AN ANALYTICAL APPROACH TO AUTOMOTIVE FUEL ECONOMY

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ABSTRACT

Experimental data and some fundamental considerations are used to derive mathematical and graphical models of engine behavior relative to fuel consumption. Dimensional analysis of the torque converter permits consolidation of much available information in a compact and useful form. A graphical representation is introduced to represent road friction.

Using raw data from engine tests, the power flow through the drive train is followed from engine to rear wheels and the resulting steady-state economy is computed by a combination of analytical and graphical techniques. The results indicate the need for refinement of method or a new approach.

The vehicle is now analyzed from the road to the engine, and the apparently complex analysis of the torque converter viewed from the output side is shown to be surprisingly easy to relate to the functional dependences already derived. The results show the expected characteristics and suggest some graphical representations for optimum economy conditions. These in turn show a remarkable peculiarity which leads to a study of non steady state driving technique. It is found that this approach gives superior results over a sizable portion of the road condition spectrum, and eventually four different techniques are advanced as optimum in different regions of operation.

OBJECT

The object of this thesis research has been to derive a simple, yet fundamentally sound method of analysis of the fuel economy potential of a given motor vehicle, and its applications to normal and competitive driving.

INTRODUCTION

Although it may be said that interest in the subject of fuel economy is as widespread as the distribution of automobiles, the fundamental relationships which govern the topic for a modern passenger automobile have not been explored to any great extent. For example, the writer's interest in the field came about as a result of an assignment as engineer on the Studebaker-Packard entries in the 1955 Mobilgas Economy Run. It was quite evident that the Studebaker-Packard team possessed the best available techniques in the competition (as evidenced by the sweepstakes victory for the second year in a row, and the near accomplishment of an actual miles-per-gallon victory over a car with a 25% smaller engine). This superiority over several competing groups of technical advisers was achieved mainly by a trial and error experimental program, which was satisfactory in spite of a rather severe (by most standards) spread of the results. In addition, the time and expense involved in the tests precluded evaluation of all but the most elementary techniques of competitive driving.

The beginning of the realization of the value of primarily analytical techniques in the subject came during a routine analysis of proposed engine sizes during the fall of the previous year. However, since the competition group was soon disbanded, a fuller study of the methods did not occur until this thesis subject was proposed and approved.

THE BASIC APPROACH

In devising a scheme to provide a sound method of analysis, it was decided to attempt to represent by empirical equations, wherever possible, each of the major devices involved, with due regard for the law of physics and engineering. In other words the equations must make sense experimentally and theoretically.

SYMBOLS USED

a = acceleration, miles per hour per second

d = deceleration, miles per hour per second

Δp = manifold depression below atmosphere, inches of mercury

E_c = converter efficiency, percent

K = size factor, $N_1/\sqrt{T_1}$

K_o = output size factor, $N_o/\sqrt{T_o}$

N_{co} = converter output speed, rpm

N_e = engine speed

N_i = converter input speed, rpm

N_o = transmission output speed, rpm

P_f = friction drag, lbs.

P_g = grade load, lbs.

P_n = net thrust, lbs.

P = total load, lbs.

Q_a = fuel used accelerating, gallons

Q_d = fuel used decelerating, gallons

Q = total fuel used, gallons

S_a = distance travelled accelerating, miles

S_d = distance travelled decelerating, miles

S = total distance travelled, miles

T_{co} = converter output torque, lb-ft

T_e = engine torque, lb-ft

T_i = input torque, lb-ft

T_o = transmission output torque, lb-ft

SYMBOLS USED (Cont'd)

V	=	car speed, miles per hour
W	=	car weight, pounds
α	=	angle of grade, radians
B. M. E. P.	=	brake mean effective pressure, lb/in^2
B. S. F. C.	=	brake specific fuel consumption, lb/bhp-hr
M. P. G.	=	miles per gallon
R. P. M.	=	revolutions per minute

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I. ANALYSIS OF THE VEHICLE

THE ENGINE

The usable information for this portion of the vehicle is contained in the part-throttle characteristics for the engine (Test 18 of the General Motors Engine Test Code). These curves (see Fig. A-1, Appendix A) show torque, brake specific fuel consumption, fuel rate, intake manifold depression, and "observed miles per gallon" (assuming no transmission slip), all as a function of percent load at eleven different engine speeds. Nine other parameters are also plotted, but are of less interest in this derivation. The curves used are for the 1956 Chevrolet with four-barrel carburetor.

Figure 1 is a cross-plot of fuel rate vs. intake manifold depression for several engine speeds. The region of interest is limited to that between 8 and 22 inches of mercury manifold vacuum (Fig. 1). It will be shown that this is the economical operating region.

Superimposed on the diagram (Fig. 1) is the family of curves represented by the equation: $Q = .00062\, N\, (29 - \Delta p)$. These are seen to represent the fuel flow fairly well, particularly at speeds of 2000 rpm and above. The equation of the curves has significance as follows: the engine is a pump with a throttled inlet; the fluid being pumped is air to which a constant percentage of fuel has been added. Thus the parameters of fuel-air ratio and volumetric efficiency are present in the equations as constants, and the only independent variables are speed and manifold depression.

Figure 2 shows torque as a function of manifold depression and speed. It is apparent that above 1200 rpm, the torque is practically independent of speed, and can be shown as a function of manifold depression only as in Figure 3. This suggests that the torque be represented by the equation $T = 10\ (23.5 - \Delta p)$. The equation indicates that the engine will supply only its own friction requirements (zero net torque) at 23.5" of manifold vacuum and 235 lb-ft of net torque at zero depression. In actual fact slightly over 22" of vacuum is the zero-output point at 1600 rpm, and the maximum torque is just below 230 lb-ft at about 1" of vacuum.

With these expressions for fuel flow and torque, an equation for brake specific fuel consumption can be derived (see Appendix A). Figure 4 shows the shape of this curve. It agrees remarkably well with an approximate curve for a 7 to 1 compression ratio engine as reported on pages 289 and 290 of reference 1, and is also in good agreement with the BSFC curves given for the Chevrolet engine at part throttle. In general it is not necessary to derive such a curve when a graphical representation is available, but if physical meaning is to be preserved and an empirical equation is desired, the procedure is quite reasonable.

As mentioned previously, the equation used to derive the BSFC relationship were valid only in a limited region of operation. However, another method of graphical analysis is available which is somewhat more general. This is the construction of a so-called performance map, which shows lines of constant brake specific fuel consumption on a torque vs. speed basis. Figure 5 shows such a map constructed from data for the Chevrolet engine. The near independence of speed at low torque is readily apparent, except for speeds below 1600 rpm and the region near 80 lb-ft at 2000 rpm. The latter disturbance seems to represent an error of one test point (40% load) on the 2000 rpm part throttle sheet of the engine test data, since road tests with the vehicle showed that no such irregularity existed at 2000 rpm (about 45 mph). This suggests an arbitrary revision of the map, as in Figure 6, which is more representative of normal internal combustion engines, except that the Chevrolet shows more than ordinary low speed sensitivity, possibly due to considerable spark retard for detonation control. The performance map represents a better approximation of the engine with regard to accuracy, but it is not easily expressed algebraically and is limited in its usefulness.

