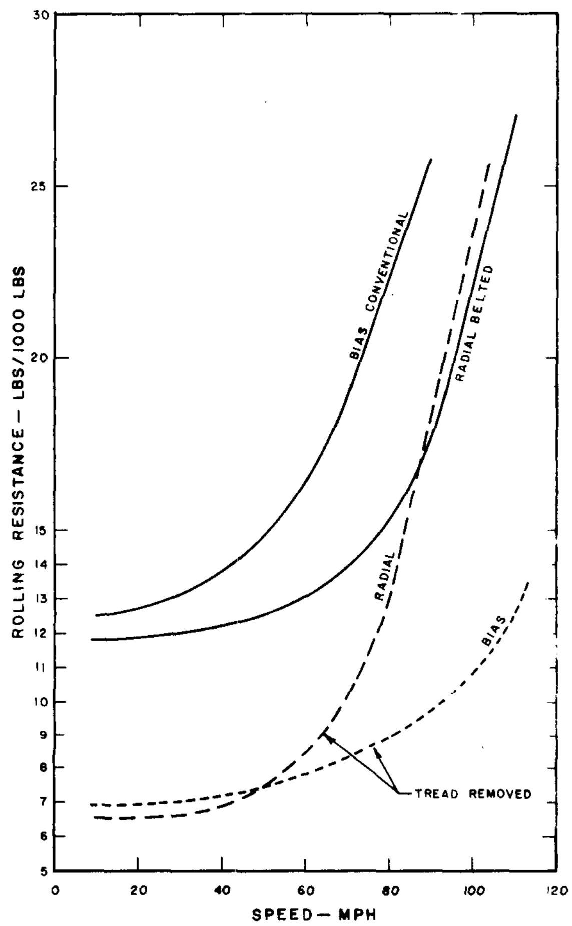


FIGURE 88. EFFECT OF TREAD REMOVAL

Compounding

The rubber component of a tire accounts for 60 percent of the hysteresis loss[76], which translates into rolling resistance and power loss, and offers an opportunity for improvement in rolling resistance by use of low hysteresis compounds, which have high resiliency characteristics. Use of such compounds, however, will normally involve penalties against other performance characteristics; and treadwear, traction, weathering, and cracking must be carefully considered before drastic changes in rubber components are incorporated in a tire[77].

Construction

The radial tire is much in the forefront of tire news because of its contribution toward improved vehicle handling, superior treadwear, and often-attributed contribution to fuel economy. The consensus is that fuel economy improves if radials are substituted for bias ply or belted bias tires, but the reported extent of improvement varies considerably depending upon the source of information. Almost all data obtained concerning comparative rolling resistance and power loss, however, confirm the reported superiority of the radial tire in this respect[80,81]. The data reported in Figure 89 support radial tire superiority.

Reduced Deflection

Rolling resistance can be reduced by minimizing deflection of the tire as it rolls under load. For the same tire porfile, outside diameter, and section width, rolling resistance decreases with increase in

[78] Clark, Samuel K. ed., "Mechanics of Pneumatic Tires," National Bureau of Standards.
[79] Floyd, C. W., "Power Loss Testing of Tires," Paper 710576.
[80] John Abbott, Staff Engineer, Tire Design, Plant 1, Dept. 460G, Goodyear Tire & Rubber Co., Akron, Ohio, 10-26-73.
[81] Dr. Marion G. Pottinger, Section Manager, Advanced Tire Dynamics, Goodrich Research & Development Center, Brecksville, Ohio, 10-15-73.

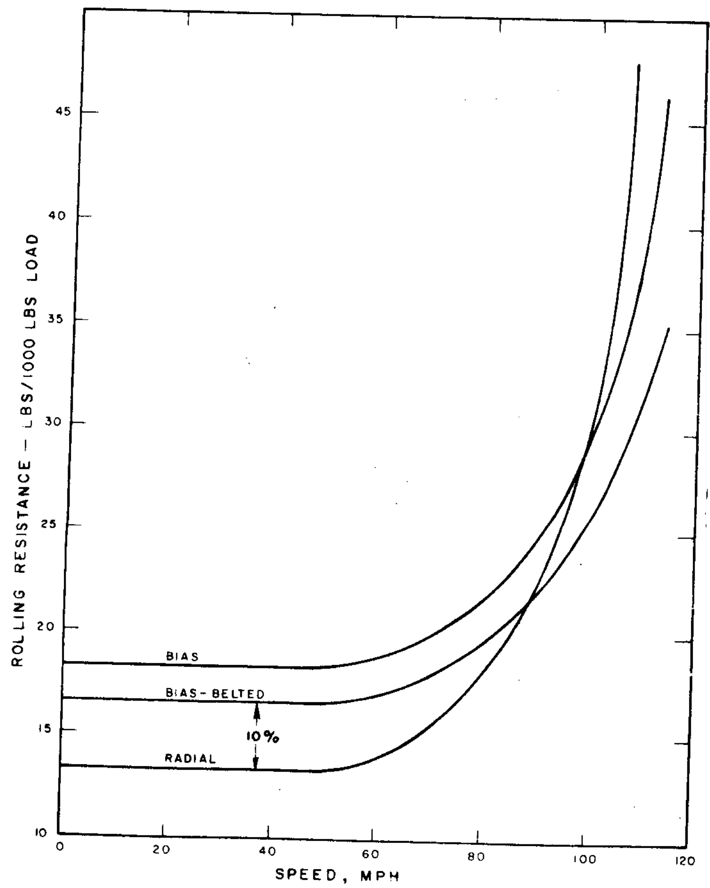


FIGURE 89. EFFECT OF TIRE CONSTRUCTION ON ROLLING RESISTANCE

rim diameter, although the improvement is quite small if the change is from 14 to 15 inches. A change from 12 to 16 in. can be significant. Rolling resistance can also be reduced by increasing rim width, thereby reducing tire deflection, although the improvement varies with tire construction, and is fairly small in any case and does not increase steadily with increase in rim width[77].

Evaluation

Effect of Reduced Rolling Resistance on Economy

The effect of reduction of rolling resistance on fuel economy, generally, has been regarded as marginal compared to effects of other vehicle and operating factors[72,75-77,79-93], yet its reduction is important to the overall subject of fuel economy. The road power consumption attributable to rolling friction (principally the tires) is dependent on road speed. Figure 90 illustrates a typical road load curve for one of the reference vehicles and the curve fit equation used to represent the experimental results. Below 30 mph the dominant portion of the road load is attributable to rolling resistance. In fact, the aerodynamic load component does not become equal to the rolling resistance until about 55 mph.

[82]Dr. Tomkins, Firestone Tire and Rubber Co., 1200 Firestone Parkway, Akron, Ohio, 1-24-73.
[83]Roberts, G. B., "Power Wasteage in Tires," Proceedings, International Rubber Conference, November 8-13, 1959, Washington, DC pp 57-71.
[84]Greenshields, R. G., "150 Mpg is Possible," *SAE Journal,* March 1950, pp 34-38.
[85]Don Ball, Tire Evaluation, Chelsa Proving Ground, Chrysler Corporation, Chelsea, Michigan, 10-15-73.
[86]Joseph Callahan, Editor, Automotive Industries, 8-3-73.
[87]R. R. Love, Assistant Chief Engineer, Engineering Office, Chelsea Proving Ground, Chrysler Corporation, Chelsea, Michigan 10-15-73.
[88]Robert Yeager, Group Leader, Tire Design Research & Development, Plant 1, Dept. 460G, Goodyear Tire & Rubber Co., Akron, Ohio 10-26-73.
[89]Dr. J. D. Walter, Division Manager, Central Research Laboratories, Firestone Tire & Rubber Co., 1200 Firestone Parkway, Akron, Ohio.
[90]Dr. R. H. Snyder, Director, Product Development Division, Uniroyal Co., 6600 E. Jefferson Ave., Detroit, Michigan 8-1-73, Letter 8-6-73
[91]Peterson, K. G., and R. E. Rassmussen, "Mechanical Properties of Radial Tires," Paper 730500.
[92]Goodenow, Gary, Thomas R. Kolhoff, and Fraser D. Smithson, "Tire-Road Friction Measuring System—A Second Generation," Paper 680137.
[93]Kelly, Kent B. and Henry J. Holcomb, "Aerodynamics for Body Engineers," SAE Paper No. 649A.

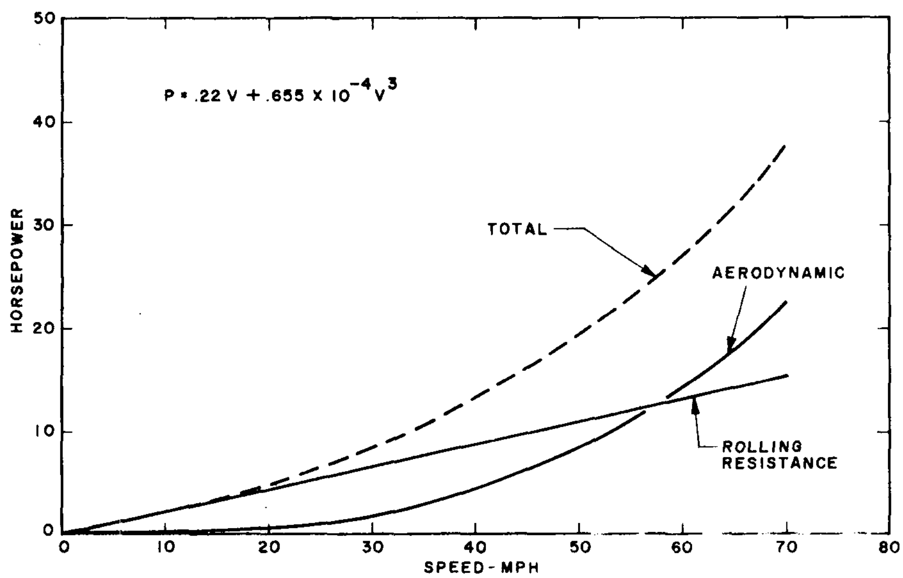


FIGURE 90. VEHICLE RESISTANCE

Based on the discussion presented in this section, it is reasonable to evaluate the economy improvement due to steel belted radial ply tires. For this calculation a 30-percent reduction in the rolling resistance component of road horsepower was considered.

As previously discussed in this report, the fuel consumption characteristics of carbureted spark-ignition engines are strongly dependent on load especially at the low torque (bmep) levels where considerable energy is also required to overcome friction and pumping power. This is reflected in the shape of the characteristic curves of the reference engine map used in the study.

The reduction of rolling resistance (for road load) reduces the torque output requirement of the engine, and the engine operates at a point of higher brake specific fuel consumption. The effect of power reduction is thus partially reduced (Figure 91), and at some speeds, the fuel consumption will be increased. At 20 mph, for example, the mileage is worsened. At higher speeds and torque outputs engine characteristics are such that specific fuel consumption is less sensitive to changes, and significant improvement in fuel economy can be noted as power demands decrease. At increasingly higher speeds, aerodynamic loads decrease the percentage influence on economy. Table 29 illustrates the above discussion. In reviewing the above table it must be remembered that the characteristic fuel economy of a vehicle is such that the mileage at 20 mph is significantly less than the mileage at 30 to 40 mph; consequently, this adverse lowering of mileage of 20 mph reduces the overall benefit to fuel economy based on the fuel economy evaluation considered in this study. It should also be noted that vehicle operation at 20 mph is associated with an extremely sensitive region of the engine map where small changes in brake mean effective pressure can result in large changes in brake specific fuel consumption.

Fuel economy during an urban driving cycle is also affected by a decrease in rolling resistance when actual road operation is considered. Use of the standard calculation technique described earlier resulted in the following percentage improvements:

Urban Cycle	Road Load	Composite
4%	4%	4%

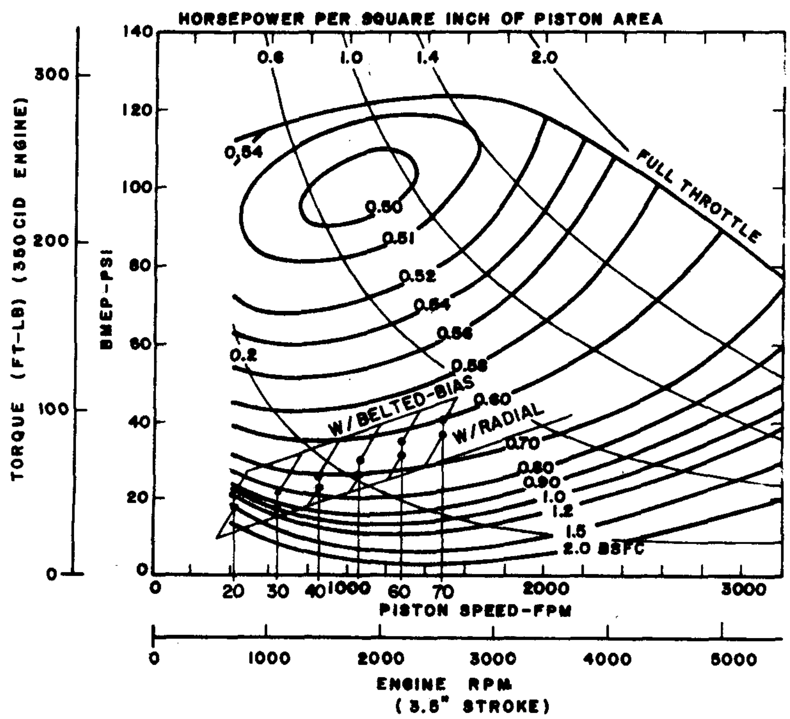


FIGURE 91. PERFORMANCE MAP OF CARBURETED SPARK IGNITION ENGINE

TABLE 29. PERCENT CHANGE IN MILEAGE WITH VEHICLE SPEED

Vehicle speed (mph)	Percent change in road load mileage
20	-10
30	+8
40	+8
50	+9
60	+7
70	+5

If reduction in tire rolling resistance is to be exploited for achievement of improved fuel economy, it must be accompanied by engine operation in higher bmep ranges, where specific fuel consumption changes less drastically with changes in bmep.

Stage of Development

The state-of-the-art in tire engineering and construction, polymer development, and compounding, and known materials are such that rolling resistance of a radial tire can be reduced 30 percent from that of a current belted bias tire.

Demonstration by 1976

Steel belted radials are currently marketed. Specific data are not available on the rolling resistance characteristics of different makes/models of tires. Several different tires would have to be screened from manufacturers test data to obtain tires that will attain a range of rolling resistance consistent with a 30-percent reduction with respect to the baseline bias ply type tire.

Reliability

The principle disadvantageous effects of further measures suggested to improve rolling resistance are slightly harsher ride, lower permissible speed, high susceptibility to impact damage and other road hazards and punctures, and reduction in miles to replacement. Higher inflation pressure and reduction in aspect ratio would result in reduced spring rate and somewhat harder ride with present vehicle suspension systems. Increasing the inflation pressure to rated pressure in a steel belted radial will not adversely affect the life and will reduce rolling resistance although the "harsher" ride penalty will be evident. Reduction in tread mass and lower hysteresis rubber compounds would result in reduced treadwear mileage, reduced protection against punctures, cuts, and carcass impact damage. Significant decrease in the tread mass would result in reduced treadwear mileage, reduced protection against punctures, cuts, and carcass impact damage. Significant decrease in the tread mass would result in a cooler running tire, but would require significant change in tire construction to yield performance equivalent to current production tires.

Cost

Present cost per mile for bias, belted bias, and radial tires varies with specific price quotation and service mileage experience; however, cost of a belted bias tire is estimated at about 55 percent of that of a radial, and mileage about 60 percent[94,95] that of a radial. Optional radial ply tires are approximately $75 higher than bias ply tires, including special suspension tuning requirements.

Safety

Safety is increased with a steel belted radial due to increased damage resistance of the tire.

Emissions

The present emission test procedure specifies values for dynamometer resistance in terms of loaded vehicle weight. In order to take advantage of reduced rolling assistance during the emission test, a manufacturer would be required to petition for a reduced dynamometer setting on the basis of data acquired for a particular vehicle design (as allowed in the regulation).

Noise

With the principle sources of tire noise being contact with the road surface and air disturbance, there appears no reason for sound intensity or frequency spectrum to change significantly as a result of use of lower rolling resistance tires.

Performance (Acceleration)

The reduction in rolling resistance will increase the margin between road load power requirements and WOT, thus making more net power available for vehicle acceleration.

Time Required for Implementation

Industry radial tire production capacity, as well as inherent characteristics of the radial, will bear on possible large scale fuel economy effects if national use of the radial tires is advocated. Radial tire production in 1972 was estimated at 7,500,000 units, or about 3.5 percent of the year's car tire total, and was expected to increase to 8 percent in 1973[95]. Estimates for radials in 1976 to 1977 by various sources are 25 to 30 percent of original equipment tires and 65 to 70 percent are replacements[95,96]. Estimates for 1982-1983 are for 60 to 80 percent of total car tires to be radials[97].

[94]Charles Martyn, Manager, Blue Ribbon Tire Co., San Antonio, Texas 11-16-73.
[95]Vila, George R., "Impact of the Radial-Ply Tire on U.S. Passenger Replacement Market," *Rubber Age*, September 1972, pp 61-67.
[96]*Tire Dealer*, March 1972, p 37.
[97]"Rubber Industry: A Glimpse of the Future," *Chemical & Chemical Engineering News*, April 17, 1972, p 10.

It is further estimated that by 1980 U.S. tire production will be exceeded by Japan's and Europe's combined, and that the latter will be almost entirely radial[98]. In this light, domestic production will be supplemented by imports. Michelin, of France, has just announced an investment of $175 million in two plants in North Carolina to begin production of radial tires in 1975[99].

Consumer Acceptance

Suspension systems tuned to the characteristics of radial ply tires are in production (at increased cost). Ride characteristics are not compromised. If higher inflation pressures are used to maximize benefits of radial ply tires some consumers may object to the ride harshness.

[98] "Radials Reshuffle Tire-Cord Lineup," *Chemical Week*, January 31, 1973, p 31.
[99] *Rubber Age*, October 1973.

19. AERODYNAMICS

Current Practice

The literature includes studies of aerodynamics as applied to cars over a span of years, with varying objectives. In the early 1930's in the U.S., and in England in the 1940's, improved fuel economy was the primary objective[100-110]. One source at this time cites development of the Chrysler Airflow for which maximum initial speed was 84 mph and fuel consumption 11.3 mpg at 80 mph. Through successive aerodynamic improvements, maximum speed was increased to 99.4 mph and fuel consumption to 17.7 mpg at 80 mph[107]. In the 1950's emphasis seemed to have shifted to gains in speed through aerodynamic applications[111]. The overall result was a reduction in aerodynamic drag between the years 1930 and 1950[112]. In the late 1960's emphasis returned to fuel economy[113-116].