The approximate equation for fuel flow has a form which suggests a rather quick method of approximating the miles per gallon. As shown in Appendix A, the expression for fuel economy does not involve either car or engine speed directly. By substituting the two representative values of 3% slip for light load and 10% slip for fairly heavy loading, the fuel economy can be approximated rather easily as in Fig. 7. Thus a first approximation of instantaneous economy is available with no more instrumentation than a vacuum gauge. Actually, such gauges, calibrated in miles per gallon as well as vacuum are sold in auto speciality stores. While originally intended for standard (no slip) transmission cars, the error introduced with an automatic transmission is not excessive in relation to other factors.

If the BSFC-torque curve is replotted on a BMEP basis, it can be used with any automotive engine of approximately 8 to 1 compression ratio. The performance map, however, is not directly applicable to any but very similar engines. Thus for a general method, it is necessary to use a representative BSFC-BMEP relation if more specific data are not available.

THE TRANSMISSION

The use of an automatic transmission with torque converter introduces the only real complication in an otherwise straightforward numerical analysis.

As shown in Figure A-2 of Appendix A, transmission characteristics are generally plotted showing input speed, torque ratio, and efficiency as functions of output speed at a given throttle setting. In order to use such information for analysis, it would be necessary to have several such curves from which large and complicated cross plots could be made for interpolation. For example, curves obtained at 1/4, 1/2, 3/4 and full throttle would, when cross plotted, define the region from 1/4 to full throttle quite well, although with considerable complexity. This and the absence of information at very light loads makes the approach quite useless.

Studies by Föttinger and Spannhake of several years ago suggest a more promising method, and an extension of their theories permits a great simplification of the problem. (Ref. 2) By dimensional analysis it has been predicted, and experiments have proved that the torque ratio and efficiency of a given torque converter are functions of the speed ratio N_o/N_i, and of no other parameter save Reynolds number to a very limited extent. An extension of the theory leads to the conclusion that another useful parameter can be defined which is also a function of speed ratio. This can be called a "size factor" and is given by dividing the input speed by the square root of input torque. Then the entire performance of a torque converter or fluid coupling can be reduced to one simple set of curves.

Derivation of the simplified curves from the normal type shown requires considerable reasoning and computation, as will be shown here.

It is desirable to separate the effects of the torque converter from the remainder of the transmission. Since the transmission has torque loss but no slip, and the converter in coupling range has slip but no torque loss, it is possible to attribute each effect to one or the other. The only additional assumption which is necessary is that the converter seal friction can be neglected. To perform the necessary computations, it is convenient to put the information in tabular form. Thus the following quantities are tabulated for each test point on the original curve set: gear ratio, output speed, input speed, input torque, indicated efficiency, and indicated torque ratio. Computations begin by finding the square root of the input torque. The input speed (3) is divided by this to give the size factor (8). The output speed (2) is divided by the gear ratio (1) to give converter output speed (9) and this is in turn divided by input speed (3) to give the speed ratio (10).

Figures 8 and 9 show size factor as a function of speed ratio for a total of nearly 90 test points involving gear ratios of 1.82 and 1.00, speeds from 80 to over 4000 rpm, and torques from 38 to 248 pound feet. Clearly the size factor is indeed a unique function of speed ratio.

The total friction and slippage losses (11) as a percentage of the total power are obtained by subtracting the percent indicated efficiency (7) from 100% in the coupling region only (speed ratio above .875). The coupling slip (12) is found in the same manner by subtracting N_o/N_i (10) from 1.000. If the coupling loss is now subtracted from the total loss, the remaining percentage is the loss through the transmission proper (13). It can be converted to torque loss (14) by multiplying by input torque (4).

Figure 10 shows friction torque loss in the transmission as a function of output speed, load, and gear ratio. It can be concluded that in direct gear (1.00 ratio) the friction depends on speed alone, and a reasonable representation of the characteristic can be extrapolated to low speeds.* In low gear (1.82 ratio) the friction depends on load in the gearset, but can again be extrapolated reasonably well.

Using mainly the extrapolated portions of the transmission friction curves, a reasonable transmission torque loss may be assigned to each point in the converter region (N_o/N_i less than .875). The indicated output torque (15) can be obtained by multiplying input torque (4) by indicated torque ratio (8), and

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* A slight error is introduced here since a portion of the transmission friction is due to the front oil pump which rotates at engine speed.

the sum of these gives the output torque for a frictionless transmission (16). Dividing by gear ratio gives the output torque at the converter (17), and dividing this by input torque provides the desired torque ratio value (18). The efficiency is then given by torque ratio x speed ratio.

The computations have been carried out for 1/4, 1/2, and full throttle conditions, both for 1.00 and 1.82 gear ratios. The results are shown in Figures 11 and 12, and a complete set of curves for the Power Glide Torque Converter is shown in Figure 13. These three simple curves contain all the information that is ever necessary in performance analysis, and the efficiency curve could be omitted, since it is derived from the torque ratio curve.

ROAD FRICTION

Representation of road friction by an empirical equation is completely arbitrary, due to the variations caused by such diverse elements as temperature, road surface, tire pressure, brake drag, etc. It is the practice of some people in the industry to use an equation of the form: friction force = C_1 x weight + C_2 x $(\text{velocity})^2$, but the writer's experience is that an exponential expression gives results more in agreement with road tests. For example, Figure 14 shows the test results for a 1955 Chevrolet sedan and the logical exponential curve which fits the test points best. To this curve has been added the transmission friction at each speed from Figure 10, resulting in the upper curve which will be used to represent road friction. As a rule of thumb, the friction can be reduced 2% for each 10 degree rise in temperature from 30°F, where the test was run.

II. COMPUTATIONAL TECHNIQUES

FIRST METHOD

With the foregoing information relative to engine, transmission, and road friction, it is now possible to compute the theoretical fuel economy for a variety of road conditions. The most general steady-state situation always will involve level road or grade. This, then, seems to be one logical field for study.

While specific fuel consumption curves were available for the engine at several speeds, these curves were based on no more than 7 test points each, with only 4 or 5 of the points in the economical operating region. It appeared desirable to avoid an interpolation of these curves if possible, since the shape was somewhat questionable (see Fig. 5). Therefore only the actual engine test points are used in this computation. Again a tabular form is most convenient, and the quantities taken from the engine test points are: speed, manifold depression, input torque, and indicated miles per gallon. The size factor is first found by getting the square root of torque and dividing this into the input speed. Then the speed ratio and torque ratio are found on Figure 13. Car speed and gross thrust are then computed, and the net thrust obtained by subtracting friction load from the gross thrust at the indicated speed. Grade ability (sin α) is then net thrust divided by car weight. The actual miles per gallon figure is easily obtained by multiplying the indicated value by the speed ratio, since the original figure is based on a speed ratio of 1.00 (no slip).