One source[117] cites typical drag coefficients of 0.4–0.5 and frontal areas of 22–30 sq ft as of 1957. Another[118] cites coefficients of 0.45–0.70 for standard sedans as of 1965. Today's cars appear "aerodynamically cleaner" than models of 10 to 15 yr ago, yet neither drag coefficients nor frontal areas appear to have decreased appreciably. The increasing popularity of stock car and sports car racing, popular in Europe for many years and more recently in this country, has undoubtedly influenced body design and styling of some of the specialty cars. In full-size sedans, efforts toward more luxurious interiors, reduction of obstacles to driver vision, and elimination of blind spots may have some responsibility for departure from cleaner aerodynamic design. Little effort has been expended on underbody cleanup such as use of belly pans because of complexities introduced for servicing the vehicle. A European import several years ago required about one hour for removal of its belly pan and another hour for reinstallation on an ordinary lubrication job[119]. The automobile industry is presently working toward aerodynamic improvement of passenger cars[119].

Drag coefficients, frontal area, and the product C_dA (product of drag coefficient and frontal area), for four of the reference vehicles are listed in Table 30. The data acquired during the road test phase of the project did not allow precise separation of rolling and aerodynamic components; the

[100] McCain, George L., "Dynamics of the Modern Automobile," SAE Journal (Transactions) July 1934, pp 248-250.
[101] Zierer, W.E. and Macanlog, J.B., Jr., "Tank Mileage," SAE Journal (Transactions) January 1939, pp 29-34.
[102] Andreau, J., "European Streamlining Slashes Air Resistance," SAE Journal (Transactions) April 1939, p 350.
[103] Tietjans, O.G., "Economy of Streamlining the Automobile," SAE Journal (Transactions), March 1932, pp 150-152.
[104] Lay, W.E., "Is 50 Miles Per Gallon Possible with Correct Streamlining?" Part 1, SAE Journal (Transactions) April 1933, p 144-356.
[105] Lay, W.E., "Is 50 Miles Per Gallon Possible with Correct Streamlining?" Part 2, SAE Journal (Transactions) May 1933, p 177-186.
[106] Horine, M.C., Altman, P., Winter, H.G., Reid, E.G., and Upson, Ralph, "Differences Between Wind-Tunnel and Road Load Conditions," SAE Journal, August 1933, pp 261-267.
[107] Wilkins, Gordon, "Next Steps in Drag Reduction," Part 1, Autocar, March 5, 1948, pp 214-216.
[108] Wilkins, Gordon, "Next Steps in Drag Reductions," Part II, Autocar, March 12, 1948, pp 240-241.
[109] Reid, John P.M., "Aerodynamics of Motoring," Part 1, The Autocar, June 8, 1951, pp 656-659.
[110] Reid, J.P. Milford, "Aerodynamics of Motoring," Part 2, The Autocar, August 3, 1951, pp 904-907.
[111] Reid, J.P. Milford, "Aerodynamic Fallacies," The Autocar, September 11, 1953, pp 322-323.
[112] Hoerner, Sighard F., "Chapter XII, Drag of Land-Borne Vehicles," Fluid-Dynamic Drag 1965, Dr. Ing. S.F. Hoerner, 148 Busteed Drive, Midland Park, New Jersey, 07432, pp 12.1–12.10.
[113] Costin, Frank, "A Dozen Years of Aerodynamics," Autosport, December 27, 1968.
[114] "How Much is Air Drag Costing You?" Heavy Duty Trucking, October 1969, pp 25-28.
[115] "Cutting Wind Drag With Airshield," Heavy Duty Trucking, October 1969, pp 28-30.
[116] Wyss, Wally, "The Flying Brick," Car Life, February 1970, pp 28-30.
[117] Taborek, Jaroslav J., "Resistance Forces, Mechanics of Vehicles-6," Machine Design, August 8, 1967, pp 101-102.
[118] "The Automotive Assembly, Automotive Series, Engineering Design Handbook," AMC Pamphlet AMCP 706-355, February 1965, HQ, USAMC, pp 5-16.
[119] Tel. Contact, Mr. Kent B. Kelly, Staff Project Engineer, Advanced Project Engines, Engineering Staff, General Motors Corp., General Motors Technical Center, Warren, Michigan (313) 575-1093.

figures in Table 30 were obtained from the sources listed. Frontal area for both Chrysler cars is the true area rather than the modified product of width and height listed below.

TABLE 30. AERODYNAMIC CHARACTERISTICS OF REFERENCE VEHICLES

Make	Drag Coefficient	Frontal Area	C_dA
Chevrolet Impala(118)(a)	0.573	24.44	14
Chevrolet Camaro(118)(b)	0.503	20.68	10.4
Dodge Challenger(119)(c)	0.45	25.6	11.5
Plymouth Fury(119)	0.527	20.70	10.9

(a)These data are for a 1973 two-door Impala, which is the only vehicle in this line for which wind tunnel data were available.
(b)These data are for a 1971 model, and were the only data available for this body style. The 1971 and 1973 differ only in minor features such as trim, and are essentially the same aerodynamically. Both models were wind tunnel tested and C_dA was determined. Frontal area was assumed as 80% of the product of maximum width and maximum height, from which C_d was calculated.
(c)These data are for a 1970 model without air conditioning; otherwise, the car is aerodynamically the same as the 1973 model.

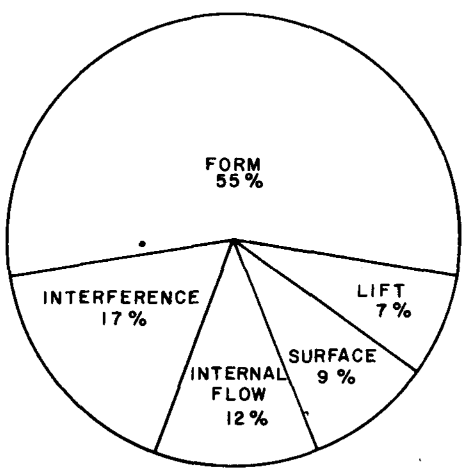


FIGURE 92. DISTRIBUTION OF VEHICLE AERODYNAMIC DRAG

Vehicle Aerodynamics

Road load horsepower consists of two major components: power required to overcome rolling resistance or mechanical drag, and air resistance, or aerodynamic drag. Mechanical drag includes tire rolling resistance, which is covered in the section on tires. Aerodynamic drag has external and internal components, that is, features that affect airflow through and inside the vehicle. One source[121] breaks down aerodynamic drag (Figure 92) as follows:

(1) Form drag–Drag due to body shape.

(2) Lift drag–Induced drag due, also, largely to body shape. It does not necessarily oppose forward motion, but its generation absorbs energy.

(3) Surface drag–Drag due to frictional resistance between body surfaces and air passing tangentially over them. It increases as the exterior area of the car body "wetted" by the airflow increases.

(4) Interference drag–Drag due to disturbances of the airflow over the body shape, caused by appendages and protuberances projecting from the basic form into the airstream. Such features as door handles, rain gutters, radio antenna mast, hood ornaments, outside rear view mirror, body columns, body trim, and roof luggage rack contribute. Projections under the vehicle such as engine pan, suspension members, exhaust system, and differential and rear axle housings also contribute to interference drag on the vehicle underbody.

(5) Internal flow drag–Drag due to the flow of air into, through, and out of the engine cooling system and passenger compartment.

Horsepower required to overcome aerodynamic resistance is a function of drag coefficient, frontal area, and the third power of the relative velocity between the car and air passing it. The drag coefficient

[120] Tel. Contact, M. William B. McNulty, Section and Development Supervisor, Performance Analysis Department, Chelsea Proving Ground, Chrysler Corp., Chelsea, Michigan (313) 475-8651, ext. 215.
[121] Kelly, Kent B., and Holcomb, Harry J., "Aerodynamics for Body Engines," SAE Paper 649A, January 1963.

is a composite of the five forms of drag enumerated above. In the usual wind tunnel determination of vehicle aerodynamic drag, the product of drag and frontal area, $C_d x A$, is determined. Frontal area is measured, and the drag coefficient is calculated. Should there be an error in measurement of frontal area, density or velocity of airflowing in the tunnel, or other test variables, the calculated drag coefficient will be in error. If this coefficient is applied to some other frontal area, the error will carry over into the result. So long as the corresponding frontal area and drag coefficient are used together, however, error will not be introduced.

Reduction of drag coefficient is the only means of reducing aerodynamic drag if vehicle size (width, height, frontal area) is to be maintained. Of the components of drag coefficient, form drag constitutes the largest segment, 55 percent. Interference drag, 17 percent, can be interpreted as including underbody drag in addition to projections along the sides, top, and front and rear ends. Internal flow drag represents about 12 percent of the total. Together, these make up about 84 percent of total drag. Effort in these areas should be most productive.

Effect of Aerodynamics on Fuel Economy

Because aerodynamic horsepower increases as the third power of the air velocity, it is of little consequence at low speeds. (See Figure 91). For the same reason, road load horsepower increases appreciably with headwinds and decreases with tailwinds. For full size cars of current design moving in still air, aerodynamic resistance becomes a significant factor in road load horsepower at about 50 mph, where it represents about one-half of the total.

The aerodynamics contribution to road load horsepower varies directly with the drag coefficient and frontal area. A reduction in either of these quantities, or their product $C_d A$, would result in a reduction of fuel consumption. The degree of fuel consumption reduction would vary with vehicle speed; several estimates are available from the literature. If aerodynamic drag were reduced 10 percent and engine and drive train characteristics remained the same, fuel consumption would be reduced 3.5 percent at 40 mph and 6.5 percent at 80 mph. Results to these extents are not achieved in practice because engine specific fuel consumption increases with decrease in road load, and changes in mechanical characteristics of the vehicle are necessary to realize potential gains from aerodynamic improvement[105,109]. For an intermediate size vehicle and no change except aerodynamics, a 10 percent decrease in aerodynamic drag results in an actual increase in miles per gallon of about 4.3 percent at 70 mph[122].

Evaluation—Aerodynamic Improvement

Target Coefficient—The British Motor Industry Association has developed a method of estimating drag coefficients[123] which appears reasonably accurate. Although it is not a substitute for wind tunnel measurements, it does provide guidelines for preliminary design. It places primary emphasis, apparently, on form drag and interference drag, yielding a coefficient for a vehicle with the radiator and grille blanked off. Experience has shown results thus determined agreeing within 7 percent of those derived from wind tunnel tests. The expression for calculating drag coefficients is:

$$C_d = 0.16 + 0.0095 \times \text{drag rating},$$

[122] Huebner, G.J. and Gasser, Donald J., "Energy and the Automobile-General Factors Affecting Vehicle Fuel Consumption," SAE Paper 730518.

[123] White, R.G.S., "Rating Scale Estimate Automobile Drag Coefficient, SAE Journal, June 1969, Vol. 77, No. 6, pp 52-53, also SAE Transactions 1969, pp 829-835.

where the drag rating is determined by reference to a table of numbers assigned to various configurations of each of nine zones of a vehicle (Table 31). The drag rating becomes the numerical sum of these numbers. Comparison of the drag coefficient so determined with a known coefficient for a vehicle provides a measure of the resulting change in fuel economy.

For estimating a reasonably achievable drag coefficient, the following ratings in each category were chosen from Table 31:

Category 1. Front End Plan Outline–Numerous current model U.S. cars are best described by rating "6", described as "squared constant-width front," or "4", "rounded corners with protuberances." Rating "2", "well rounded outer quarters," is selected for this category as representing effectiveness consistent with styling acceptable to the U.S. public. (2)

Category 2. Front End Elevation–Numerous 1973 model U.S. cars, particularly the luxury lines, are considered as corresponding to a rating of "5", "high squared front, with horizontal hood." Rating "3", "medium height rounded front, sloping up," is selected as representing some 1974 models, and a probable future trend as vehicle front ends are styled to accommodate massive shock absorbing front bumpers. (3)

Category 3. Windshield/Roof Junction (Cowl and fender cross section)–A rating of "4", "Hood flush with square edged fenders," is considered representative of many current U.S. production cars, particularly the Impala and Galaxie reference vehicles. A rating of "1", "Flush hood and fenders, well-rounded body sides," is considered practical, and consistent with the ratings for Categories 1 and 2. (1)

Category 4. Windshield Plan–A rating of "2" "Wrapped-round ends," is selected as a practical compromise between undistorted vision, satisfactory wiper blade sweep, and reasonably good aerodynamic effectiveness. It has been used extensively in U.S. vehicles in the past. (2)

Category 5. Windshield Peak–A rating of "1", "Rounded", is selected as representative of most current production U.S. vehicles. (1)

Category 6. Roof Plan–A rating of "2", "tapering to front and rear (max. width at BC post) or approximately constant "width", is selected as consistent with U.S. styling to accommodate three people in both front and rear seats. (2)

Category 7. Rear Roof/Trunk–A rating of "1", "Fastback (roof line continuous to tail)", is chosen as consistent with some current U.S. styling, and with Category 6. (1)

Category 8. Lower Rear End–A rating of "2", "Small taper to rear or constant width," is selected as consistent with the Category 6 conformation and with provision of reasonable trunk space. (2)

Category 9. Underbody–A rating of "1", "Integral, flush floor, little projecting mechanism," is selected as a realistic objective in spite of some unsatisfactory experiences with belly pans. (1)

TABLE 31. DRAG RATING SYSTEM

Table 2 - Drag Rating System

Rating	Category 1. Plan Outline	Category 2. Elevation(b)
	FRONT END:	
1	Approximately semicircular	(a) Low rounded front, sloping up
		(b) High tapered rounded hood
2	Well-rounded outer quarters	(a) Low squared front, sloping up
		(b) High tapered squared hood
3	Rounded corners without protuberances	Medium height rounded front, sloping up
4	Rounded corners with protuberances(a)	(a) Medium height squared front, sloping up
		(b) High rounded front, with horizontal hood
5	Squared tapering-in corners	High squared front, with horizontal hood
6	Squared constant-width front	
	Category 3 Cowl and fender cross-section	**Category 4. Windshield plan(c)**
	WINDSHIELD/ROOF JUNCTION	
	Flush hood and fenders well-rounded body sides	Full wrap-round (approximately semicircular)
2	High cowl, low fenders	Wrapped-round ends
3	(a) Hood flush with rounded-top fenders	Bowed
	(b) High cowl, with rounded-top fenders	

cont'd

Table 2 (cont'd)

Rating	Category 3. (cont'd)	Category 4. (cont'd)	
4	Hood flush with squared-edged fenders	Flat	
5	Depressed hood, with high squared edged fenders		
	Category 5. Windshield peak	**Category 6. Roof plan**	
1	Rounded	Well- or medium-tapered to rear	
2	Squared (including flanges or gutters)	Tapering to front and rear (max. width at BC post) or approximately constant width	
3	Forward-projecting peak	Tapering to front (max. width at rear)	
	Category 7. Rear Roof/Trunk(d)	**Category 8. Lower Rear End**	**Category 9. Underbody(e)**
1	Fastback (roof line continuous to tail)	Well- or medium-tapered to rear	Integral, flush floor, little projecting mechanism(1)
2	Semi fastback (with discontinuity in line to tail)	Small taper to rear or constant width	Intermediate
3	Squared roof with trunk rear edge squared	Outward taper (or flared-out fins)	Integral, projecting structure and mechanism(2)
4	(a) Rounded roof with rounded trunk		Intermediate
	(b) Squared roof with short or no trunk		
5	Rounded roof with short or no trunk		Deep chassis(3)

(a) Wing mirrors. Include in protuberances if at the wing leading end. Otherwise add 1.

(b) Add: 3 for separate wings; 4 for open front to wings (above bumper level); 2 for raised built-in headlamps; 4 for small separate headlamps; 7 for large separate headlamps.

(c) Add: 1 for upright windshield; 1 for prominent flanges or rain gutters.

(d) Add: 3 for high fins or sharp longitudinal edges to trunk; 2 for separate wings; Note: In all the ratings in this column, the trunk is assumed to be rounded laterally.

(e) (1) car 1 and 7; (2) car 12; (3) cars 10 and 24. Intermediate ratings applied from vehicle examination.

Note: Throughout table, the word "taper" or "tapered" refers to the plan view

Using the numerical sum of 15 from the above 9 categories as the total drag rating, the drag coefficient becomes:

$$C_d = 0.16 + 0.0095 \times 15 = 0.16 + 0.14 = 0.30$$

This drag coefficient should be considered to be the minimum theoretically possible. It was arrived at mechanically by the formula from the reference quoted and does not consider the impact on the vehicle design and styling to achieve this end. It is plausible that a substantial alteration of frontal area would also follow as a result of an attempt to minimize the drag coefficient.

Design of an automobile to achieve the lowest aerodynamic drag for a given package size would not be concentrated in the area of the drag coefficient but would also include consideration of alteration in the vehicle external size, i.e., the form of the car—fender lines, wheel openings, etc.

The aerodynamic component of reference vehicle road load horsepower characteristic was reduced by 10 percent to reflect the potential power demand reduction, and fuel consumption calculations were made. The result of these calculations is summarized below:

Maximum Aerodynamic Improvement

Urban Cycle	Road Load	Composite
1%	2%	1.5%

Stage of Development—The state-of-the-art in aerodynamics is such that cars with lower drag coefficients could be designed and built. Frontal area could also be reduced, achieving a reduction of about 10 percent by reducing height and width to approximately the height and width of the current intermediate size models. For example, the 79.8 in. width and 56.1 in. height of the Fury yield a block area of 4480 sq inches. The current Malibu's width of 76.6 in. and Gran Torino's height of 53.0 in. give a block area of 4060 in., approximately a 10-percent reduction. It is reasonable to achieve a 10-percent drag reduction by adopting the current technology of slightly reduced car size. Achievement of the objective of maximum drag reduction would require experimental determination of drag coefficients for initial models, and experience from field use for evaluation of passenger comfort, wind noise, vehicle servicing, and vehicle handling characteristics. Application of aerodynamic improvements is also visualized as involving formability of materials and development of joining techniques and would, accordingly, be related to weight reduction measures.

Demonstration by 1976—Even if given immediate priority, the lead time necessary for design, experimental testing, and production layout would make a 1976 demonstration date very critical. Any demonstration models would have to be handmade because of drastic departures from current body contours, production methods, finishing details, and changes in materials and joining methods.

The variability of aerodynamic drag factors as evidenced by Table 30 suggests that various 1973 or 1974 production models be assessed to determine shapes (or specific vehicles) having the greatest immediate potential of reduced aerodynamic drag with respect to the full-size reference vehicle. This vehicle then could serve adequately as the demonstration vehicle. In addition to the determination of an adequate demonstration vehicle, substantial data on the variability of vehicle design factors would be available.

Reliability—Changes in airflow for engine cooling and air-conditioning system functioning, and passenger compartment ventilation could result in problems in both areas for early prototypes, although sufficient lead time for investigation of such possible problems would minimize their effects. An indirect effect on reliability would be the influence of aerodynamic improvements such as belly pans on accessibility of mechanical components for maintenance and repair.

Cost—Changes in cost have not been assessed at this time. Additional body components such as belly pans would result in additional material and forming costs and, depending on specific underbody measures employed, maintenance costs would be increased somewhat by repair of severe dents and tears in a belly pan, or replacement of it. The incorporation of reduced frontal area (10 percent) by a small decrease in car dimensions will not change costs since cars of this size are already in production.

Safety—Possible effects of aerodynamic improvement on dynamic braking and directional stability would require investigation. The reduced aerodynamic drag at high speeds would result in higher loads on the vehicle brake system for equivalent stopping capability in these speed ranges, with attendant needs for corresponding dissipation of heat generated. Directional stability at higher speeds would be influenced by the relative locations of center of drag and center of gravity of the vehicle.

Emissions—Aerodynamic alterations resulting in a lower road load would affect emissions only as a result of reduced dynamometer power absorption settings during the EPA test. Power settings are currently specified according to loaded vehicle weight, although modifications are possible on the basis of an application, supported by test data, from the manufacturer to the EPA.

Noise—Until specific body conformations are worked out, it is difficult to estimate noise intensities or frequencies resulting from aerodynamic improvement. Certainly, improved airflow and reduction of eddies and other air disturbances should result in an improved noise environment in the vehicle.

Performance (Acceleration)—Reduction in road load resistance would not affect normal vehicle acceleration at low speeds, and would improve it at high speeds, assuming no engine changes are incorporated. Substantial lowering of aerodynamic loads would also allow a reduction in the installed horsepower requirement.

Time Required for Implementation—The Mustang was introduced in 1965 on the Falcon chassis, so that it was at that time primarily a new body concept on an existing chassis. The Camaro, introduced in 1967 following the demonstrated popularity of the Mustang, was an almost entirely new design, conceived probably 3 yr earlier. Achievement of a low-drag coefficient vehicle would require a minimum of 1 yr of engineering development. Considering the lead time for the auto industry, it is estimated that readying a low-drag coefficient vehicle would be accomplished by the 1980-model year. The extent of the possible drag reduction would depend upon the outcome of wind tunnel tests conducted during the development program.

20. WEIGHT

Current Practice

The three full-size reference vehicles, Fury, Galaxie, and Impala, are from 400 to 600 lb heavier and from 6 to 14 in. longer than corresponding 1960 models. For the Fury, 58 percent of the 430-lb weight increase came between 1960 and 1967 models, while only 26 percent of the 14 in. length increase came during the same period. For the Galaxie, all of the 600-lb weight increase and all of the 6.5 in. length increase came between the 1967 and 1973 models. For the Impala, 83 percent of the 580-lb weight increase and 53 percent of the length increase came between the 1967 and 1973 models[123,124,125]. Weights of the three full-size reference vehicles are plotted against model year in Figure 93, and for the lighter vehicles in Figure 94. The 1970 models of all the full-size reference vehicles were almost equal in weight. It is obvious that weights of the Galaxie and Impala increased dramatically during the 1967 to 1973 period, but data for previous years plus Figure 93 indicate a fairly uniform weight increase for the Fury between 1960 and 1973. Widths of the three vehicles remained essentially constant during the 1960 to 1973 period while heights decreased slightly, probably accompanied by decreased ground clearance.

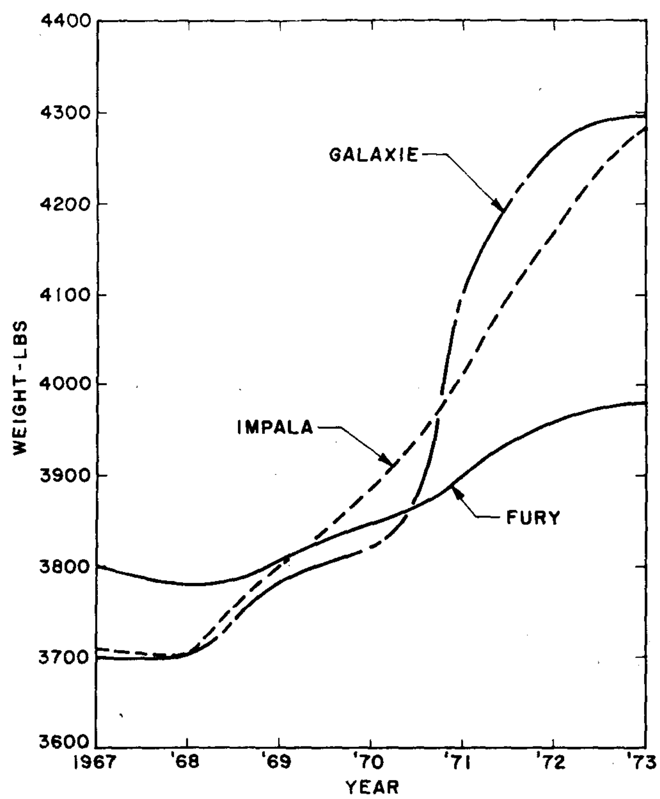


FIGURE 93. WEIGHTS OF VEHICLES BY YEARLY MODEL FULL-SIZE VEHICLES

[123] *Statistical Issue, Automotive Industries,* March 15, 1960.
[124] *1967 Almanac Issue, Automotive News,* April 24, 1967.
[125] *1973 Almanac Issue, Automotive News,* November 19, 1973.

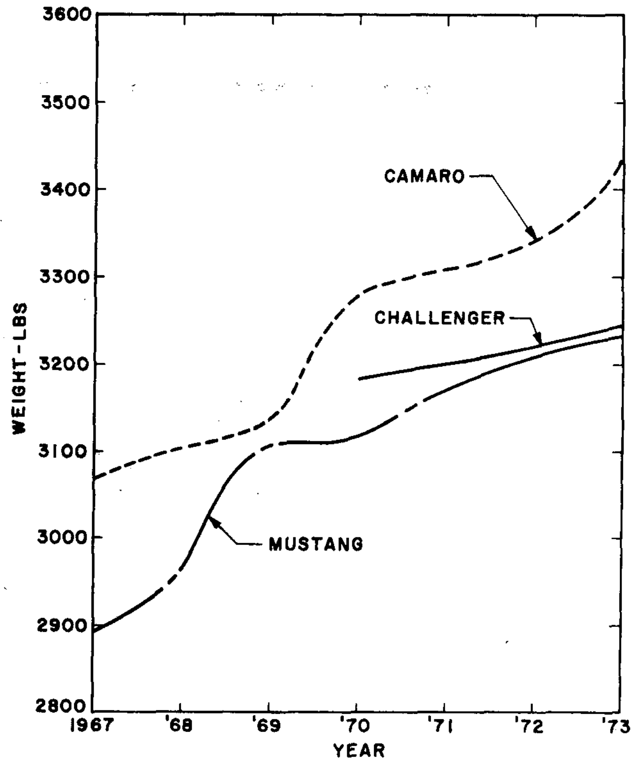


FIGURE 94. WEIGHTS OF VEHICLES BY YEARLY MODEL
LIGHTER VEHICLES

In Table 32, total vehicle weight, length, and weight per inch overall length are tabulated for each of the full-size vehicles.

It is interesting to note that the 19.5 lb/in. unit weight of the Galaxie, applied to the overall 1967 length, would result in a 150-lb lighter 1973 car. Similarly, if the 19.3 lb/in. unit weight of the 1973 Impala was applied to the 213.2 in. overall length of the 1967 model, a weight decrease of 160 lb would result. The weight of the Fury per unit length has remained fairly constant over the period, but if the 1973 value of 17.8 lb/in. was applied to the 1967 overall length, a weight decrease of 180 lb would result. Admittedly, federally mandated safety features and components of emission control systems have added some length and weight, but it appears that the mere increase in car length has also been responsible.

TABLE 32. LENGTH/WEIGHT RATIOS FOR REFERENCE VEHICLES

Year		Galaxie			Fury			Impala	
	Weight	Length	lb/in.	Weight	Length	lb/in.	Weight	Length	lb/in.
1967	3700	213.0	17.4	3800	213.1	17.8	3700	213.2	17.4
1968	3700	213.0	17.4	3780	213.0	17.8	3700	213.2	17.4
1969	3780	213.3	17.7	3808	214.5	17.8	3800	214.7	17.7
1970	3820	213.9	17.9	3844	215.3	17.9	3888	215.9	18.0
1971	4100	216.2	19.0	3900	215.1	18.2	4010	216.8	18.5
1972	4260	218.4	19.5	3960	217.2	18.2	4170	219.9	19.0
1973	4300	219.5	19.5	3980	223.4	17.8	4280	221.9	19.3

TABLE 33. MATERIAL BREAKDOWN FOR 1973 FURY

Material	Weight (lb)
Carbon steel	2105
Galvanized steel	75
Aluminized steel	31
Alloy steel	95
Stainless steel	13
Iron, cast, malleable, nodular	651
Aluminum	75
Fluids & lubricants	216
Rubber	201
Glass	96
Plastics	125
Soft trim, composition, and similar	110
Paint and protective dips	26
Sound deadeners, sealers	77
Lead and body solder	27
Zinc	65
Copper	26
Total weight	4014

TABLE 34. RUBBER PARTS IN THE AVERAGE 1974 CAR

Part	Weight (lb)
Seat cushion and crash pads	30
Engine mounts, body insulation, bumper parts	15
Hose	10
Belts, strips, tubes, grommets, etc.	95

For a 1973 Fury with a V-8 engine, automatic transmission, power steering, power brakes, and radio, but not air-conditioning, the following weight breakdown by materials is reported in Table 33.[26] Of this total, approximately 3000 lb is steel or iron, the majority of which is in stressed components where structural integrity is a primary requirement.

Another source[127] estimates about 300 lb of rubber in 1974 cars, of which tires account for less than half. Some of the rubber parts, on the average are shown in Table 34.

Plastics use is increasing at a rate of 10 to 15 lb/yr[128], and it is being used to varying degrees in different cars.

Oldsmobile is using acrylic modified low profile sheet molding compound in the energy-absorbing front bumper system, and in 1972 all 88 and 98 panels were tooled only for SMC[129].

"Soft face" applications of plastics are being investigated in bumpers, fenders, and door panels[130,131], although weight reduction appears to be an objective concurrent with reduced damageability rather than a prime one.

Aluminum use appears to be primarily for trim applications and other purposes where its corrosion resistance and appearance rather than strength/weight ratio are important. Recent exceptions are the Vega engine block and Vega and Camaro bumpers[132].

High strength, low alloy steels are being investigated by the steel companies and the auto makers as means of reducing weight, but they have not as yet been used extensively in automobiles[133,134].

Weight Reduction Potentialities

Potentialities for weight reduction lie basically in using less of present component materials or lighter materials for the same components. The first may be accomplished by reduction in gage or

126"What are cars made of?", *Automotive News*, September 10, 1973.
127Wolf, Ralph F., "Rubber Use in 1974 Autos," *Rubber Age*, October 1973, pp 36-44.
128Waddell, Richard L., "How Ford Decides on Plastics–or Not," *Ward's Auto World*, September 1973, p 71.
129Williamson, Don, "Materials men stress reducing car weight," *Automotive News*, March 26, 1973, pp 29-30.
130Remarks by Edward N. Cole, General Motors Corporation at the Society of Plastics Engineers, Detroit, Michigan, September 10, 1973.
131"GM tests 'friendly fenders' and plastic door panels," *Automotive News*, September 24, 1973, pp 10, 16.
132Waddell, Richard L., "How They're Battling the Bulge-Aluminum–Bumpers Now, Body Parts on the Way," *Ward's Auto World*, September 1973, pp 42-43.
133Waddell, Richard L., "How They're Battling the Bulge-Steel- 'High Strength Low Alloy' Sums it Up," *Ward's Auto World*, September 1973, pp 40-41.
134Thompson, Donald B., "Auto's Weight Reduction Push is Challenge for Steelmakers," *Industry Week*, September 10, 1973, pp 26-29.

dimension, redesign to reduce size or volume or to combine several components in one. For example, thickness of body sheet metal may be reduced, lightening holes may be incorporated in door, hood, or deck lid inner panels, and thickness of walls and webs of castings may be reduced. Use of lower density materials such as aluminum, magnesium, and plastics is an alternative method being pursued. A third avenue is a combination of the two approaches, such as redesign of the brake master cylinder to replace the integral cast iron fluid reservoir with plastic.

Use of aluminum and plastics is being investigated. Substitution for currently used steels, however, must take into account other properties such as formability, tensile strength, modulus, aging, temperature effects, paint and other coatings receptivity, etc.

Unreinforced polypropylene and polyethylene compounds weigh only about 15 percent as much as steel, but they have tensile strengths of around 4000 psi. With glass reinforcement, tensile may be increased to 12,000 to 15,000 psi for injection or blow molding. Tensile strengths of around 50,000 psi to as high as 250,000 psi are attainable with low-pressure, molded phenolics, polyesters, and epoxies, using glass mats, woven fabric, and filament winding. The polyesters, using glass mat or woven fabric reinforcing, are the most widely used plastics, one use being in car bodies[135]. Stampable, glass-reinforced, sheet molding compounds can, reportedly, be processed in existing sheet steel stamping presses with a minimum of modification, avoiding formability problems encountered with high strength steels and aluminum alloys and added expense for dies appropriate to their properties[136,137,138]. Additional thickness to achieve suitable tensile and modulus prevents gage for gage substitutions, but overall weight reductions of better than 65 percent are claimed[137].

Plastic wheels are one plastics application, offering not only weight reduction, but also consequent reduction in unsprung weight and rotational inertia. A Citroen SM with plastic wheels won the 1971 Morocco road rally, and Michelin Tire in France has been experimenting with them. A recently developed filament wound plastic wheel weighs half as much as magnesium and has eight times the tensile strength[139]. On this basis, wheel weight should be only 2 to 3 lb as compared to 21 lb for a steel production wheel, or nearly 100 lb difference for five wheels.

Technical problems bearing on use of plastics in high volume include ability of the industry to produce raw materials in sufficient quantity, recycling scrap, paintability of the formed product, and production rates per piece.

The many proprietary plastics formulations lead to questions of substitution and interchangeability, and influence the possible availability of assured supplies of raw materials on a large scale. Statistics and projections for plastics production are in terms of billions of pounds, which must be broken down into types to determine suitability for auto industry needs. For example, United States production was one billion pounds in 1946, 20 billion pounds in 1971, 24 billion pounds in 1972, and 227 billion pounds by the year 2000[130] is forecast. The portions of these quantities that could, or would, be used by the auto industry is dependent on a number of factors. Certainly, different formulations are useful in different applications within the same vehicle. One estimate is 200 to 300 lb of plastics per vehicle by 1980, provided raw material feedstocks are available.

[135] "Materials in Design Engineering," Materials Selector Issue, Mid-October 1966-67, Vol. 64, No. 5, pp 231-33.
[136] "Plastics Big Savings are in the Plant," *Industry Week*, October 29, 1973, p 81.
[137] "Reinforced Sheet Turns Detroit into Stamping Ground for Thermoplastics," *Plastics World*, November 1971, pp 204-5.
[138] Waddell, Richard L., "How They're Battling the Bulge-Plastics–Versatility, Weight, Cost Are the Keys," *Ward's Auto World*, September 1973, pp 44-45.
[139] Norbye, Jan P., "Plastic Auto Wheel–Stronger Than Steel," *Popular Science*, October 1973, p 18.

Shortages of plastic parts are affecting delivery of current model cars, and questions concerning future petroleum supplies and division of those supplies between using industries as affecting plastics feedstocks raise further questions regarding the availability of plastics in the kind and quantities necessary[140].

Pontiac had trouble with painting and porosity problems in 1971 with its "T-37" front-end assembly, which experience reacted unfavorably to sheet molding compounds for some time thereafter[141]. Plastic gravel shields between new energy absorbing bumpers and front-end grilles, however, are apparently accepting paint to match car colors without trouble. Still, plastics with thermal properties to accommodate higher ambient or paint-oven temperatures are desired[140].

Present production rates for auto components range from 250 to 500 pieces per line[142] to 600 pieces/hr[133]. Cycle time for one sheet molding compound operation is 15 sec, or 260 pieces/hr, comparing favorably with the lower rates[137]. Injection molded, or sheet layups, undoubtedly have slower cycle times, working to their disadvantage. To some extent, lower production rates are compensated for by the facility with which plastics lend themselves to consolidation of a number of parts into a single unit as compared with present sheet metal practices of welding together several formed sheet metal components to produce a single completed unit[136,137].

Aluminum weighs only one-third as much as steel, and, in some alloys, is equal to mild steel in tensile strength. Its elastic modulus, however, is only one-third that of steel[133], which affects its application and weight advantage. Gage for gage, it tends to flutter and dent excessively and, accordingly, is usually 15 to 30 percent thicker in similar applications, with the result that the weight reduction in substituting it for steel is only 50 to 60 percent[143]. Experience of the French Panhard with aluminum is cited as an example of weight reduction achievement. The aluminum body shell of the Panhard Dyna 54 weighed 1474 lb as compared to a weight of 2470 lb in steel. The reported cost differential was $200 for aluminum over steel[129].

The following weights for aluminum as compared to steel body components are reported in Table 35 by an aluminum company. The weight reduction for each component and the total for all is about 60 percent. Comparative weights reported by one auto manufacturer for hoods and rear deck of two of the reference vehicles are shown in Table 36.[145] The weight reduction for aluminum in this case is about 50 percent.