Plots are now made of: grade ability, road speed, and fuel economy as functions of manifold vacuum at each engine speed, Figures 15, 16, 17 and 18.

On Figure 15, a horizontal line is drawn at the desired grade ability value, and the necessary manifold vacuum at each engine speed is recorded. These values are applied to the fuel economy and road speed curves and the results cross plotted to show fuel economy as a function of steady speed on any grade (Fig. 19). The resulting curves, based on 4 to 6 computed points each do not tell much about what is happening. Specifically, the clutch point of the converter, which should cause a notch in most such curves is not evident. Reviewing the steps which led to these results, the most revealing information seems to lie in Figure 15, where some extensive interpolations have been made. The curves would be expected to resemble somewhat the torque ratio curve of the converter, as indeed they do, except for the rather obvious peculiarity of the 2000 rpm curve at 9 1/2" vacuum both in Figures 15 and 18 (see previous comment in engine section). It appears that interpolation of engine data was only postponed for one computational step, then performed where curve shape was even more dubious. Even if this were changed, however, the method would still supply only one computed point for each of the 4 to 6 useful engine speeds and can thus give only a vague idea of the desired curve. It is possible to apply the performance map information to this method at the expense of a considerable increase in complication, but another approach seems to offer more promise.

SECOND METHOD

If car speed and grade are chosen, these define the necessary torque and speed at the transmission output, through application of Figure 14 and the component of car weight acting downgrade, W sin α. At this point the solution of the computational problem seems to be a trial-and-error determination of the engine operating conditions which satisfy the output parameters. This would lead to more complication than in the first method, but the difficulty can be eliminated by a still further extension of the concept of a size factor, as follows: Define first an "output size factor", K_o *given as* $N_o/\sqrt{T_o}$, *then substitute for these their* equivalents in terms of input quantities and the dimensionless torque and speed ratios. In this manner (see Appendix A) the new parameter K_o is found to be another unique function of speed ratio and has been so plotted in Figures 20 and 21. With this information it is now possible to define quickly all operating parameters from the speed and grade quanties under steady-state conditions. The economy can be computed for the exact road conditions desired without multiple computations and interpolations. Again, the tabular form simplifies computations, which have been carried out for several grades and plotted in Figure 22. The dotted line above the 5% curve shows the economy available on a 5% grade if the coupling is locked out, and indicates the two basic effects of the coupling. First is the economy loss due to slip and second the shift of the most economical operating point to a 10 mph higher speed.

These curves indicate that for each percent grade there is one operating point which results in maximum economy, and that at about 7% grade, there are two such points, one in the converter region (lower speed) and one in the coupling region. Above 7% grade, the optimum point is always in the converter range, while at lesser grades it is always in the coupling region. This can be shown graphically by selecting the optimum speeds from Figure 22 and plotting these against the grades to which they apply, as in Figure 23, or by selecting maximum economy points and plotting against grade as in Figure 24. Thus, where economy is the only consideration, Figure 23 defines completely the technique to be used, and Figure 24 shows the optimum results obtainable.

III. EXTENSION OF THE METHODS

CLOSED THROTTLE TECHNIQUE

It is a little known, but well established fact that the fuel used by carbureted internal combustion engine is essentially independent of speed when the throttle is closed. (Ref. 3) This fact was determined experimentally on the test car and the fuel rate was found to be approximately .4 gallons/hour. It was also found that the car would descend a 5% grade in gear with closed throttle at a steady speed of 35 mph. This means that during such conditions, a fuel economy of 35 ÷ .4 or 87.5 miles per gallon is being realized, which is quite startling at first. This led to a calculation of a hypothetical situation wherein an equal distance was travelled up and down on 5% grades at 35 mph. The figures show an average economy of 23.3 miles per gallon while on a level road at the same speed only 22.7 miles per gallon is obtainable. This phenomenon, also little known but fairly well established results from using the engine is a more efficient region and storing up potential energy which is later used for propulsion.

The preceding makes the coasting ability of the vehicle of considerable interest, whence an approximate calculation of retarding force has been made and plotted in Figure 25. The engine friction curve has been determined from 3 downhill terminal speed runs by subtracting road friction from total apparent friction at low speeds in low gear, and calculating the equivalent engine friction at higher speeds in high gear. The resulting approximate terminal coasting speeds are plotted in Figure 26, which also shows the curve for optimum open-throttle speeds in the downhill region. The resulting economy is plotted for closed and optimum open throttle conditions in Figure 27. Thus it can be seen from Figures 23 and 24 that optimum economy is achieved by different simple techniques in the three regions as follows:

Closed throttle on downgrades steeper than 2.2%.

Open throttle in coupling range from -2.2% to + 7% grade.

Open throttle in converter range on grades steeper than 7%.

SURGE TECHNIQUE

It would seem that if one could simulate the effect of a hill while driving, it might be possible to achieve the greater economy which is normally realized in alternate climb and descent of grades. A limiting technique can be imagined where the vehicle is accelerated to a speed of $V + \Delta V$, then allowed to coast to a speed of $V - \Delta V$. Thus the acceleration could take place in an efficient region, and the deceleration with closed throttle would achieve the substantial instantaneous economies shown in the previous section. For a practical application, we can assume a variation of ± 5 mph from a given speed, and compute a fuel economy for alternate acceleration and deceleration. To make the calculations reasonably practical, it will be assumed that acceleration is a constant over the speed range. This makes it possible to use the elementary integrated equation of motion,

$$S = \frac{V_2^2 - V_1^2}{2a}$$

instead of making a 10 or 20 interval numerical integration. It can be shown that this approximation will give slightly optimistic results.

The computations have been made for median speeds of 30, 40 and 50 mph at progressively varying engine torques, and the results plotted in Figure 28. The fact that the optimum torque for each speed is about 120 pound-feet seems logical because this is the region of maximum engine efficiency. What seems most interesting is that all three curves show a higher economy at their peaks than at steady throttle.

Since the optimum acceleration torque seems to be about 120 pound-feet, the analysis was extended to 2% and 4% downgrades using 121 pound-feet of engine torque (it is convenient to work with perfect squares). The results are plotted in Figure 29, with closed throttle and steady throttle results shown for comparison. The surge technique shows a slight advantage (1/2 MPG) on level road, a significant gain (4 MPG) on a 2% downgrade, and a 30 MPG advantage on a 4% downgrade when compared with steady-throttle technique, but the 4% downgrade surge economy is inferior to closed-throttle coasting. It then appears that optimum fuel economy is attained by four different techniques in four regions:

Grades above 7%; steady throttle, converter range.

Level to 7%; steady throttle, coupling range.

Level to 3% down, surging, with acceleration at 120 lb-ft.

Steeper than 3% down, closed throttle coasting.