TABLE 35. COMPARATIVE ALUMINUM AND STEEL WEIGHTS[144]

Component	Weight (lb)	
	Aluminum	Steel
Hood	35	90
Trunk lid	30	75
Doors (4)	100	250
Front fender (2)	60	140
Bumpers (2)	50	130
Miscellaneous	30	70
Total	305	755

From 1961 through 1963, Buick and Oldsmobile produced a 215 cu in. V-8 with aluminum block and heads, Buick with 9.0:1 and 11.0:1 compression ratios and Oldsmobile with 8.25:1 and 10.25:1. One problem with the V-8 heads was stripping of sparkplug holes when plugs were changed frequently. Reportedly, the engine was satisfactory, but public acceptance was not good. Many are still in service, and tooling and manufacturing rights have been assigned to the Rover Company in England. Complications of the all aluminum design include the necessity of steel

140 "Oil Shortage Spinoff–Automakers run short of Plastic Parts," *Industry Week*, November 26, 1973, p 85.
141 "RP Innovations vie for Auto Market," *Plastic World*, November 1973, p 204.
142 Callahan, Joseph, "Chrylser's Weight Watchers," *Automotive Industries*, October 15, 1972, pp 27-31.
143 Telephone communication August 1, 1973, with Donald J. Funk, Automotive Specialist, Reynolds Metals Co., 16000 Northland Drive, Southfield, Michigan.
144 Cochran, C. Norman, "Aluminum-Villain or Hero in Energy Crisis?" *Automotive Engineering*, June 1973, pp 57-61.
145 Telephone communications, October 1, 1973, Mr. Harry T. Tillotson, Manager, Body Safety Engineering Department, Ford Motor Company, Dearborn, Michigan.

TABLE 36. HOOD AND DECK WEIGHTS IN STEEL AND ALUMINUM[145]

Component	Weight (lb) (Actual) Steel	(Calculated) Aluminum
4500-lb Inertia weight vehicle		
Hood	73.9	33.9
Deck	37	19
3500-lb Inertia weight vehicle		
Hood	32.7	16.4
Deck	23.1	10.8

valve seat inserts. The Vega engine uses a die cast aluminum block and cast iron head[146].

An opinion in opposition to the above is that an aluminum alloy suitable for an engine has not been found. Aluminum is not suitable for casting difficult shapes; corrosion in water jacket passages can be a problem,as can galvanic corrosion where dissimilar metals are in contact, such as valve seat inserts, cylinder head, and manifold bolts or studs, etc.[147]

The automobile makers are working with aluminum in body components as a means of reducing weight. Bolt-on components such as hood, doors, deck lid, and station wagon tailgates are the principal candidates. The hood is of greatest interest because it is the lowest stressed. The formability of aluminum is a problem because it does not have drawing characteristics of presently used steels. Bend radii will have to be increased, which body stylists do not like[145]. Spring back in forming is different from steel, requiring changes in die design[142]. Stresses in bolt-on body shell components are principally wind loads and will be the same for either steel or aluminum. Accordingly, door hinges, locks, and other hardware will still be made of steel. Problems of joining have not been solved; welding is slow and expensive, and combinations of steel and aluminum will encounter galvanic corrosion in contact areas[145]. Adhesives and metal stitching are being investigated[142,145]. Inplant repair techniques are not yet satisfactory to the auto makers and an acceptable filler material for dents has not yet been found. Paint booths must be developed that will handle both steel and aluminum[148].

Claims are made that costs for aluminum use will be reduced because stamping plants generate scrap equal to 30 to 50 percents of the total material used, and aluminum scrap brings 0.19/lb as compared to steel's 0.02/lb. The automobile manufacturers want assurance that aluminum from scraped cars can be recycled into a pure product. Aluminum alloys currently proposed for exterior and interior body panels pose recycling problems. Types 2036 and 5182 are not compatible and would have to be segregated. Technology necessary for doing so does not exist[148].

TABLE 37. ALUMINUM USE PROJECTION 1973-1980

Year	Body Component
1973-74	Front bumper systems
1974	Rear bumper systems
1975	Hood
1976	Trunk lid
1976-77	Doors
1980	Fenders and rocker panels

Current industry thinking regarding use of aluminum, increasing from around 80 lb per car in 1973 to 200 lb in 1980, is indicated in Table 37[132]. If a 50-percent weight reduction through substitution of aluminum for steel is assumed, the projected 120-lb increase in weight of aluminum used would reduce net weight only 120 lb. Presumably, the items listed are exterior shell only, excepting the bumper systems. If inner panels, brackets, stiffeners, etc., are included, the weight change becomes much more significant.

The aluminum companies say there will be no supply difficulties[145], yet a severe national shortage of aluminum was reported near the end of November[149].

[146]Telephone communication, December 4, 1973, Mr. Donald Dunlap, Chief Salesman, Control Foundry Division, General Motors Corporation, Saginaw, Michigan (517) 754-9151.
[147]Telephone communication, December 3, 1973, Dr. D. C. Williams, Professor of Metallurgical Engineering, Ohio State University, Columbus, Ohio 43210 (614) 422-5770.
[148]"More Aluminum in Autos Seems Certain, but Recycling is Hurdle," *Industry Week*, December 3, 1973, pp 24-26.
[149]"Aluminum Work Cut," Washington (AP) dateline, San Antonio Express Newspaper, November 29, 1973.

High strength, low alloy (HSLA) steels offer a potential 15 to 20 percent weight decrease through reduction in gage for body panels from 0.027 in. to 0.035 in. common today, with yield strengths of 45,000 to 80,000 psi as compared to carbon steel averages of 32,000 to 35,000 psi currently used. Some HSLA steels are being used in front door side guard beam assemblies and others are being developed with improved fabrication properties[151].

HSLA steels currently have yield points of about 50,000 psi. At higher levels, ductility deteriorates so that forming becomes difficult. As the thickness of the material decreases, particularly for body panels, tendencies to flutter increase[152]. Fisher Body is moving from 40,000 to 60,000 psi yield strength stock for front door outer panel bars (guard beams) on a trial basis, and it is investigating the manufacturability of steels in the 80,000-psi range. So far, mills have not been able to produce, or have not produced, 80,000-psi sheet in the thickness associated with auto body panels and do not produce a cold-rolled sheet in this class with any formability. Use of high-strength steels of this order appears to be limited to structural component applications[134].

TABLE 38. VEHICLE LENGTH AND WEIGHT, 1967-1973

Make	Length (in.)			Weight (lb)		
	1967	1973	Incr.	1967	1973	Incr.
Impala	213.2	221.9	8.7	3700	4280	580
Galaxie	213.0	219.5	6.7	3700	4300	600
Fury	213.1	223.4	10.3	3800	3980	180
Average increase			8.9			453

TABLE 39. COMPARATIVE DATA FOR 1973 FULL-SIZE AND INTERMEDIATES

Model	Length (in.)	Width (in.)	Height (in.)	Weight (lb)
Full-size				
Impala	221.9	79.5	54.5	4280
Galaxie	219.5	79.5	54.3	4300
Fury	223.4	79.8	56.1	3980
Intermediate				
Malibu	213.3	76.6	53.8	3695
Torino	212.0	79.3	53.0	3838
Satellite	213.3	78.6	53.7	3720
Differences				
Impala/Malibu	8.6	2.9	0.7	589
Galaxie/Torino	7.5	0.2	1.3	454
Fury/Satellite	10.0	1.2	2.4	355
Average	8.7			466

The auto industry hopes to reduce the weight of a full-size car from an average of 4300 lb to 3500 lb by 1979[136], and a Detroit slogan is "A thousand pounds out of the 'B' body by 1980"[153]. One avenue is reduction of car size using conventional materials, another is use of lighter materials with no reduction in car size.

Table 38 shows the growth of the three reference vehicles in size and weight from 1967 to 1973.

Table 39 compares the sizes and weights of the 1973 full-size reference vehicles with their intermediate versions.

These data indicate that the growth in size and weight of the 1973 reference vehicles, on the average, is about the same as the difference between the intermediate and corresponding full-size vehicles, approximately 9 in. in overall length and 450 to 500 lb in gross weight. Differences in height and width are not of consequence. In other words, if the 1973 intermediates were to become considered full-size cars and the current full-size cars were dropped or were to be considered "luxury" cars, the current intermediates would be equivalent to the 1967 full-size cars in size, and their weight would be reduced approximately 500 lb from present full-size cars.

[151] "Auto's Weight Reduction Push is Challenge for Steelmakers," *Industry Week*, September 10, 1973, pp 26-29.
[152] Telephone communications, October 5, 1973, Don Horan, Automotive Marketing, U. S. Steel Corp., Detroit, Michigan (313) 354-4511.
[153] Telephone communications with Stephen Sikes, General Manager, G.R.T.L. Co., Southfield, Michigan (313) 352-3935.

Adherence to present vehicle sizes dictates redesign to use lighter weight materials: aluminum, various plastics, and high-strength, low alloy steels.

Engine—The engine and transmission make up the highest concentration of mass in the vehicle. Components such as crank and camshafts, valves, lifters, followers, connecting rods, gears, and chains are steel and probably will continue so. Others, formerly of iron or steel, have advantageously made use of aluminum. Pistons are now almost universally aluminum alloy, and automatic transmissions are changing from steel and iron castings to aluminum. Iron generator housings have been replaced by aluminum alternator housings. The engine block, heads, and intake manifold, however, are cast iron in almost all American-built cars, but they are adaptable to aluminum construction provided suitable alloys are used, and inserts incorporated for valve seats, sparkplug holes, etc.

Body Shell—Front and rear-end body shell components are currently being made of plastic, some rigid, some soft. Large components are adaptable to aluminum, plastics, and combinations, including retention of steel components. Some of these are hood, deck, front fender, quarter panel, and doors. In some cases, outer skin may be aluminum and inner panels plastic, with either aluminum or steel brackets and other supports. Simplicity of manufacture and joining favors all-aluminum.

Wheels—Magnesium, aluminum, and plastic wheels are available, each with weight saving advantages. Aluminum is considered most practical in large-scale production.

TABLE 40. WEIGHT REDUCTION, 4500-LB INERTIA WEIGHT VEHICLE

Item	Weight, Material	
	Present	Proposed
Engine block, Chev. 350	163 (C.I.)	82 (Al)
Engine heads (2)	86 (C.I.)	43 (Al)
Intake manifold	46 (C.I.)	23 (Al)
Hood (Galaxie)	74 (St.)	34 (Al)
Deck (Galaxie)	37 (St.)	19 (Al)
Front fender assy (2) (AMF)	123 (St.)	62 (Al)
Quarter panel assy (2) (AMF)	198 (St.)	99 (Al)
Doors (4) (AMF)	175 (St.)	88 (Al)
Wheels (5) (SwRI)	105 (St.)	53 (Al)
Radiator support assy (AMF)	28 (St.)	14 (Al)
Side panel assy (2) (AMF)	184 (St.)	156 (HSLA)
Shelf & deck assy (AMF)	98 (St.)	83 (HSLA)
Dash-cowl assy (AMF)	40 (St.)	34 (HSLA)
Floor pan assy (AMF)	250 (St.)	212 (HSLA)
Roof (AMF)	52 (St.)	44 (HSLA)
Front bumper (estimated)	150 (St.)	100 (Al, plastic, HSLA)
Rear bumper (estimated)	100 (St.)	70 (Al, plastic, HSLA)
Total	1909	1216

Notes: 1. 693 LB Reduction
2. C.I. – Cast Iron
Al – Aluminum
St – Steel
HSLA – High Strength Low Alloy

Body Framework—The body framework is subject to major stresses, and high-strength, low alloy steels are presently most applicable. Among these applications in a unitized body construction are the two side panel assemblies, shelf and deck assembly, dash-cowl assembly, floor pan assembly, and roof[154].

Bumpers—Many types of energy absorbing bumpers are in production and under development. Aluminum, steel, plastics, rubber, and hydraulics are all being used, and combinations of this type offer good compromises between satisfactory functioning and weight reduction.

Other—Rear axle housing and drive shaft contribute significantly to vehicle weight and are adaptable to aluminum construction. Others, lightly stressed or unstressed, such as rocker covers, timing gear and chain covers, air filter housing, fan, fan shroud, etc., in aggregate, are significant weight constituents and are adaptable to either aluminum or plastics.

Possible changes in materials and weight reduction for various components are indicated in Table 40. The suggested aggregate

[154] "Tradeoff & Integration Systems Studies," Final Report, Contract DOT-HS-257-2-514, Section 12, Producibility Document No. ASL-TIS-103, 30 June 1973, for U.S. Department of Transportation, National Highway Traffic Safety Administration, 400 Seventh Street, S.W., Washington, D.C. 20590, by AMF Incorporated, Advanced Systems Laboratory, Golota, California 93017.

weight reduction does not include the additional compounding effect which reduces overall vehicle weight an additional 30 to 50 percent of each pound of weight removed[136,144,153].

Review of the discussion on alternative methods of weight reduction reveals that there are several avenues of approach to achieving weight reduction, and the magnitude also varies. Based on the considerations of Table 40, it appears that an extensive materials change could result in a weight savings of about 700 lb, including a change to an aluminum engine. Table 35 estimates the potential at about 450 lb. It is our conservative opinion that a 500-lb weight reduction is feasible for the 4300-lb curb weight vehicle. This results in a final vehicle weight of 3800 lb with size unchanged.

Alternately, the vehicle size can be reduced to the intermediate level with existing technology to achieve the same 3800 lb.

Effect of Weight on Fuel Economy

Rolling resistance, acceleration, and hill climbing (grade) capability are all functions of vehicle weight and affect fuel economy. The extent to which they do varies with engine and other vehicle characteristics, which is indicated in following paragraphs.

Evaluation—Weight Reduction

Reduction in Fuel Consumption

Weight reduction can reduce the rolling resistance of tires and, hence, the horsepower required for equivalent road load performance. It can also reduce the horsepower required for accelerating and hill climbing, hence, engine size for equivalent performance.

Fuel Consumption—Steady-State and Road Load

The weight reduction proposed are (1) to lower the curb weight to 3800 lb with a reduction to intermediate size and (2) to lower the curb weight to 3700 lb with the vehicle size unchanged. Both of these weight change options will result in a vehicle inertia test weight, based on EPA regulation, of 4000 lb (a 500-lb test weight decrease).

As a means of demonstrating the effect of weight change on fuel economy, calculations were performed according to the standard procedure for a vehicle having a curb weight of 3600 lb. The road horsepower was modified by comparison to that for the reference vehicle in order to account for the effect of weight on tire rolling resistance. The results of the calculations showed the following percentage improvements:

Urban Cycle	Road Load	Composite
7%	6%	7%

In the event that the proposed weight alteration is affected, then it would be possible to reduce engine displacement without changing the performance characteristics of the vehicle. This step would allow operation of the engine in a more favorable region of the engine map, and the fuel economy benefit would be enhanced.

Stage of Development

The first option, that of making the current "Intermediate" car the "Full Size" is, essentially, in production by all the major U.S. auto builders.

All the major auto manufacturers are working with aluminum, plastics, and high-strength, low alloy steels and are developing forming, joining, repair, and finishing techniques and have little experience with serviceability and public acceptance. It is doubtful that any have firm plans and procedures for specific materials and fabrication techniques in total.

Demonstration by 1976

The first option is demonstrable immediately and could undoubtedly be improved and refined by 1976.

Several automobiles, including the Kissel in the 1920's and the Panhard Dyna 54 more recently[129], have had aluminum body shells, and Reynolds Metals Co. currently exhibits an "all aluminum" car on its showroom floor[132]. The extent to which it is built of aluminum and whether or not it is entirely functional are not known. The technology exists to build a car, lightened in the manner proposed, and at least demonstrate its fuel economy characteristics. The necessity of doing so, however, due to the cost of the prototype and the moderate benefit, appears to be only of academic interest to the subject of the fuel economy potential.

Reliability

The first option, the "Intermediate" size vehicle, is on the road today and has demonstrated its reliability for several years.

The second option, replacement of current materials to achieve a lighter car of the same size, lacks experience in long-term strength, fatigue, corrosion resistance, and other service factors to warrant production except on a step-by-step basis until the concept and fabrication methods and techniques are proven.

Cost

For the first option, car costs will remain stable except for inflationary trends and further modifications to meet safety and emission control requirements with additional add-on components. The cost of the vehicle of the "smaller" lighter weight can be expected to be less.

For the second, all alternative materials are more expensive than the commercially used counterpart materials on the basis of cost per unit weight. For example, prices for mild steel, high-strength, low alloy steel, and aluminum are given as \$0.07, \$0.15, and \$0.54/lb, respectively, by a steel producer[133]. Another source cites hot and cold rolled sheet mild steel in various grades and finishes used in the auto industry at \$0.08 to \$0.10/lb. Aluminum sheet and structural stock is estimated at 3-4 times that cost of cold rolled steel[155,156]. Prices for sheet molding compounds of the grades and types of interest to the auto companies are quoted in the range of \$0.40 to \$0.60/lb. Others are considerably higher priced. Relative weights, however, offset to some extent the price differential.

[155] Telephone communication with John R. Newell, Newell Salvage Co., San Antonio, Texas.
[156] Conversation with Frank Vitiello, Southwest Research Institute Machine Shop, San Antonio, Texas.

With aluminum weight approximately one-third that of steel, gage for gage, its price is competitive with steel. Similarly, with plastics weights only one-seventh to one-eighth that of steel, prices are also competitive. To meet strength and stiffness requirements, however, gage for gage substitutions are generally not feasible, so that resultant material costs will be higher. Similarly, HSLA steels are some 15 to 20 percent above mild steels in yield point and can provide about that weight reduction. Resultant material costs, however, will be about 90 percent above costs for currently used materials in like applications, involving approximately 600 lb of a 4200-lb car. Assuming base material cost represents 10 percent of total finished product cost, the resultant increase in material cost will be about 1.3 percent. Significant increases in labor costs are not anticipated. The increase in cost for substitution of aluminum will be of about the same order. Use of plastics and combinations of materials entail costs largely indeterminate at this time because of limited experience with forming, joining, and finishing such materials. Production rates with the suggested materials are another indeterminate factor for the same reason. Unless current rates are maintained or improved, costs will increase.

Several indeterminate possible cost reductions are involved. For one, the scrap value of aluminum is reported to be 0.19/lb as compared to 0.02 for steel[148] and aluminum recycles with less energy expenditure than steel[144]. For another, the overall cost of fabricating some plastics parts in short runs (20,000 to 600,000) is less costly than the same production in steel[137]. And, as previously stated, components formerly composed of multiple steel stampings requiring welding and grinding into a finished single piece are now being molded of plastic as a single piece, finished and ready for painting. In one case, one plastic front end now replaces 15 to 18 metal parts[128]. The net result is reduced labor costs.

TABLE 41. WEIGHTS AND PRICES OF 1973 AUTOMOBILES

Vehicle	Price ($)	Weight (lb)	Price/lb
Impala	4222	4379	0.96
Malibu	3820	3790	1.01
Galaxie	4245	4387	0.97
Gran Torino	3847	3933	0.98
Fury	4151	4075	1.02
Satellite	3759	3720	1.01
Average			0.99

Broadly, new car factory list prices for a considerable time have followed a "dollar per pound" rule of thumb, and prices for 1973 reference full-size vehicles and their intermediate counterparts are very close to this rule, including such options as air-conditioning, power steering, and power brakes. Table 41 indicates this correspondence.

It is believed that, ignoring inflation pressures, prices would tend to adhere to the rule of thumb, however, a minimum increase of 2 to 3 percent is indicated due to more expensive materials.

In summary, the extensive use of alternate materials and construction will result in a cost increase of $150 to $200 for the "full-size" car with a lowered weight of 3800 lb. Reduction of car size within presently demonstrated production technology through a size change (intermediate body style) will also result in a car of 3800 lb, but at present prices this car will be $400 to $500 cheaper than the reference vehicle. Although the consumer may prefer a 1-ft longer car at a higher price, the reduction in cost by using conventional technology could offset to a degree some of the proposed increases due to the refinement of engine technology discussed in earlier sections of this report.

Safety

The current "Intermediate" size vehicle, the first alternative, complies with 1973 Safety Standards and has a service and performance record of 7 to 8 yr.

The second option can be built to comply with 1973 Safety Standards but will require extensive prototype test and development before release to production.

Emissions

Automobiles are in production in the displacement and weight ranges discussed. No further problems are anticipated in meeting emission standards by the reduction in vehicle weight to achieve better economy.

Noise

For the first option, noise characteristics would not be different from present.

Reduced gages and lighter materials in body skin and panels would probably result in flutter and noise characteristics different from current production vehicles, but more annoying than harmful. Such conditions could be corrected as experience developed.

Performance (Acceleration)

For the first alternative, performance would be no different from present production vehicles.

For the full-size, lighter car, acceleration capability could be enhanced by the reduced vehicle mass; however, displacement reductions would be possible so that performance would remain at 1973 levels.

Time Required for Implementation

For the first option, no time lapse would be involved.

For the second, to the usual 3 yr lead time for model change, should be added 1 to 2 yr for development of fabrication procedures, 1 yr for test and development, and 1 yr for field test and determination of public acceptance, or a total of 5 to 6 yr for complete implementation. The current energy situation bears to an indeterminate extent on availability of both plastics and aluminum, plastics because of their dependence on petroleum feedstocks and aluminum because of the enormous electrical demands for bauxite reduction.

Consumer Acceptance

For the first alternative, public acceptance has been established.

For the second, the association of aluminum with aerospace, recreational vehicle, and marine industry would probably enhance consumer acceptance. Conversely, the general public attitude toward plastics in the automotive field connotes "cheapness," which would detract from consumer acceptance of the use of plastics on a large scale. A concentrated public relations effort would be necessary to educate the consumer to the merits, advantages, and acceptance of plastics.

21. ACCESSORY DRIVES

Many accessories, to perform properly at low vehicle and engine speeds, have drive ratios greater than one, i.e., the ratio of accessory speed to engine speed is greater than one. Typical ratios are the following:

Accessory	Ratio
Water pump and fan	1.25
Alternator	3.12
Power steering	1.18
Air pump	1.25
Air conditioning compressor	1.4

When directly coupled by V-belt to the crankshaft, these accessories often operate at speeds higher than is necessary to perform their respective functions. This increased speed of an accessory results in increased horsepower consumption, higher noise levels, and reduced accessory life.

To improve this engine/accessory speed ratio, accessory drives which limit the maximum speed of the accessories and/or reduce the ratio of accessory speed to engine speed can be used.

Accessory drives can consist of one or more of the following types:

(a) Viscous drive with temperature control

(b) Viscous drive with torque limited speed

(c) Gear or pulley change to obtain two or more speeds (Discussed in Air-Conditioning Section)

(d) Hydrostatic transmission

(e) Friction drives speed limited or temperature controlled.

Items (a), (b), (d) and (e) are discussed in this chapter. In general, the effect on fuel economy of each device is quite small.

Present Usage

Fan clutches are presently in widespread use to control and limit the speed of engine fans. In addition, electrically controlled friction clutches are used to engage air-conditioner compressors. These are the only drives, other than direct gear or belt drives, currently used to drive automobile accessories.

Fan Clutches

To adequately cool engines at idle with high ambient temperatures and with air conditioning, large fans with up to seven blades and pitches to 2.5 in. are used. These fans generally are driven at approximately 1.25 times engine speed. These large fans, if run at high speeds, require a large power input (fan horsepower requirements increase as the cube of fan speed). Fortunately, however, the high airflow capacity of these fans is not all needed for cooling. Two methods can be used to "unload" the fan at high engine speed. One method is to use flex-blade fans which have flexible blades that flatten or reduce their pitch at high speed. The pitch reduction tends to limit the fan airflow and, thus, fan power requirements are controlled to acceptable limits. The second method used to "unload" a fan when the engine is running at high speed is to use a fan clutch which reduces the ratio of fan speed to engine speed.

Fan clutches are used extensively on automobiles, especially those cars that are equipped with air-conditioning. Although some friction drive clutches are used, the viscous-drive type dominates. Generally, a temperature sensitive control is used with the viscous drive clutch. This control senses the air temperature in front of the fan and partially disengages the fan when the temperature is less than a specified design value (approximately 140°F).

A viscous fluid shear principle is applied to the fan clutch design in the following manner.

A drive plate totally enclosed within the clutch housing is attached directly to the clutch input shaft (assembled to the water pump shaft and pulley). A predetermined clearance between the drive plate and the inner surfaces of the clutch housing is established at assembly.

The clutch housing and the fan blade assembly are mounted to the input shaft by a sealed bearing and are free to rotate independently of the drive plate and input shaft.

The interior chamber of the clutch housing is filled with a given amount of silicon base oil. Centrifugal force resulting from the rotation of the clutch, coupled with the constant pumping action designed into the unit forces the silicon base oil evenly about the inner surfaces of the clutch in the close clearance or drive area. The drag between the driving and the driven members is thus increased by the presence of the oil, causing the clutch action.

A control valve (operated by a temperature-sensitive bimetal coil or strip in the airstream on the front of the clutch) regulates the amount of oil pumped in or out of the close clearance (drive) area. This action determines the fan speed in relation to the drive pulley and the radiator core airflow temperature (Figure 95).

Effect of Fan Drive on Fuel Economy

Using a viscous-drive type of fan clutch, the power requirement variation between the clutch engaged and disengaged conditions will be appreciable; however, the actual difference during a driving cycle will depend on the cooling requirements (heat load and vehicle speed).

Under the disengaged condition, there is a small measurable difference in power consumption between the declutched solid fan/viscous drive and the flex fan. Water pump and fan power requirement are illustrated in Figure 96; further data are located in Appendix B. The power savings are not appreciable, particularly when compared to the high power requirements of the rigid fan, shown for reference in the same figure. It may be observed that the power savings potential of either fan drive is appreciable when compared to the solid mount fan. It should be emphasized that the rigid fan without viscous drive is not used on factory air-conditioned cars, and that factory air-conditioned cars only were considered during this study. It is recommended that further consideration be given to the application of viscous clutches to nonair-conditioned cars as a method of improving economy.

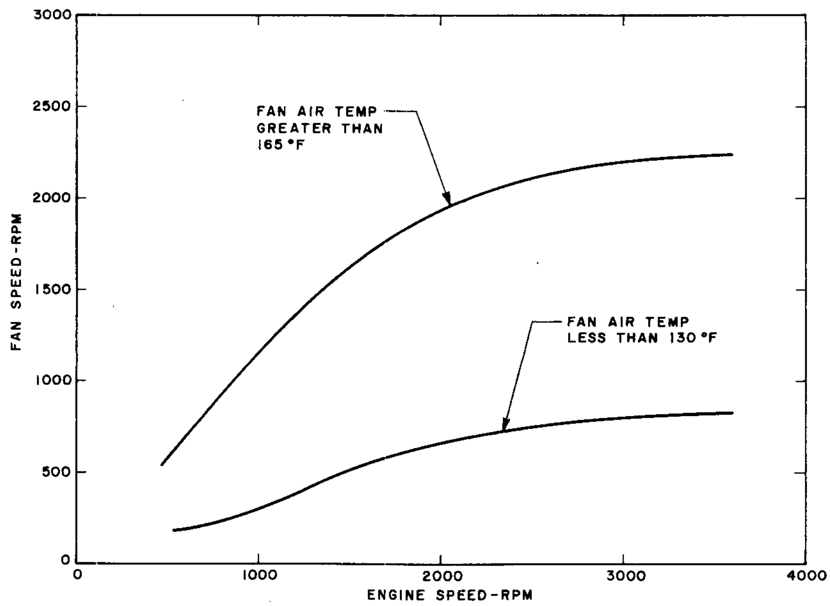


FIGURE 95. FAN CLUTCH OPERATION

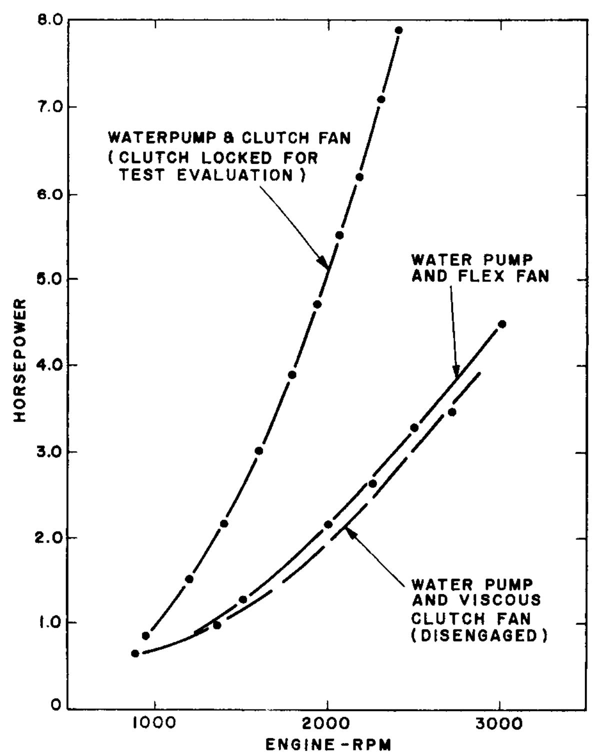


FIGURE 96. POWER REQUIREMENTS–WATER PUMP AND FAN

Since the power savings of a viscous drive rigid fan compared to a flex fan are not large at low engine speeds, no improvement in urban mileage would be measurable. It is expected, however, that road load fuel economy at 70 mph would be increased by about 1 percent.

State of Development

The state-of-the-art of fan clutches is well developed and clutches are factory equipment on most air-conditioned cars.

Demonstration by 1976

Fan clutches have been demonstrated in use.

Reliability

Reliability is well established on cars in service.

Cost

Costs of fan clutches are not a deterrent to their usage; mass production tooling has been developed to assure low production costs.

Safety Standards

Department of Transportation Safety Standards are not compromised.

Emissions

Emissions will not be affected.

Noise

Engine noise is reduced through the use of a fan clutch. In fact, the reduction of engine fan noise was a motivating factor in original development work on fan clutches. If engine speeds are increased, such as by a change to a smaller engine, engine noise during acceleration can be reduced by the use of the viscous drive to limit fan speed. The difference is noise between flex fans and viscous clutch and rigid fan was not investigated.

Performance

Performance will not be appreciably affected.

Time Required for Implementation

Fan clutches are presently a production item, and presumably could be installed on all air-conditioned cars with a minimum of lead time.

Friction Type Speed Limiting Accessory Drives

Development work has been done by one manufacturer on a speed limiting accessory drive which allows slippage of friction-drive plates whenever a present output speed is

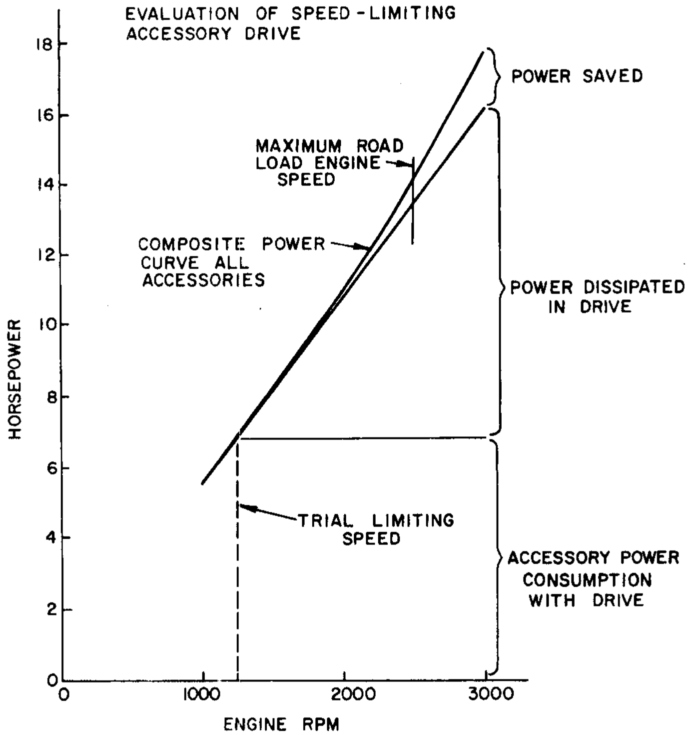


FIGURE 97. EVALUATION OF SPEED-LIMITING ACCESSORY DRIVE

reached.[157] The system was designed to provide an output speed that will provide improved accessory performance at low engine speeds. If the slip speed of the drive is designed to be relatively low, say 1200 rpm, accessory power consumption will be low, but the life of the drive friction surfaces may be short. The slip speed must be chosen to provide satisfactory accessory performance and would probably be chosen to be about 1800 rpm for most vehicles. Tradeoffs between required speed for good accessory performance, power savings, and drive durability are required. Limiting drive speed to excessively low speeds will reduce the capacity of the cooling system and the air-conditioning system.

Power Savings/Fuel Economy Improvement

As stated above, the choice of drive limiting speed is a factor in evaluation of fuel consumption. As a "best" case of fuel economy improvement evaluation, a limiting speed of 1200 rpm was considered. Figure 97 illustrates that the power savings through the use of such a drive are small. This is primarily due to two effects, first the drive slips at a preset speed, but slips at the torque obtained at that speed. The torque requirements of accessories are virtually constant throughout the speed range of 1000 to 2000 rpm (typical of reference vehicle operation); consequently, the power consumption is about the same with or without the limiting speed. Power savings accrue only at higher speeds where accessory torque increases rapidly. The fuel economy benefit would be about 1.5 percent at 70 mph road load. At about 60 mph road load no advantage would exist.

In view of this maximum benefit in comparison with the disadvantage of loss of cooling capability and air-conditioning system capacity, such drives are not recommended for fuel consumption improvements.

State of Development

The technology is available for speed limiting accessory drives. Experimental units have been tested for performance and durability.[158]

Demonstration by 1976

Experimental units have been tested.

[157] Hann, M. M., "Design Considerations when Applying Hydraulic Drives to Vehicles," SAE Paper No. 670740, Society of Automotive Engineers, New York, New York.
[158] "Speed Limiting Accessory Drive," Descriptive Bulletin from Borg Warner, Spring Division.

Reliability

Reliability would be a function of the amount of slip that the drive would have, i.e., low design output speeds would require high slippage at high engine speeds. Reliability of other components such as air-conditioner compressors would be improved because these units would operate at lower speeds.

Cost

Initial cost of the drive has not been determined. Some cost benefits would accrue because of the elimination of some pulleys, fan clutch, steering pump oil cooler, etc.

Safety

Speed limiting drives present no apparent safety hazard.

Emissions

No change in emissions is anticipated through the use of speed limiting drives.

Noise

A reduction in accessory noise will be achieved because high speeds will be eliminated. This is particularly true for the cooling fan.

Performance

No significant change in vehicle acceleration is expected.

Implementation

It is expected that a friction speed limiting clutch could be introduced within 2 yr of its approval by an automobile manufacturer.

Due to the low potential effect on fuel economy, this type of drive is not recommended for further consideration.

Hydrostatic Drive

Current Technology

Hydrostatic transmissions are presently in common usage in garden tractors, riding mowers, and light industrial applications. The hydrostatic transmission offers infinitely variable speed control without shifting gears.

In design, the hydrostatic transmission usually combines a variable displacement pump with a fixed displacement motor. Axial piston pumps are generally used because of the ease in controlling the fluid flow to the motor through a variable pitch swash-plate[157,159]. Other components of a

[159]Moyer, D. W., "A Simple Transmission for a Deluxe Estate Tractor," SAE Paper 660586, Society of Automotive Engineers, New York, New York.

hydrostatic transmission are an auxiliary pump, check valves, pressure regulator, and filter. Total weight of a typical transmission capable of transmitting 6 to 7 hp is approximately 35 lb.

Application to Automobile Accessory Drives

An accessory drive running at a constant speed, say 1500 rpm, would provide good performance of accessories such as air-conditioner compressor, alternator, power steering pump and air pump. Good low speed cooling would be assured by including the cooling fan and water pump. These latter items, however, might require higher operating speeds at high engine speeds combined with high ambient temperatures.

The application of a hydrostatic drive to obtain a constant speed accessory drive has been reviewed.

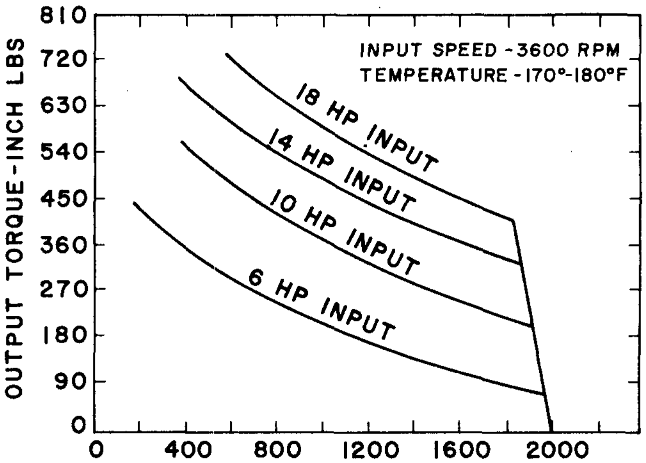


FIGURE 98. OUTPUT TORQUE VERSUS OUTPUT SPEED
MODEL 10 LIGHT DUTY TRANSMISSION

The hydrostatic drive speed control which is normally used to provide an infinitely variable output speed could be used to provide a nearly constant output speed as the input speed (engine speed) varies. An automatic means of regulating the output speed would have to be developed.