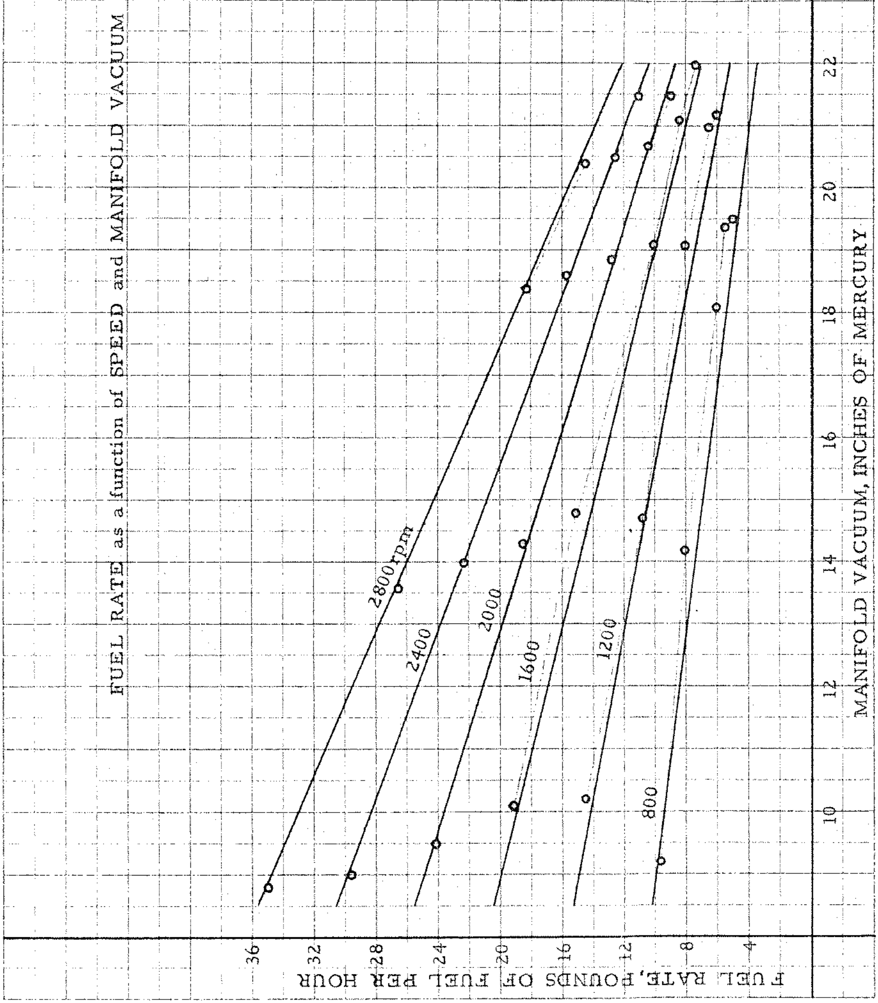
TIME CONSIDERATIONS

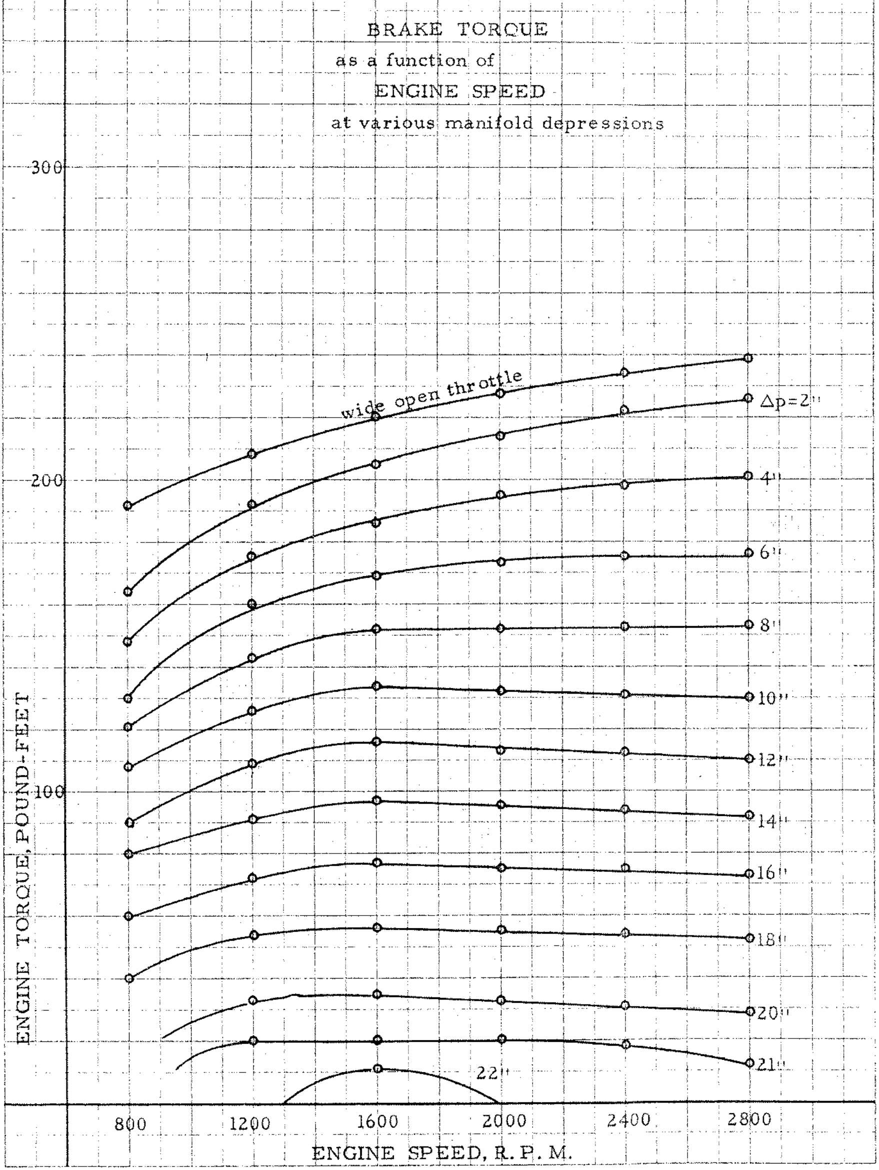
All the conclusions regarding optimum economy have been drawn with the assumption that time was not a factor. In the specific instance of the Mobilgas Economy Run, the course must be completed in a given time, and the conclusions must be revised to take this into account. If the Economy potential can be expressed as a straight-line function of speed on any given grade, it can be shown by elementary calculus that maximum economy with limited time is achieved with speeds predicted from a cubic equation. Similarly it can be demonstrated that it is easier to make a trial-and-error solution of the equations by using the curves already derived. Such solutions indicate that the economy is substantially independent of uphill speed over a large range, but small advantages can be realized by careful analysis.