Figure 98 shows a performance map of one manufacturer's hydrostatic transmission. Using 1500 rpm as a desirable output speed and an output load requirement of 8 hp (required by the reference vehicle at 1500 engine rpm and including air-conditioner), the output torque would be 336 in. lb. Approximately 12 hp input would be required. This is an output efficiency of 67 percent based on an input speed of 3600 rpm. Assuming the 67-percent efficiency holds for all input speeds (an optimistic assumption), then at all engine speeds, the accessory load would be 12 hp. However, without accessory speed control, the reference vehicle accessory load does not exceed 12 hp until about 2250 rpm.

If 1000 rpm is selected as the accessory speed, then the accessory power required will be 5.3 hp and the transmission input requirement will be approximately 9 hp, an efficiency of 59 percent. The reference vehicles accessory power requirement does not exceed 9 hp until 1700 rpm. Since most of the LA-4 cycle and one-half of the road load fuel consumption calculations are made at less than 1700 rpm, it is apparent that the hydrostatic transmission does not offer any fuel mileage improvement when applied at a constant speed accessory drive.

Evaluation Summary

Reduction in Fuel Economy—The efficiency of a hydrostatic drive will affect any gains made by driving accessories at an adequate but reduced speed. Major breakthroughs in design (especially for high volume, inexpensive units suitable for automotive application) would be necessary before such a system could be considered.

Fuel Consumption—Fuel consumption, by comparison to that of the reference vehicle, is not expected to show an improvement. Mileage calculations were not made because specific data on a suitable hydrostatic drive unit are not available for the range of input speeds of interest. Explicitly, such hardware is not in existence at present.

State of Development—Although hydrostatic drives are fully developed for vehicle propulsion, the application of these drives to accessory requirements would require further development, especially in output speed regulation.

Demonstration by 1976—SwRI has not learned of any hydrostatic accessory drive development work that will be demonstrable by 1976.

Reliability—Reliability would not be a deterring factor. Reliability of hydrostatic drives has been demonstrated on garden tractors and similar vehicles.

Cost—Present hydrostatic units for garden tractor applications cost approximately $75.00 for a 10-hp capacity unit. This cost would be reduced if designed and mass produced for the automotive market. Accessory costs and maintenance might eventually be reduced if operating speeds were reduced and made constant.

Safety—Hydrostatic accessory drives present no apparent safety hazard.

Emissions—Since the engine loading would be increased at low speed due to the inefficiency of the drive, it is expected that some slight increase in gaseous emissions would be observed.

Noise—Hydrostatic drives are noisy. Noise levels of 80 DBa at 50 ft during drive-by tests have been reported. Mounting the drive to isolate this noise from the passenger compartment would be difficult.

Performance—There would be no significant change in vehicle performance.

Time Required for Implementation—If an anticipated fuel savings provided a development incentive, the implementation of a hydrostatic drive would probably be achieved prior to 1980.

22. AIR-CONDITIONING

In this section, both operation and performance of automotive air-conditioners are discussed. Possible modifications to conventional systems, such as constant speed drives and load dependent drives, are considered; and several popular alternative refrigeration schemes are listed. It is concluded that, within the framework of this study, major changes in cooling systems for automobiles are not likely. However, several modifications to the present systems show promise for decreasing the fuel economy penalty associated with air-conditioning.

Air-conditioning has become an accepted feature of the modern automobile; in some sections of the country the majority of new cars are so equipped. It could be argued that the proper approach to fuel economy would be the elimination of air-conditioning as an option, but consumer acceptance of such a move would be minimal. The desire of consumers to be "cool" in warm climates is as valid a consideration as the desire for "warmth" in cold climates. The direct energy penalty of the air conditioning system in comparison with a coolant waste heat heating system is not valid, since in cold ambients, the fuel consumption of the vehicle during normal driving is also significantly increased above that required in warm climates. (See Appendix G.) A reasonable approach would involve optimization of air-conditioning systems to provide comfort while reducing the present fuel economy penalties. Although it is possible that many owners would accept some compromise in air-conditioner performance in order to realize fuel savings, this factor has not been carefully evaluated, and it must be assumed that operation of future systems must result in a level of comfort at least equal to that of the 1973-model year.

Automotive air-conditioning systems operate on the *vapor compression* cycle; a schematic diagram is shown in Figure 99. In operation, a refrigerant (usually R-12, or dichlorodifluoromethane) is transformed from a liquid to a vapor in the *evaporator*. This boiling process absorbs energy from the air entering the passenger compartment of the vehicle. In order to dispose of this excess energy, the pressure and temperature of the refrigerant are raised in the *compressor*, and the refrigerant is allowed to flow into the *condenser*. The condenser is typically located in front of the vehicle radiator; the energy absorbed by the refrigerant in the passenger compartment is transferred to the outside air. During this process, the refrigerant condenses and the resulting liquid is directed back to the evaporator. At the evaporator inlet, the pressure of the refrigerant is reduced by the *expansion valve*, and the cycle is repeated.

A complex control system allows the air-conditioner to operate in the sophisticated manner to which modern consumers have become accustomed. When automotive air-conditioners were first used, the system was turned off or on by the signal from a thermostat located inside the vehicle; the controlled element was a *magnetic clutch* which engaged the compressor. The load changes as a result of compressor cycling were frequently discernable to the operator of the vehicle, and a perceived consumer demand caused the adoption of the *suction throttling valve* as a component of the system in the more expensive vehicles. In some systems controlled by a suction throttling valve, the compressor is engaged by the magnetic clutch in some control positions, such as "defrost," that do not require passenger compartment cooling. The flow of refrigerant is continuous; if the thermostat indicates that there is no demand for passenger compartment cooling at a given moment, then the refrigerant is evaporated outside of the duct containing passenger compartment air. This technique greatly decreases the load variations caused by compressor cycling, but the result is a considerable quantity of unnecessary cooling. It should be noted that some vehicles are equipped with elaborate control systems which require only the specification of a desired temperature by the operator. Basic system operation with these control systems is identical to that outlined above.

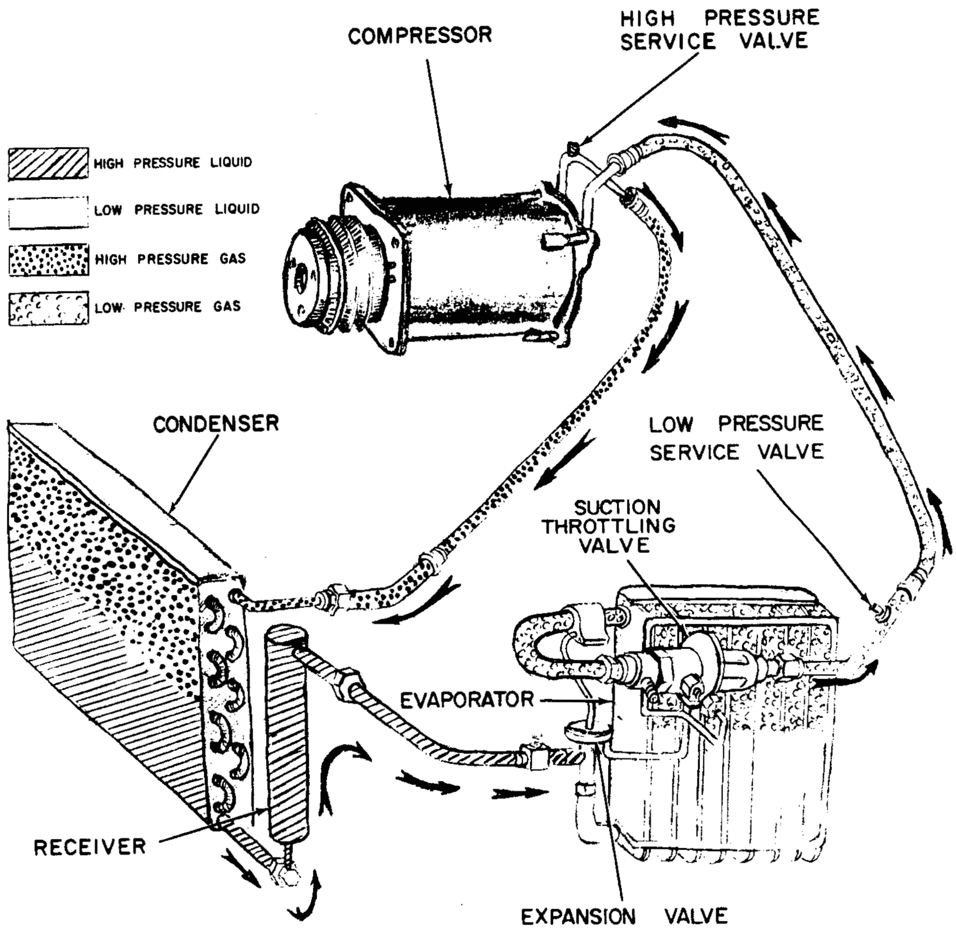


FIGURE 99. AIR-CONDITIONING SYSTEM

It should also be observed that the air-conditioning system provides a function other than cooling; moist air passing through the evaporator is dehumidified. Some vehicles are equipped with control systems which allow dehumidification at moderate ambient temperatures; incoming air is cooled below the dew point and then reheated to an acceptable level for occupant comfort. The refrigeration system, therefore, may be in operation even when outside temperatures are not excessive.

Air-Conditioner Performance

Automotive air-conditioners are designed to provide acceptable comfort levels while the vehicles are idling or moving in stop-and-go traffic. In addition, the systems are expected to begin producing a substantial flow of cool air immediately after the engine is started, and vehicles which have been soaking in the sun are expected to cool rapidly. It should be observed that some unique conditions prevail; in a vehicle an individual may preferentially direct a stream of chilled air directly at his body, although it is unlikely that this condition would be considered comfortable in a commercial or residential setting.

The design load for an automobile air-conditioning system at idle, after a comfortable interior temperature has been attained, is about 11,000 Btu/hr; during "pulldown," or attainment of a comfortable condition, the load is about 23,000 Btu/hr[160]. At highway speed, the heat transfer coefficient at the outer surface of the vehicle may be ten times that for a motionless vehicle. In addition, the infiltration rate at highway speed is larger than that for a stationary vehicle; the net result is an approximate doubling of the load. The continuous highway load is, therefore, about equal in magnitude to the "pulldown" load; in usual refrigeration practice, the load would be regarded as about two tons. Although the above values for refrigeration load were estimated from values reported in the literature and not obtained from direct measurements, they appear to be appropriate for considerations of contemporary vehicles.

The amount of work required to produce a given quantity of refrigeration, say one ton, is strongly dependent upon the characteristics of the refrigeration system. Some large, stationary air-conditioning systems approach the theoretical efficiency, in terms of horsepower per ton, dictated by the refrigerant and the operating temperatures; but in automotive units the horsepower requirement is much higher because compromises have been made to assure large capacity with small size and variable speed capability. Available information indicates that current automotive systems operate in the range from 2 to 5 hp/ton[161]. In terms of refrigeration efficiency, a 1-hp/ton rating would correspond to a coefficient of performance of about 4.7. By comparison, the automotive system would operate at a coefficient of performance of 1 to 2.

Measurements have been made of the actual power required to operate the compressor of an automotive air-conditioning system; the data are shown in Figure 100. In addition, Figure 101 depicts the change in fuel economy that can be attributed to air-conditioning. It should be noted, however, that these data were obtained at a particular system operating condition. The input power to an air-conditioner depends upon the thermal load applied to the evaporator and upon the ambient conditions; a change in either load or ambient temperature would alter the power curve. This feature of the system has particular implications with regard to proposed test cycles for fuel economy. If the test is to be conducted with the air-conditioner in operation, then the test specification must include details of the operating conditions. For example, a reproducible test would require designation of the ambient temperature, blower speed, and refrigerant temperature and pressure at the evaporator and condenser. Fuel economy numbers for several air-conditioned vehicles can be compared only if the tests are conducted under the same system load conditions.

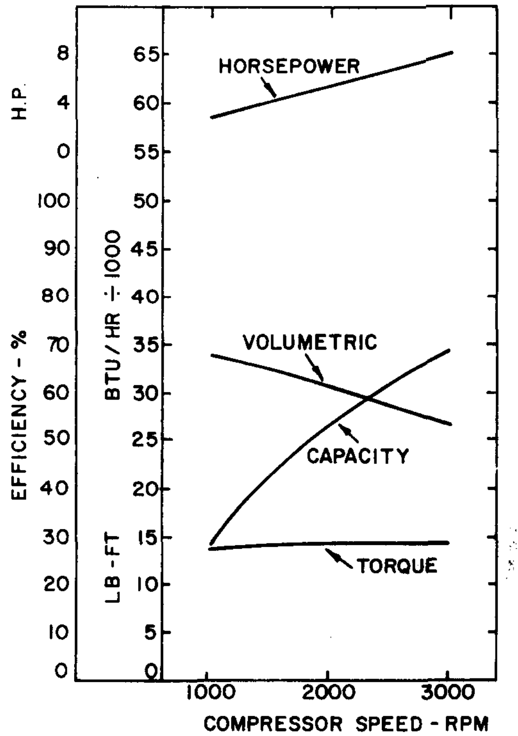


FIGURE 100. AIR-CONDITIONING COMPRESSOR CHARACTERISTICS

The capacity of an automotive air-conditioner is a function of the compressor speed. A system designed to provide for idle load and "pulldown" can have a capacity of about 40,000 Btu/hr (more than three tons) at a compressor speed of 3000 rpm (Figure 102). Since this capacity is considerably in excess of the load at highway speed, it represents an area in which economy could be realized by better

[160] Zahn, Willard R., "Factors Influencing Automotive Air Conditioner Evaporator Optimization," SAE Paper 690131, 1969.
[161] Akerman, Joseph R., "Automotive Air Conditioning Systems with Absorption Refrigeration," SAE Paper 710037.

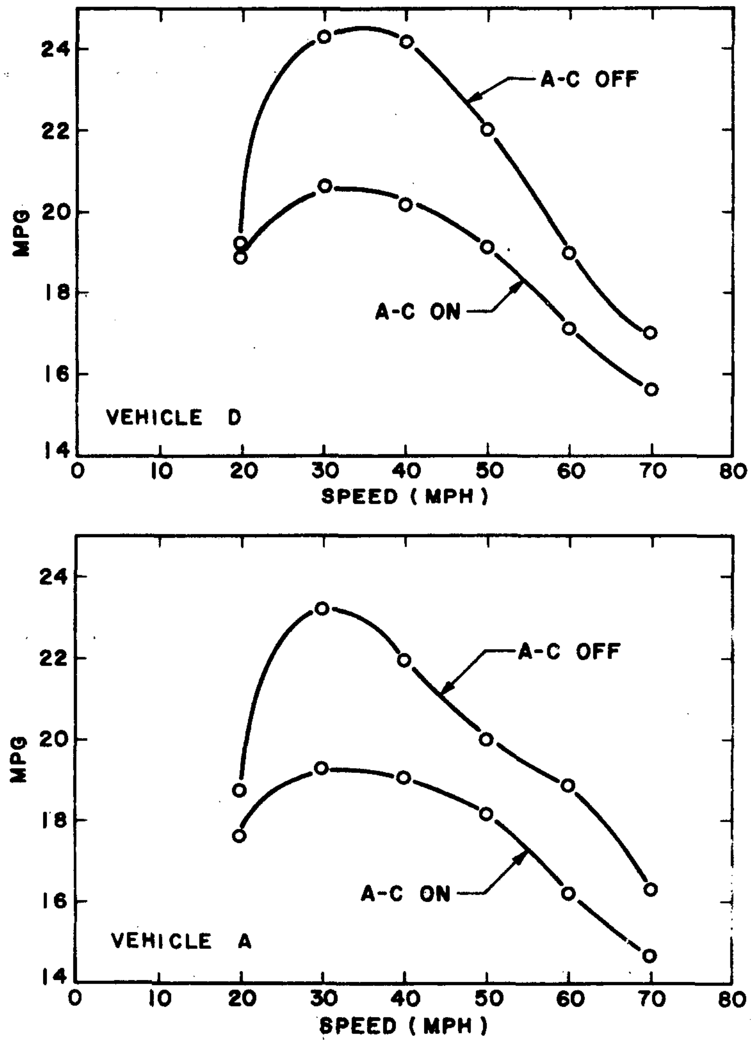


FIGURE 101. ROAD-LOAD FUEL ECONOMY

matching of the compressor speed to the load. It should also be mentioned that the standard compressor is designed for high volumetric efficiency (~80 percent) at low speed; this efficiency deteriorates to about 50 percent at highway speed[162]. Since the difference in efficiency may be attributed to the necessity for variable speed (and high speed) operation, a further advantage of more precise load matching is apparent.

Numerous alternatives can be proposed that will fulfill the basic function of passenger compartment cooling while offering economy of operation by comparison with conventional systems. Although many of the possible systems are impractical at the present time, their potential attractiveness and advancing state-of-the-art require that they be reconsidered periodically. The following

[162] Moore, G. H., Jr. and K. B. Bjorkman, "The Automotive Air Conditioning Compressor–A Design Challenge," ASHRAE Journal, May 1964.

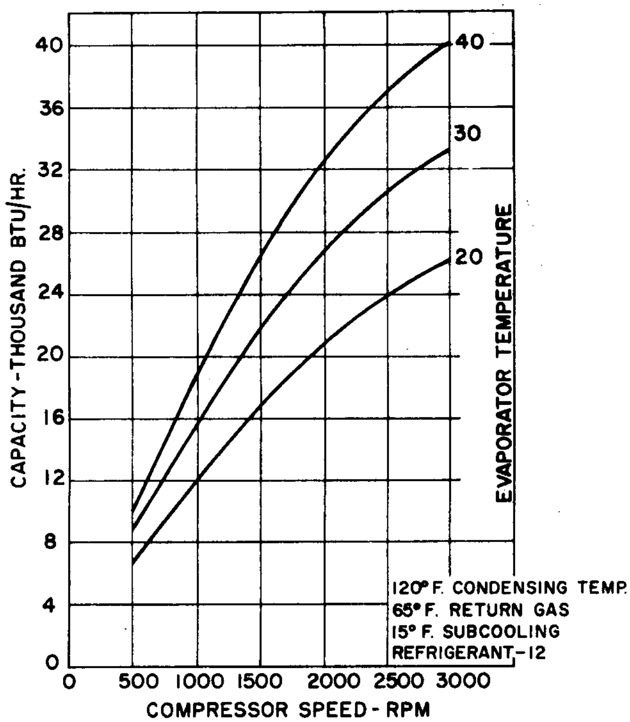


FIGURE 102. COMPRESSOR CAPACITY

sections contain a summary of a number of possible automotive air-conditioning systems; each is considered in the context of technological practicality and fuel economy.

Vapor Compression System

The conventional vapor compression automotive refrigeration system could be modified in several ways to produce an enhancement of operating economy. In the following paragraphs the implications of various changes are discussed.

Elimination of Suction Throttling Valve

In the systems which employ suction throttling valves, the compressor operates whenever the operator selects any form of climate control other than fresh air ventilation. Refrigerant that is supplied to the evaporator in excess of the amount required by the thermostat is evaporated outside the passenger compartment. The use of a thermostatically controlled magnetic clutch allows compressor operation only when cooling is actually required; depending upon operating and environmental conditions, a substantial saving should be possible. The thermostatically controlled clutch is still used on some systems installed in less expensive car lines. Clutch engagement is occasionally noticeable to the operator, but the minor annoyance can, in most cases, be offset by benefits in the economy area. A more serious problem might occur in vehicles equipped with cruise controls; the response of those systems to sudden changes in load would require careful evaluation.

Change in Refrigerants

The capacity of a refrigeration system can be increased by changing the refrigerant. For example, Refrigerant 500 is sometimes substituted for Refrigerant 12 as a means of increasing the capacity of marginal stationary systems. The performance increase, however, is not sufficient to warrant this action for automotive systems. Ammonia could be used as a refrigerant with excellent results, but it is not compatible with copper, and it is thought to be more dangerous than Refrigerant 12. Although Refrigerant 12 has a reputation for being completely inert and nontoxic, some possible detrimental effects have recently been reported[163].

Changes in Compressor Efficiency

As shown in Table 42, the volumetric efficiency of a compressor drops dramatically as the speed increases. This is in part due to the design, since inexpensive reed valves are used. However, the

[163] Taylor, G. J., and W. S. Harris, "Cardiac Toxicity of Aerosol Propellants," Journal American Medical Association, 214:1, p 81, 1970.

TABLE 42. COMPRESSION VOLUMETRIC EFFICIENCY

Compression rpm	Car mph (1.15 to 1 drive ratio)	Typical Conditions pressure (psig)		Compression ratio	Volumetric efficiency (%)
		Suction	Discharge		
1200	20	30	135	3.0	78
1200	20	15	195	7.0	65
1925	40	10	165	7.2	57
2730	60	6	155	8.1	50

compressor is also required to operate over an extremely wide speed range, and high volumetric efficiency would be difficult to maintain with any valve system. Because of the changes in rear axle ratios during the past few years, the difference in engine speed between idle and highway cruise has decreased; this fact should allow more efficient compressor design. If necessary, a speed limiting control might be used to disconnect the compressor at high speeds; such a control would serve the dual purpose of preserving the compressor and making more power available when very high speeds were required.

It should also be possible to design a variable displacement compressor using many of the features of current products. Many compressors presently in service are of the swash plate design; hydraulic motors having variable displacement and this same design have been manufactured for years. If the displacement could be varied as a function of the load on the system, then a high volumetric efficiency could be maintained.

Constant Speed Drive

It has been suggested that overall refrigeration system efficiency could be increased through the use of a constant speed accessory drive. In this configuration, the compressor would operate at a single speed independent of engine speed, and the compressor efficiency could be optimized. Selection of the appropriate speed is difficult, however, since compressor capacity is proportional to speed, and the system load is not a direct function of speed. System loads are almost equivalent at highway speeds and during "pulldown" at idle; a reduced load occurs between these conditions. Adequate operation at a single constant speed, or even at two separate constant speeds, would be difficult to provide. It should also be noted, parenthetically, that a good constant speed drive has not been located; those that are available are characterized by either high cost or low efficiency.

A constant-speed drive system could be designed if some provision were included for refrigerant storage. During periods of reduced demand, the system would store liquid refrigerant in an appropriate vessel; this accumulated liquid could be used to meet "pulldown" loads. There are several disadvantages; the stored refrigerant could be depleted by several short trips, some loads at high speeds could probably not be accommodated, and the storage vessel would be large and heavy. Ammonia would be, by far, the best refrigerant for this application, and its toxic and corrosive properties have previously been enumerated.

Variable Speed, Load Dependent Drive

It is possible to envision an air-conditioning system driven by a power source which is sensitive to the demand on the system. For example, an electrically driven compressor could be thermostatically controlled to operate at a speed, and therefore a refrigerant capacity appropriate to the load on the system. Improvement in compressor efficiency could be expected, since the compressor would not be required to tolerate the very high speeds and shock loads associated with belt drive, magnetic clutch operation. The load on the vehicle electrical system would be substantial during operation of the system at high loads; some increase in electrical system capacity would be required. However, the load on the vehicle would amount only to the weight of the system unless the occupants expressed a demand for cooling.

Other Refrigeration Systems

Although the vapor compression cycle is used more extensively than any other means of cooling, there are several other techniques which might be considered for automotive use. The following sections contain a brief description of various alternative cooling methods.

Thermoelectric Cooling

A thermoelectric cooling device is characterized by junctions between dissimilar metals; upon application of current to the circuit, a low temperature location and a high temperature location may be observed. At present, such devices operate at very low efficiencies, and the quantity of hardware required to satisfy an automotive load would be enormous.

Constant Volume Heating

A device capable of heating a refrigerant at constant volume could be used to operate a refrigeration system with heat that would otherwise be wasted in the exhaust. However, such devices are bulky and not readily adaptable to flow processes. It is possible that some developments from Stirling Cycle heat exchanger research may be useful in the future, but adequate hardware is not presently available.

Air Cycle Cooling

Air cycle, or reversed Brayton cycle cooling is commonly found on aircraft equipped with gas turbine engines. The system requires large quantities of compressed air; in an automotive installation, this air could be supplied by an engine-driven compressor or by cylinders of the engine dedicated to the compression function. The compressed air is directed through an aftercooler and a turbine; the work output of the turbine could be used to drive other engine accessories. The low temperature air at the turbine exhaust can be blended with ambient air and supplied to the passenger compartment. With the proper control system, this arrangement offers excellent opportunities for load matching, and it could easily serve as a heater for the passenger compartment. The refrigerant compressor, evaporator, heater core, and fan could be eliminated; the condenser would be retained as an aftercooler, and the air compressor and turbine would be added. It should be noted that the quantity of air required is substantial; calculations indicate that even at somewhat reduced cooling system capacity, about one-fourth of the engine displacement should be used in the form of a double-acting compressor to supply the air. At least one system of this type, consisting of a compressor and turbine on a single shaft, is being tested, but specific results are not available[164].

Absorption System

From an overall viewpoint, the absorption cooling cycle is quite similar to the vapor compression system. The difference may be ascribed to the use of a chemical process and heating instead

[164] "Automotive News," July 30, 1973, p 16.

of mechanical compression as a means of raising the pressure of the refrigerant. The following steps outline the process:

(a) Refrigerant vapor leaving the evaporator is dissolved in a liquid in the *absorber*.

(b) The pressure of the liquid is increased by a *pump*. This requires less input work than the compression of vapor.

(c) The refrigerant vapor is driven out of solution through the application of heat in the *generator*.

(d) The vapor is routed to a condenser which is virtually identical to that for a conventional vapor compression cycle.

On the whole, the absorption system is much less efficient than the vapor compression cycle. However, a majority of the energy input occurs in the form of heat supplied to the generator. Thus, if a large quantity of thermal energy is available at low cost, the absorption system is economically attractive. Since an automobile exhaust system is a source of thermal energy which is usually wasted, the absorption system has been considered repeatedly for this application.

A thorough study of the thermodynamic feasibility of absorption cycles for automobiles has been reported[61]. The results indicated that an average engine would support only about one ton of absorption refrigeration at idle and low speed; this quantity would not adequately satisfy the pull-down load. There are further disadvantages; absorption systems typically require large tanks and considerable quantities of liquid. Liquid storage in the confined space of a passenger car would present a problem, and liquid sloshing in the tanks might prove to be a nuisance or a safety hazard. The only acceptable refrigerant at the present time is ammonia, which has been discussed previously and eliminated on the basis of safety. Furthermore, the removal of energy from the exhaust may not be compatible with exhaust emission controls.

The absorption refrigeration cycle, particularly in a system that incorporated a refrigerant storage capability for transient response, offers attractive possibilities for mobile environmental control. However, a considerable quantity of development effort would be required before a competitive system could be realized.

Evaluation—Air-Conditioning

It would be possible to decrease fuel consumption by eliminating air-conditioning as an option for passenger cars, but this would not be consistent with the huge demand for air-conditioning which exists across the country as a whole. There will always be some expenditure, or some decrease in economy, when air-conditioning systems are operating. However, it should be possible to identify design changes in the air-conditioning systems which will decrease the present penalty.

Identification of Improvements

Sophisticated techniques, such as thermoelectric cooling, do not appear to be sufficiently advanced at the present time for use in automotive systems. However, several modifications to existing systems can be envisioned which would increase efficiency without appreciable degradation of performance. The basic alteration involves control; the system can be disconnected when it is not specifically required to provide passenger compartment cooling. This alteration would compromise the dehumidification capability of the system, which is important in some parts of the country. If the system controls include the dehumidification function, then an effort should be made to communicate to the operator the cost penalty associated with system operation in this mode.

It would appear that considerable improvement is possible in the area of compressor efficiency. The reed valves employed on current designs are not amenable to operation over a wide speed range; volumetric efficiency could be enhanced by more attention to this area. Also, as mentioned previously, the swash-plate design seems applicable to a variable displacement capability; if the displacement were controlled by load then increased system efficiency should result.

One of the most desirable systems would be composed of the conventional components with a compressor drive which was not dependent upon engine speed. For example, if the compressor was connected to an electric motor, then it could be operated at a speed appropriate to the required capacity regardless of the engine speed. Independent control could be achieved, and the compressor design could be revised to take advantage of lower speeds and more uniform loading. Actually, such an arrangement would require substantial changes in the vehicle electrical system, since the loads (1 to 2 kW) are far in excess of any existing loads except the starting motor. As a compromise, a two-speed belt drive could be used. This arrangement would allow one engine speed to compressor speed ratio for low loads and another for high loads; the system controls would be required to select the appropriate ratio. At least one manufacturer is presently developing such a drive, but exhaustive data are not yet available. Another feature that should allow enhanced compressor performance would be a limit on compressor maximum speed. The air-conditioning system could be disconnected at speeds greater than, say, 3000 rpm; this would make more engine power available for emergency use and remove some of the constraints associated with compressor design.

Reduction in Fuel Consumption

Fuel consumption calculations were performed according to the procedure outlined previously. The calculations were precisely the same as those for the baseline vehicle (4500 lb, 350 CID) except that an extra 5.25 hp was added to the accessory loads at 2000 rpm; this figure was obtained from the measured values of compressor horsepower. The fuel economy penalty associated with this increased load is shown in Table 43. These figures are representative of the effect of an air-conditioning system on the fuel economy for the baseline vehicle. It is highly improbable that any air-conditioning technique would allow fuel consumption figures lower than those for the reference vehicle; the intent of the modifications proposed herein is the reduction of the penalties cited in Table 43.

TABLE 43. AIR-CONDITIONING PENALTY

Driving mode	Percent decrease in mpg
LA-4	4.8
Road load	8.5
Average	6.1

Stage of Development

The two-speed accessory drive previously mentioned is allegedly in the development stage in the laboratories of one of the automobile manufacturers. Other systems, such as absorption cycles and thermoelectric devices, are in mature stages of development for certain applications, but none are appropriate for automotive use for the reasons outlined previously.

The variable displacement compressor would be quite similar in design to existing hydraulic pumps and motors, which have been in production for quite some time. Improvements in compressor volumetric efficiency could be obtained by redesign of the valve system.

Status by 1976

The two-speed drive can easily be demonstrated by 1976 if, in fact, demonstration units have not already been assembled. The variable displacement compressor, because of its similarity to existing devices, should also be capable of demonstration by 1976.

More exotic systems, such as the absorption cycle, would require substantial development work prior to demonstration in mobile applications.

Reliability

The two-speed drive is mechanically more complex than a simple magnetic clutch; this additional complexity will require more attention to component selection and manufacturing. However, the use of such a drive would relieve some of the most severe stresses on the compressor, and fewer problems should occur with that portion of the system.

Cost

Considering the drive and the compressor as a single unit, the hardware, maintenance, and repair costs should be equivalent to those observed for current systems. The increased costs for the drive should be offset by reduced costs for the compressor. If, on the other hand, a serious attempt is made to increase the efficiency of the compressor, then the system cost might be increased by $10 to $15 per unit.

Safety

The air-conditioning system has no direct effect on the safety standards, although it could be argued that a connection exists between driver comfort and safety.

If a change to a substance such as ammonia for the refrigerant is contemplated, then serious points concerning safety are raised. The halocarbon refrigerants have the advantage of being accepted by the industry, but many questions regarding refrigerant safety have yet to be resolved.

Emissions

The presence of an air-conditioning system has little effect on engine emissions. It is remotely possible that the operation of the system could change the operating characteristics of some very small, underpowered vehicle on the LA-4 cycle, but such vehicles must be considered marginal in any case.

Noise

Air-conditioning systems generally have the effect of reducing noise levels insofar as vehicle occupants are concerned because the vehicle is operated with the windows closed. The noise emitted by a properly operating compressor is inconsequential. The main source of additional noise from air-conditioned vehicles is the larger cooling fan required by the system; this noise is not regarded as excessive in existing designs.

Vehicle Performance

The trend in American vehicle design has been toward vehicles which can support air-conditioning systems with no noticeable change in performance level. If an effort is made to increase fuel economy through, for example, reduction of engine size, then the presence of an air-conditioning system will become a more important factor in vehicle design. It may be practical to supply vehicles with different engines depending upon whether or not they are air-conditioned, and the performance of vehicles equipped with retrofit or aftermarket air-conditioning systems may be degraded.

Implementation Time

The design changes suggested in this discussion, two speed drives, variable displacement compressors, and valve improvements, could be implemented by the 1980-model year. Accelerated implementation to, say, the 1978-model year would be possible at increased cost to the manufacturers.

The absorption system does show promise for vehicle application, but an exhaustive development and design program would be required prior to implementation. The 1980-model year would be a good target for demonstration of such a system.

At least one firm is presently engaged in a demonstration program for air cycle cooling[164]. Depending upon the results, the system may be scheduled for production by the 1980-model year.

Consumer Acceptance

The air-conditioning system is perhaps unique in that it is installed in a vehicle purely for consumer comfort; it serves no purpose associated with mobility of the vehicle. However, in many parts of the country, air-conditioning is regarded as essential, and consumers who have become accustomed to it would not take deprivation lightly. The vehicle operator uses the environmental control system daily; any attempt to conserve fuel by degrading air-conditioning system performance would probably result in more consumer complaints than would a similar change to another vehicle system. It is quite likely that many drivers will continue to insist on high performance air-conditioning, and effort should be directed toward minimization of the economy penalty.

23. COOLING SYSTEM

Current Practice

Engines are characterized by thermal efficiencies no higher than 25 percent; this figure implies that less than one-fourth of the energy contained in the fuel is converted into work during the combustion process. The remainder of the energy liberated during combustion is transported away from the engine by: (1) the cooling system; (2) the exhaust; and (3) direct heat transfer processes (conduction, convection, radiation). The quantity of energy transported by the cooling system can be on the order of 40 percent of the energy content of the fuel[1]. This portion of the energy liberated during combustion flows through the metal parts of the engine to either the cooling jacket or the outside surface; with some small engines it is possible to extend the outside surface, with fins, by an amount sufficient to accommodate the required heat transfer. Typical automotive engines, however, utilize a liquid cooling system to assist the energy transfer.

An automotive cooling system contains several components which play a vital role during operation. *Flow passages* are formed in the major engine components during the casting process; these passages allow close proximity between the coolant and critical engine parts such as exhaust valve seats. A *thermostat*, located at the point where the coolant leaves the engine block, is used to regulate the coolant flow and, thereby, control engine temperature. A compact, liquid-to-gas, convective heat exchanger, commonly termed the radiator, is used to transfer the energy liberated within the engine to the atmosphere. Airflow through the radiator is maintained by a *fan*, and coolant circulation is maintained by a *water pump*. Typically, the fan and the pump impeller are powered by the engine from the same belt-driven hub.

The traditional coolant for engines, by virtue of availability and heat transfer characteristics, is water. However, water freezes at a temperature common to automotive use, is incompatible with some common materials, and will not function as a lubricant for the coolant pump. All of these inadequacies can be alleviated by the addition of another substance to the water; in recent years ethylene glycol has achieved virtually universal acceptance for this purpose. The usual mixture, recommended by most manufacturers for all-season use, requires 40 to 50 percent ethylene glycol in the system.

As engines became larger and horsepower outputs increased over a period of years, the loads on vehicle cooling systems increased while available space decreased. More recently, emission controls have required still higher loads and higher engine operating temperatures. Designers were faced with demands to increase the heat transfer capability of the system, particularly the radiator, and the universally accepted solution was temperature increase through pressurization. Operation of the system at a pressure higher than atmospheric provides a higher boiling point for the coolant according to Figure 103; therefore, the engine can operate at higher temperature and the radiator exhibits increased effectiveness as a result of the increased temperature difference between the coolant and ambient.

It should be noted that the use of pressurized cooling systems has caused somewhat restrictive attitudes to develop within the industry and the public. For example, boiling in the cooling system is synonymous with catastrophic failure. This is generally true for a pressurized system, since boiling either causes or results from a loss of coolant. The usual result of continued operation is complete loss of coolant and engine failure. Despite the adverse image, however, there are features of the boiling process which can be used to advantage in a properly designed system.