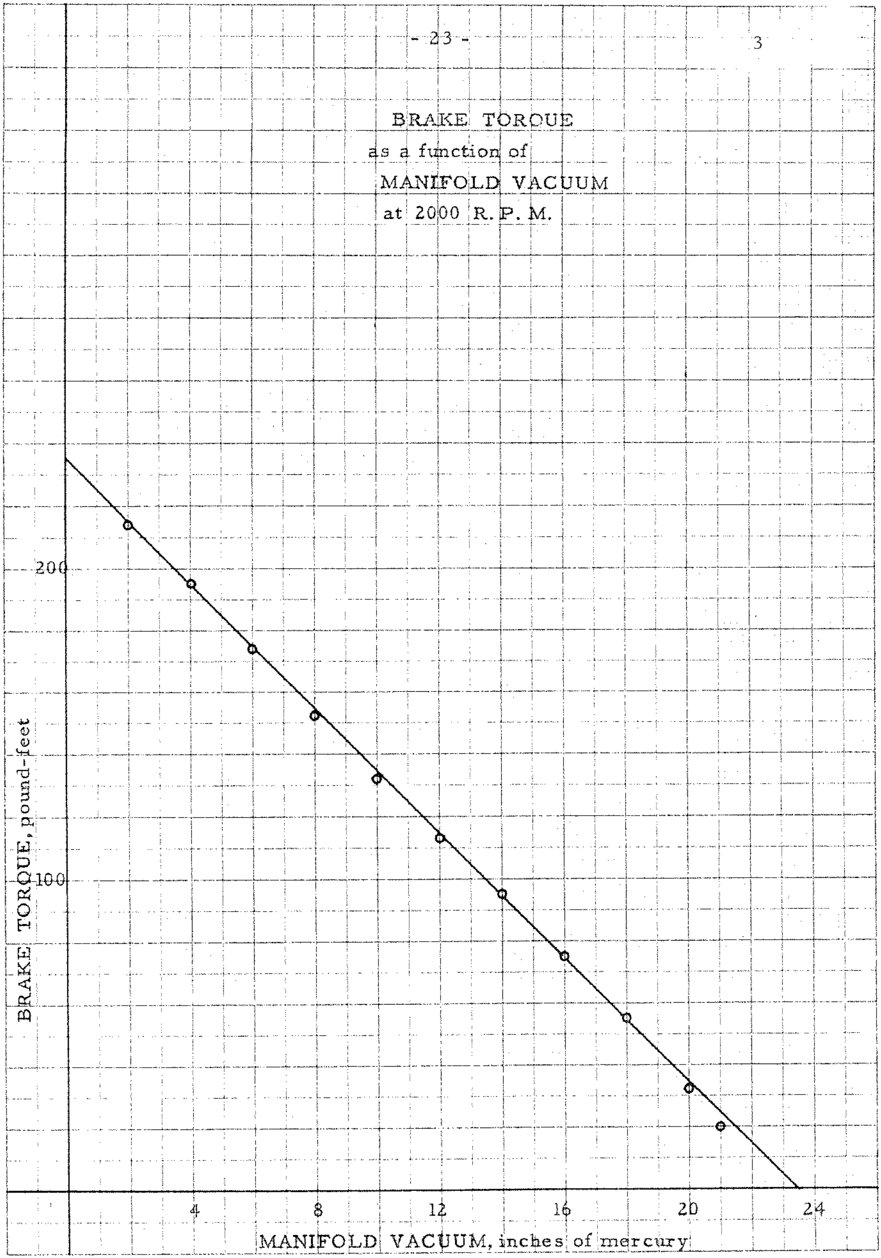
SUGGESTIONS FOR FURTHER RESEARCH

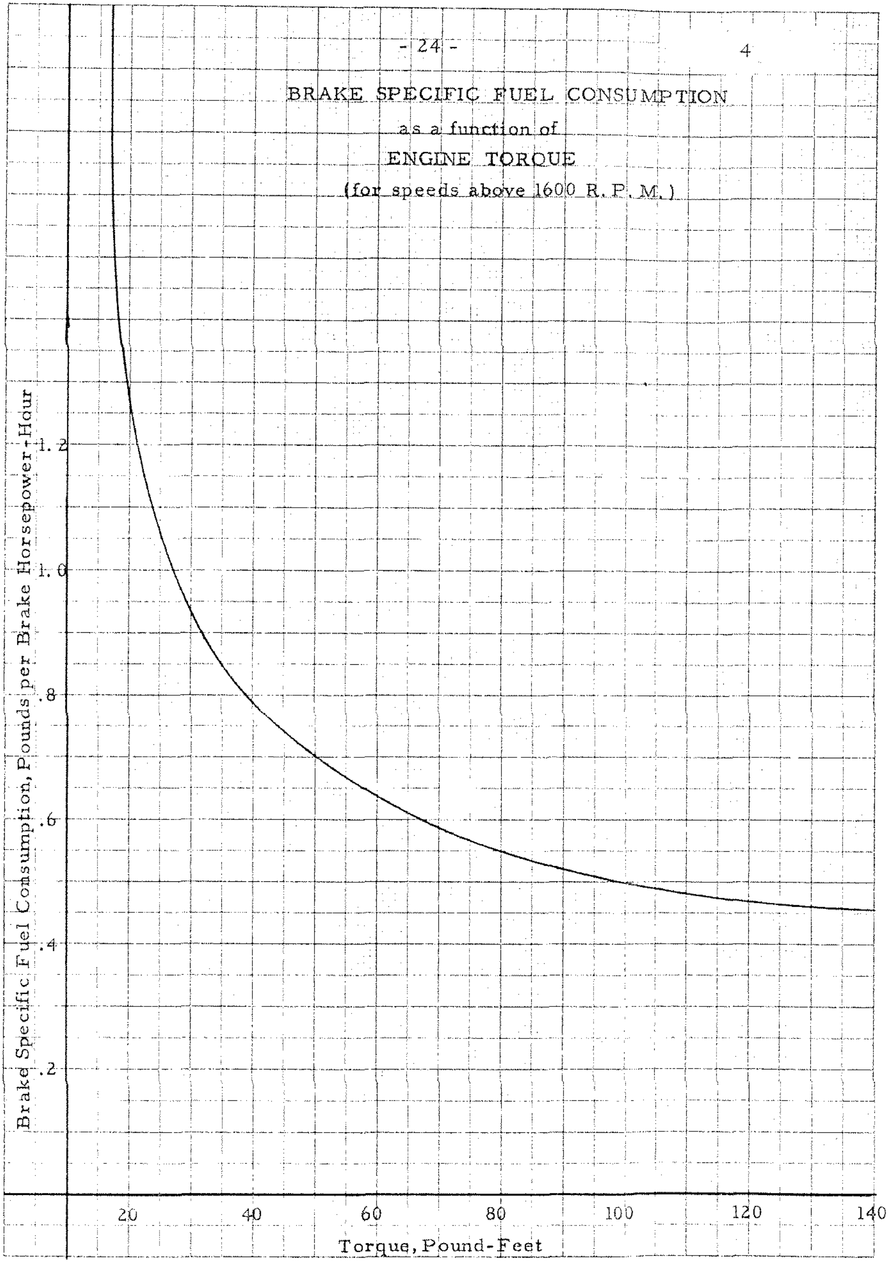
The analysis of surge technique indicates the possibility of using these methods of analysis for transient conditions, such as acceleration from a standstill, when certain simplifying assumptions are made. In fact it appears that optimum acceleration should be governed by the same considerations as in surge conditions, and that the locus of optimum efficiency points of the engine-transmission combination would define the desired technique quite well. This however implies a technique of nearly constant manifold vacuum, which is a strictly unnatural method, and perhaps unreasonable. The normal technique would be better approximated by holding a constant throttle angle during acceleration. In this case a numerical integration is necessary, since torque and fuel flow with constant throttle angle are both functions of speed, and can be plotted from engine performance curves. The analysis would be somewhat time consuming from the computational standpoint unless analytical expressions could be found in order to avoid the numerical integrations. It is quite likely that such a study would turn up some interesting and possibly unexpected results.