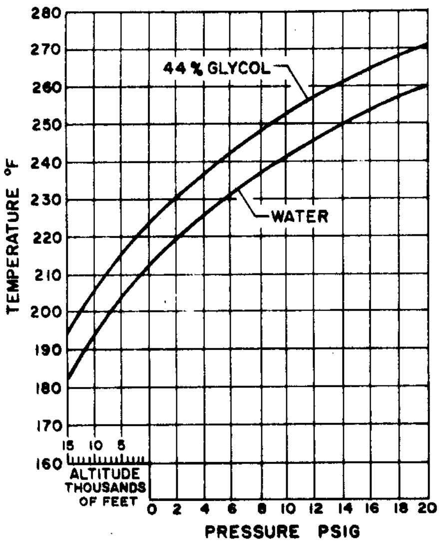


FIGURE 103. BOILING POINT OF WATER AND 44-PERCENT GLYCOL AT VARIOUS PRESSURES[165]

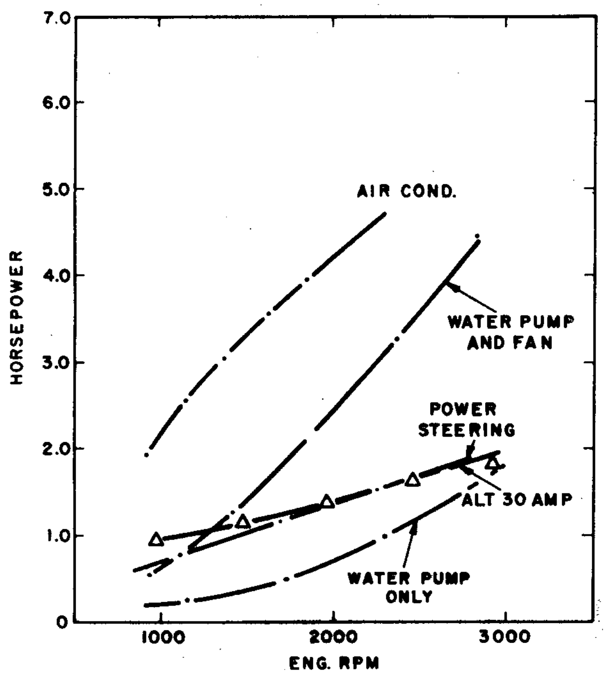


FIGURE 104. ACCESSORY POWER REQUIREMENTS

The power requirements associated with the cooling system have been obtained by direct measurement; typical values for the coolant pump and fan combined are presented in Figure 104. The road load power (70 mph) is on the order of 2 to 2.5 hp for the standard system; the associated coolant flow rate is on the order of 40 to 80 gal/min[165]. In general, cooling systems which consume 4 to 5 percent of gross engine power are considered good.

Ebullient Cooling Systems

As an alternative to the system described in the preceding paragraphs, it is possible to cool the engine by taking advantage of the properties of a boiling liquid. This method is far from new; it was used in some of the earliest automobiles and has been re-evaluated at intervals throughout automotive history[166]. Basically, an ebullient cooling system employs a pure substance or an azeotropic (fixed boiling point) mixture as the coolant. The coolant is supplied to the engine from a standpipe; the geometric arrangement is such that the level of liquid is maintained above the top of the engine block. After an initial warmup period, cooling is affected by local boiling of the coolant with attendant bubble formation. The bubbles rise to a separator directly above the engine; liquid coolant is routed back to the inlet to the engine block, and coolant vapor passes to a condenser. The condenser is physically similar to the conventional radiator; in order to maintain a reasonable size, a fan would be rquired to stimulate airflow. The condensed coolant is returned to the standpipe either by gravity flow or by a small pump for supply to the engine block. A diagram of the basic system is shown in Figure 105.

The ebullient cooling system has several distinct advantages by comparison with conventional systems. First, the system

[165]Beatenbough, P. K., "Engine Cooling Systems for Motor Trucks," SAE – SP-284, 1966.
[166]Herfurth, W. R., "Twenty Years Fleet Experience with Engine Temperature Control," SAE – SP-194, 1961.

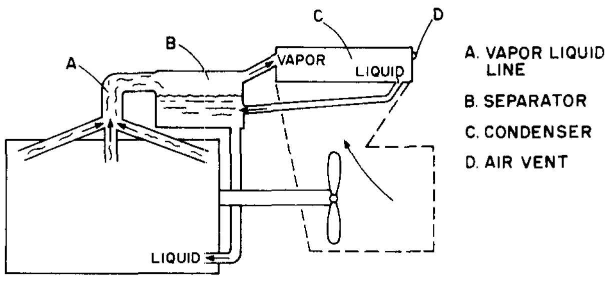


FIGURE 105. SCHEMATIC OF EBULLIENT COOLING SYSTEM WITHOUT A CONDENSATE RETURN PUMP

need not be pressurized; engine operating temperature is specified by the composition of the coolant. Operation at atmospheric pressure lowers the stress on cooling system components, particularly flexible hoses, and the atmospheric pressure system is much more tolerant of small leaks.

In addition, the ebullient system minimizes the temperature differences in the engine block; the nature of the boiling process dictates that the transition from liquid to vapor will occur at the same temperature throughout the engine. For the ebullient system, a given quantity of energy transferred to the coolant supplies the heat of vaporization (latent heat) rather than increasing the coolant temperature. The boiling process allows energy to be transferred at lower values of the temperature difference between metal and coolant; the result is increased uniformity of the metal temperatures throughout the engine.

It should also be noted that the driving force for coolant circulation in the ebullient system is purely thermal; circulation does not require a pump. The circulation of coolant, therefore, continues after the engine is shutdown; this feature prevents the excessive temperatures which can occur when engines are suddenly stopped after operation at heavy loads.

Several investigators have converted conventional truck and automobile cooling systems to ebullient systems; the comparisons are discussed in the literature[166,167]. In most cases, these tests have been conducted with conventional engine blocks and radiators, and good results have been reported. This indicates that it is possible to convert existing engines to ebullient cooling without extensive redesign of the passages in the engine block. Furthermore, the existing radiator will serve as an adequate condenser, although an optimum system would employ a somewhat different design.

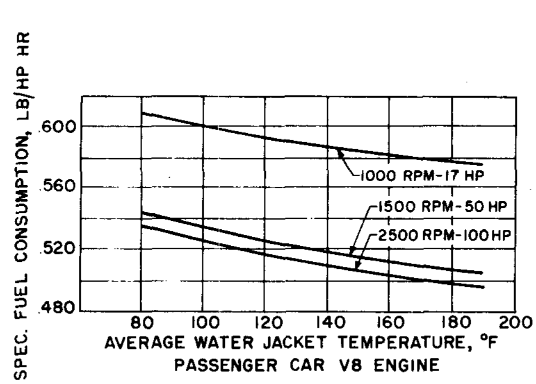


FIGURE 106. EFFECT OF WATER JACKET TEMPERATURE ON SPECIFIC FUEL CONSUMPTION

Effect of Cooling System on Fuel Economy

Several areas of possible improvement in cooling system design offer advantages from the standpoint of fuel economy. The three most prominent alterations, which are consistent with the use of an ebullient cooling system, are as follows:

(1) Operation of the engine at higher temperatures

(2) Reduction of warmup time

(3) Reduction of fan and water pump power

The effect of engine operating temperature on fuel economy is well documented as a consequence of basic thermodynamics, and documentation for operating engines is available from several sources[167,168,169]. Figure 106, from Reference 169, is representative. Engine cooling is required

[167]Tacchella, A. A., J. S. Fawcett, and A. N. Anderson, "Dual-Circuit Ebullition Cooling for Automotive Engines," SAE Paper 887B, August 1964.
[168]Brabetz, J. C., and D. S. Pike, "Engines Like to be Warm," SAE Paper 891A, August 1964.
[169]Kazlauskas, P. P., "Coolant Temperature Effects on Engine Life and Performance," SAE – SP-194, 1961.

for the purpose of avoiding structural problems in metal engine parts and preventing breakdown of lubricants; the traditional operating coolant temperatures have evolved by virtue of the necessity to prevent boiling in the conventional cooling system. Modern engines and lubricants are quite capable of operation at coolant temperatures in excess of 200°F. Such temperatures may be readily achieved with ebullient cooling systems by proper selection of the coolant mixture, whereas elaborate pressure systems are required for high temperature operation with conventional cooling systems. At present, the most readily available azeotropic mixture is "Dowtherm 209" having a boiling point of 209°F. It should be noted, however, that suitable fluids for higher temperature have yet to be developed. Thermostats of the reference vehicles are set at about 200°F; however, these could also be changed to achieve an equivalent 209° to 210°F or higher.[170]

The time required for an engine to reach operating temperature appears to have a pronounced effect on fuel consumption; economy is generally poor for cold engines. The primary factor in warmup economy is the extent of vaporization of the fuel, and a thorough discussion of this aspect is located elsewhere in this report. Figure 107, taken from Reference 1, indicates that considerable time may be involved in the attainment of fully warmed-up economy after a vehicle is started. It should be noted that Figure 107 pertains to the vehicle as a whole and not simply the engine. Furthermore, although the figure may not be quantitatively accurate for 1973-model year vehicles, it does illustrate the effect of vehicle warmup on fuel economy. The thermal response of the cylinder block plays an important role in the warming process; the thermostat found in conventional cooling systems is used to restrict coolant flow when the temperature is below some specified value. However, most vehicle cooling systems are equipped with a thermostat bypass which allows some coolant circulation while the engine is cold. This bypass is not necessary with the ebullient cooling system; coolant flow occurs as a result of thermal effects only, and the result is a more rapid rise in cylinder block temperature. Since pumping power would no longer be expended, water passage could be made smaller to reduce the mass of water contained in the block for more rapid warmup.

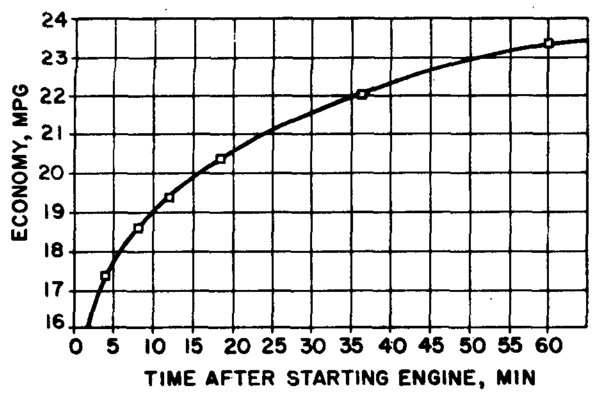


FIGURE 107. WARMUP TIME FOR AUTOMOBILE IN 20°F ATMOSPHERE AT A CONSTANT SPEED OF 30 MPH

It has previously been pointed out that the fan and coolant pump consume 4 to 5 percent of the gross engine power output. In the ebullient system, the main coolant circulation pump is eliminated, although a small pump (probably an intermittently used electric pump) is used in most systems to change the elevation of the liquid coolant. With careful design, maximum advantage may be obtained from gravitational and thermal effects, and the pumping requirements can be minimized. If the condenser (radiator) can be adequately sized, then the fan can be eliminated from the ebullient system. However, cost and styling considerations will probably dictate a configuration similar to that in current use, and a fan will be required for high cooling system loads. Instead of continuous operation as a result of belt drive from the engine, it should be possible to operate the fan on demand from the cooling system. Demand operation could be achieved with an appropriate sensor and an electric clutch, similar to the air-conditioning system clutch; the fan should be allowed to free-wheel when not engaged.

[170]Geschelin, J., "Dow Chemical Fills Cooling Gap," *Automotive Industries*, August 15, 1970.

Evaluation—Ebullient Cooling System

Reduction in Fuel Consumption—The use of an ebullient cooling system can reduce fuel consumption by reducing the power required for the coolant pump and fan, by allowing operation at somewhat higher temperatures, and by reducing warmup time. With proper system design, a reduction of 4 percent in gross engine horsepower should be possible.

Fuel Consumption—Steady State and Road Load—Reduction of warmup time and operation at higher temperatures are not amenable to direct analysis by the standard method adopted for fuel economy calculations in this report. However, some experiments were conducted; Figure 108 shows the temperature response at two locations in the cooling system of a test vehicle during an LA-4 cycle. It can be observed that the engine warms rather quickly with the conventional cooling system; decreasing the warmup time would require changing the slope of the ascending portion of the curve. The ebullient system could accomplish this to the extent that bypass flow during warmup is eliminated.

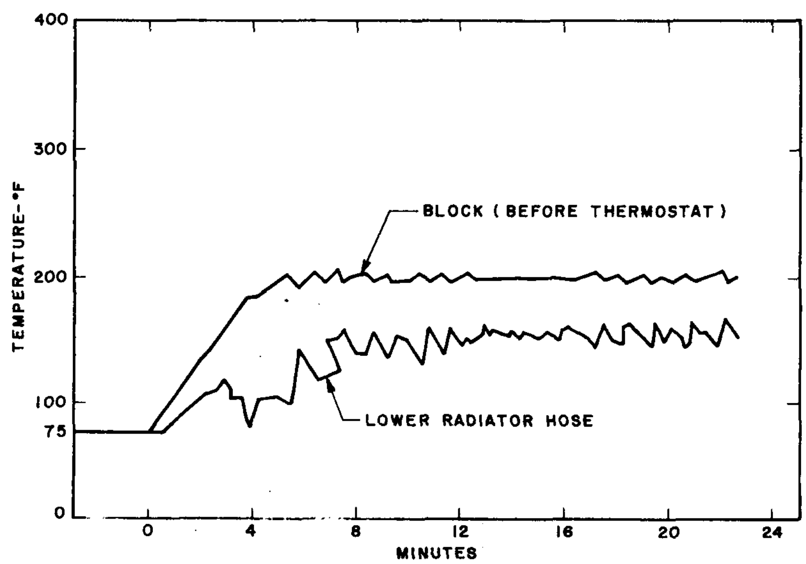


FIGURE 108. TEMPERATURE RESPONSE–LA-4 CYCLE

Calculations were performed using the standard fuel economy calculation procedure for a reduction in reference vehicle accessory horsepower by an amount equal to the fan and water pump loads. The result of this reduction in load was an increase in fuel economy amounting to about 3 percent. It should be observed, however, that vehicle operation during the LA-4 cycle and road loads occurs at relatively light engine loads. Specific engine map data will be required in order to accurately assess the impact of the cooling system on fuel economy.

Stage of Development—The technology required for ebullient cooling systems is available; experimental systems have been constructed using standard components. A commitment to this form of engine cooling would allow redesign of the engine block coolant passages for optimum effect, but investigations cited in the literature indicate that conversion of most existing engines is feasible.

Demonstration by 1976—Ebullient cooling systems have been demonstrated, and the results have been reported.

Reliability—The ebullient system should be more reliable than current pressurized systems. A small leak in a pressurized system results in accelerated loss of coolant; loss of coolant from a system at atmospheric pressure would occur at a much lower rate. Furthermore, the system is virtually exempt from engine failure as a result of belt or pump failure; thermally induced circulation can maintain an appreciable coolant flow in a properly designed system. The ebullient system should also contribute to the reliability of other engine components, since the thermally induced flow allows coolant circulation after the engine is stopped. The danger of component damage due to overheat after engine shutdown is, therefore, minimized.

Cost—The hardware cost of the ebullient system should not exceed that of the conventional system. The radiator is retained, and a tank or standpipe is added. However, the use of low pressure hose and the elimination of the thermostat and water pump should provide cost reductions. The cost of the small condensate pump and the magnetic clutch and sensor to activate the fan should offset the savings due to the thermostat and water pump. The coolant solutions, which may or may not be mixed with water, should be available at approximately the same cost as conventional antifreeze liquid.

The maintenance costs for the system should be equal to those for pressurized systems; system cleaning and coolant replacement would occur at the same intervals.

The repair costs should not exceed those for conventional cooling systems. Hoses should be less expensive, and the other components, such as tanks, will not be appreciably different from those in current use.

Safety—There should be no conflict between the 1973 Safety Standards and the ebullient cooling system. In fact, because of the tendency toward coolant loss inherent in pressurized systems, the ebullient systems should be more amenable to adequate protection.

In addition, the ebullient system should be less hazardous than conventional systems in ordinary service. With existing pressurized cooling systems, violent boiling and steam emission can occur if the radiator cap is removed from a vehicle which has been stopped after a period of operation. Since the ebullient system would operate at atmospheric pressure, the danger to operators and service personnel should be minimized.

Emissions—The effect of the cooling system on contaminant emissions may be observed in the areas of engine operating temperature and engine warmup time. During the recent model years, when emphasis on emission control has been intense, the loads on vehicle cooling systems have increased steadily. The increased loads have occurred as a result of retarded spark used for emission control, and higher coolant temperatures and system pressures have been observed. With ebullient cooling systems, the coolant temperature is fixed by the composition of the coolant mixture, and increased loads cause an increase in the quantity of vapor generated. The condenser (radiator) must have a capacity appropriate to the maximum expected load, but within the design limits the coolant temperature will not vary after the warmup period. This feature should allow more precise adjustment of other emission control systems.

The time required for an engine to attain operating temperature is important to emission control technology; fuel vaporization in carburetion systems and threshold temperatures in control devices are related to engine temperature. In most conventional cooling systems, a thermostat bypass allows circulation of coolant during warmup; in the ebullient system, the coolant is passive until thermal effects induce a flow. More rapid attainment of design operating temperature should be possible with the ebullient system, and this trait has been observed in reports of experiments. Decreased warmup time should allow better control of cold start emissions.

Noise—There should be no difference between the sound levels of ebullient systems by comparison with conventional systems. The main source of noise is the cooling system fan, and a fan will be present in the proposed ebullient system. Operation of the fan only at high loads, however, should provide a reduction in noise level during most vehicle operating conditions. There is occasionally a sound output associated with the boiling process, but this should be effectively muffled by the walls of the cylinder block.

Performance (Acceleration)—No degradation of acceleration capability is expected with the ebullient system. In fact, acceleration performance during periods when the fan is not operating should be improved.

Time Required for Implementation—Studies reported in the literature indicate that conversion to ebullient cooling should be possible for most existing engines. It should be possible to test various current production vehicles to determine the applicability of ebullient cooling; models which show promise might be equipped with ebullient systems if they are produced in subsequent model years. Satisfactory history of use in customer service would motivate optimized design of engine block coolant passages for other models.

Consumer Acceptance—The cooling system is one segment of the vehicle with which most operators have some contact; cooling system failures, which usually occur without warning, are responsible for a large percentage of incompleted trips and roadside incidents. Any increase in cooling system reliability would be desirable.

Generally, the heater for the vehicle interior uses engine coolant as a source of thermal energy. An auxillary electric pump would be required to convey 209°F liquid from the block to the heater. The net effect of the heater function would not be altered by the use of an ebullient system. Furthermore, a decrease in engine warmup time would allow faster heating of the vehicle interior; this would be a desirable feature from the consumer standpoint. The auxiliary pump would have a very small capacity by comparison with conventional coolant circulation pumps, and the impact on overall system economy should be small.

Manufacturers typically offer, as options, heavy-duty cooling system for service at high speeds or heavy loads such as trailer towing. This option would be available with the ebullient system, and the increased system capacity would appear in the form of a larger condenser.

One of the main reasons for the delay in adoption of ebullient cooling systems is probably the impact of cooling system design on vehicle styling. System capacity is not a primary issue; some experiments show that the quantity of coolant can be reduced with the ebullient system, and the system volume is not significantly greater than that of a conventional system. However, optimum system design would require location of major components above the engine. This region is presently crowded, and the required compromises between engineering design and vehicle design have probably been regarded as unacceptable.