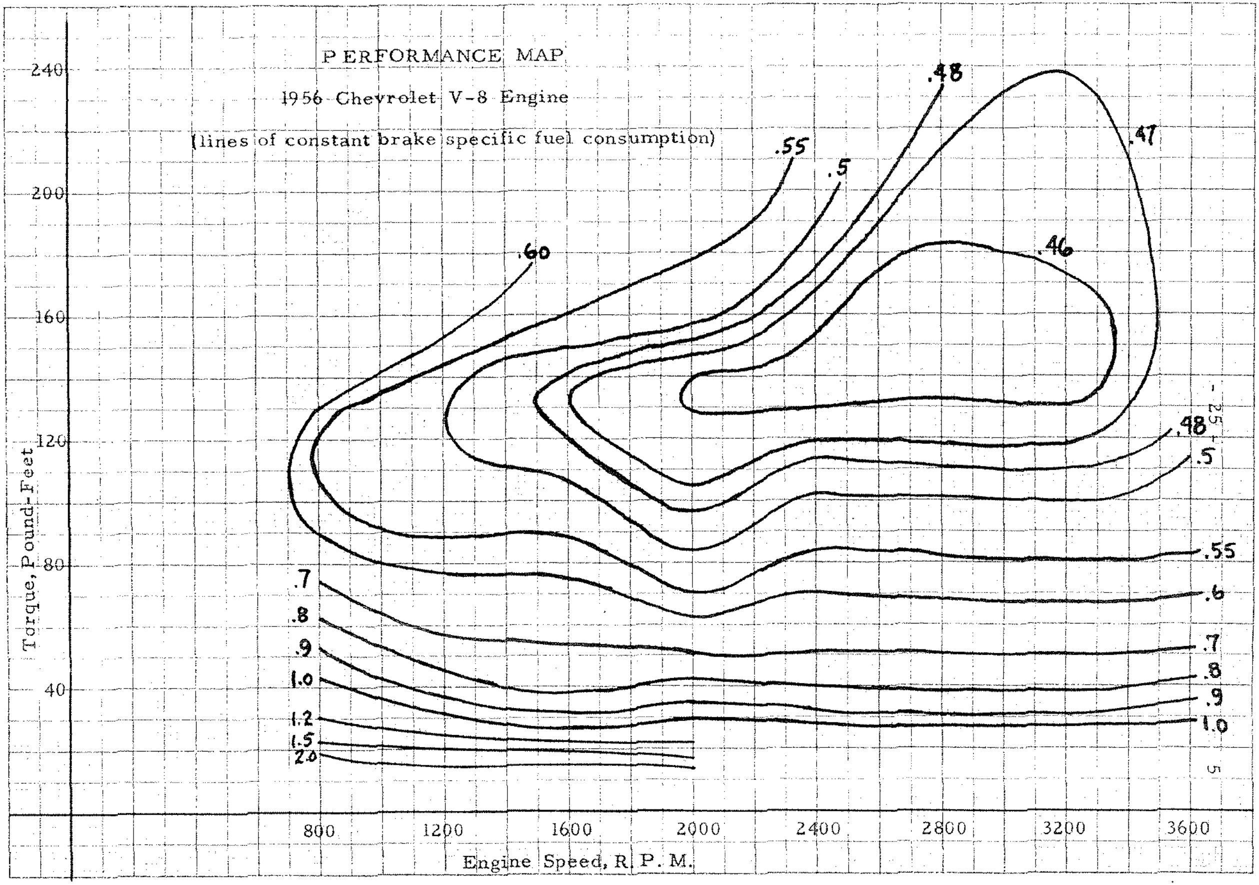
1


FUEL RATE as a function of SPEED and MANIFOLD VACUUM
2800 rpm
2400
2000
1600
1200
800
MANIFOLD VACUUM, INCHES OF MERCURY
10
12
14
16
18
20
22
FUEL RATE, POUNDS OF FUEL PER HOUR
36
32
28
24
20
16
12
8
4

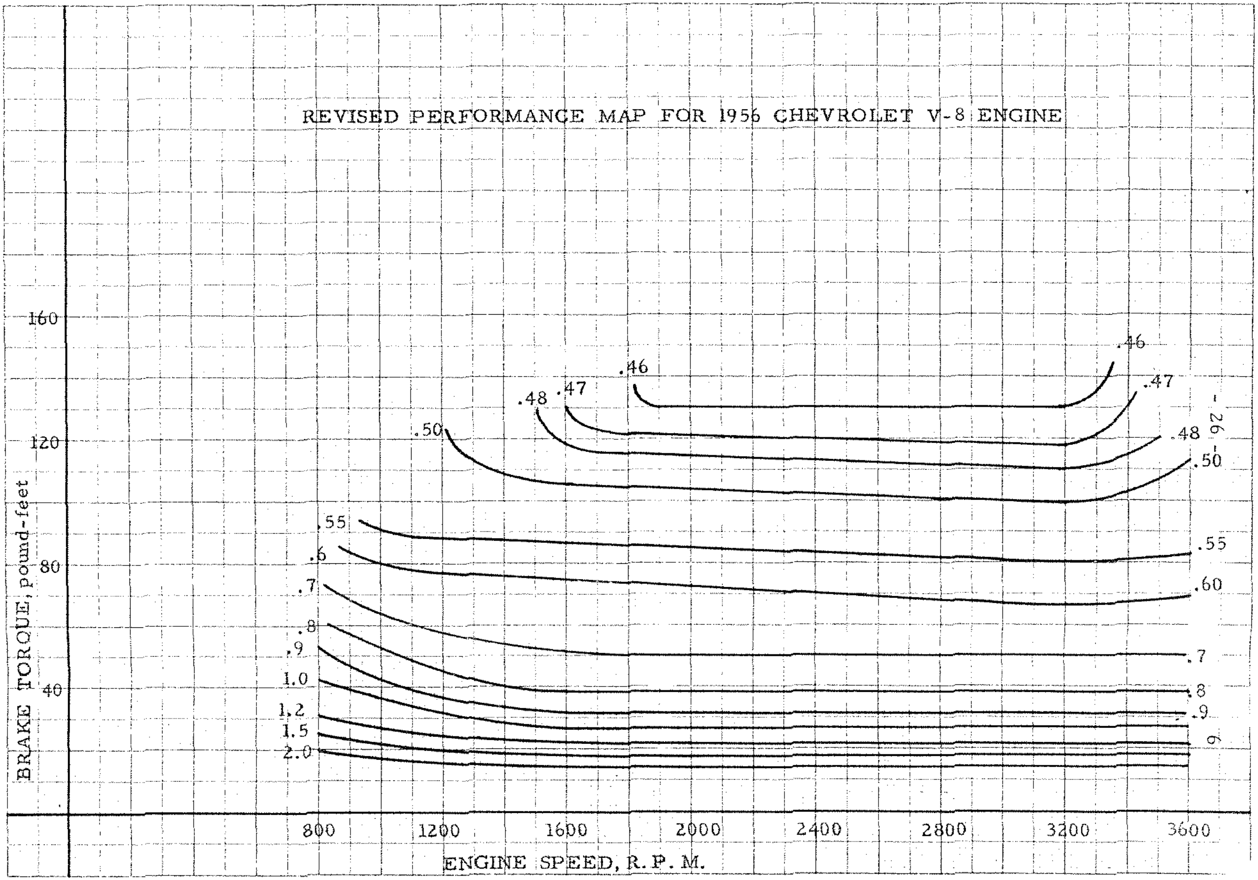

BRAKE TORQUE
as a function of
ENGINE SPEED
at various manifold depressions
wide open throttle
$\Delta p = 2"$
4"
6"
8"
10"
12"
14"
16"
18"
20"
21"
22"
300
200
100
ENGINE TORQUE, POUND-FEET
800
1200
1600
2000
2400
2800
ENGINE SPEED, R.P.M.

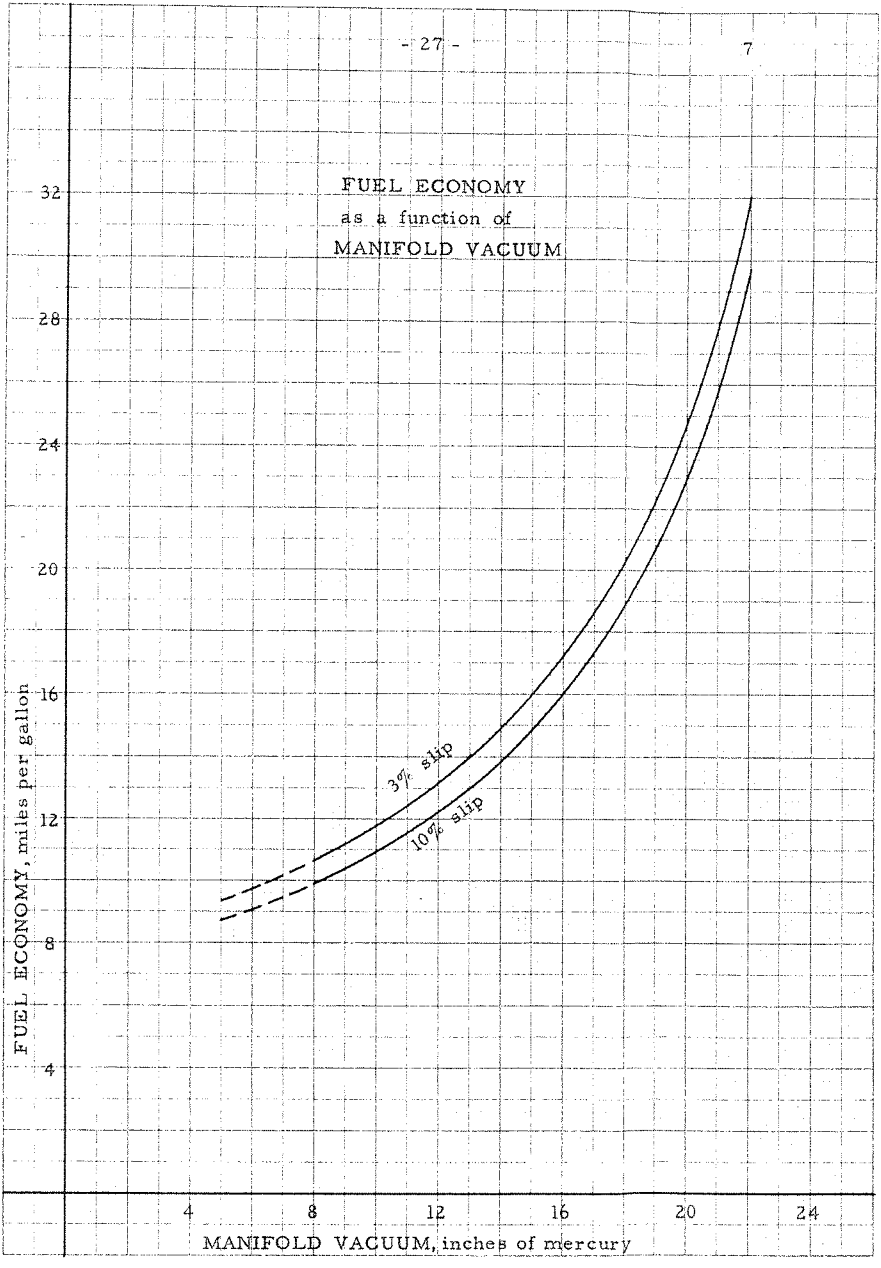

BRAKE TORQUE
as a function of
MANIFOLD VACUUM
at 2000 R. P. M.
200
100
BRAKE TORQUE, pound-feet
4
8
12
16
20
24
MANIFOLD VACUUM, inches of mercury

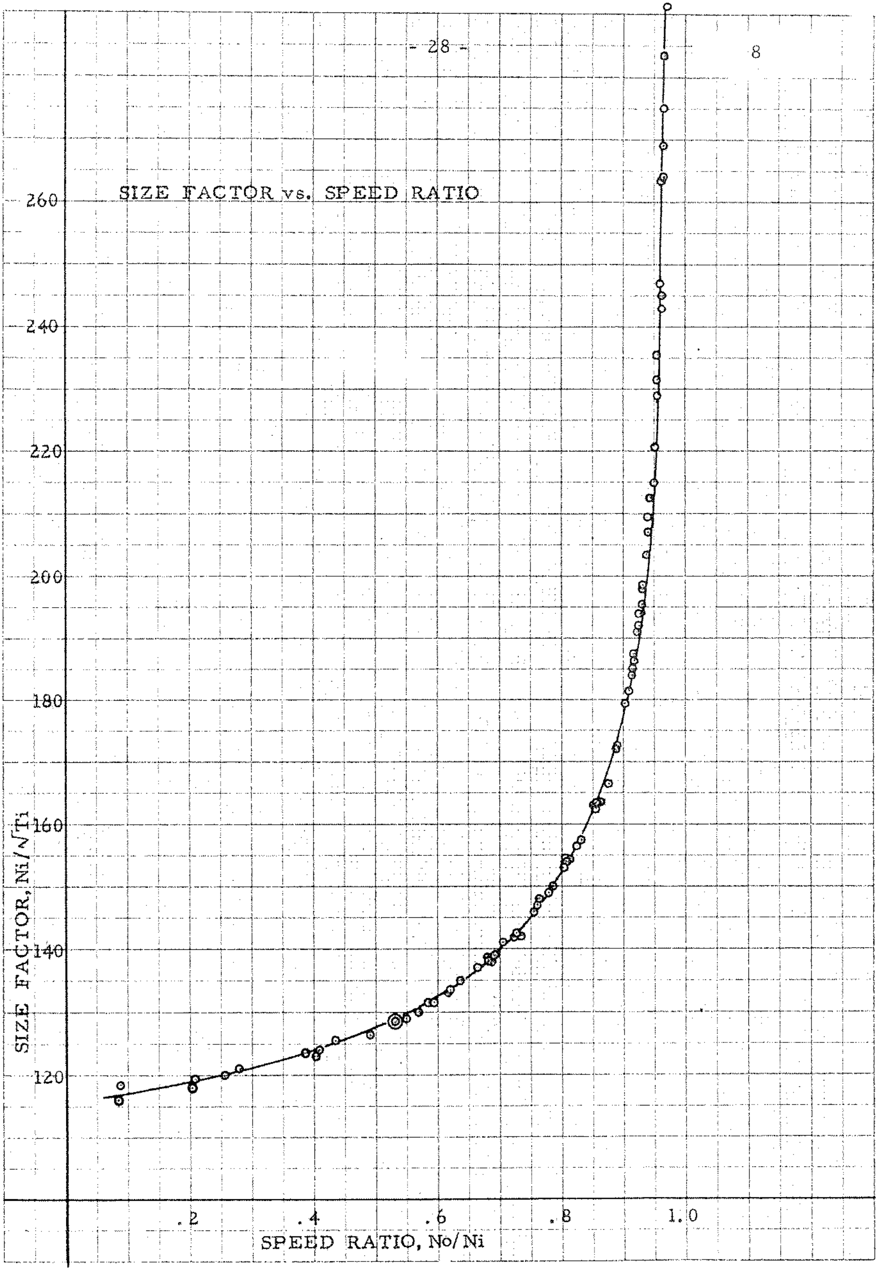

BRAKE SPECIFIC FUEL CONSUMPTION
as a function of
ENGINE TORQUE
(for speeds above 1600 R.P.M.)
Brake Specific Fuel Consumption, Pounds per Brake Horsepower-Hour
1.2
1.0
.8
.6
.4
.2
20
40
60
80
100
120
140
Torque, Pound-Feet



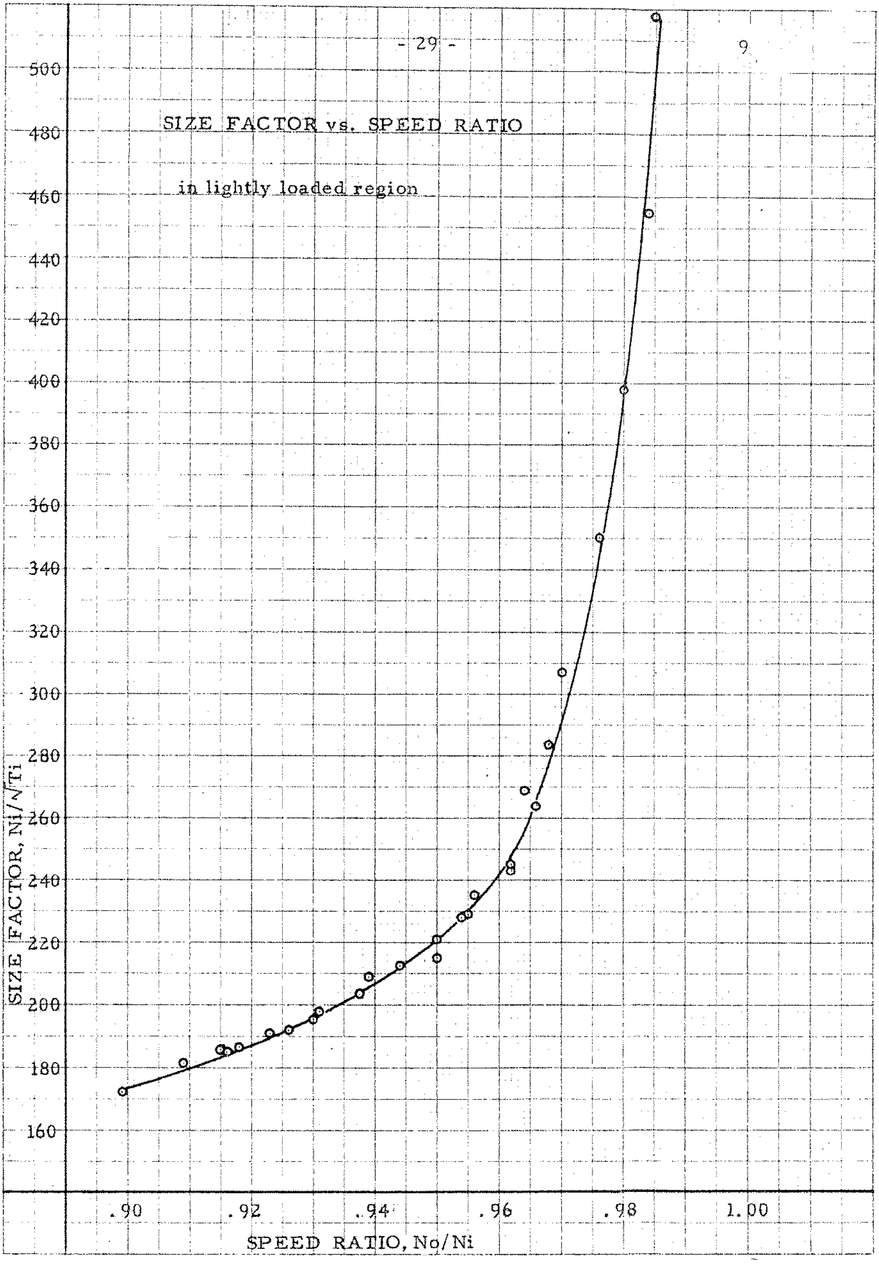
PERFORMANCE MAP
1956 Chevrolet V-8 Engine
(lines of constant brake specific fuel consumption)
Torque, Pound-Feet
240
200
160
120
80
40
Engine Speed, R. P. M.
800
1200
1600
2000
2400
2800
3200
3600
.60
.55
.5
.48
.47
.46
.6
.7
.8
.9
1.0
1.2
1.5
2.0

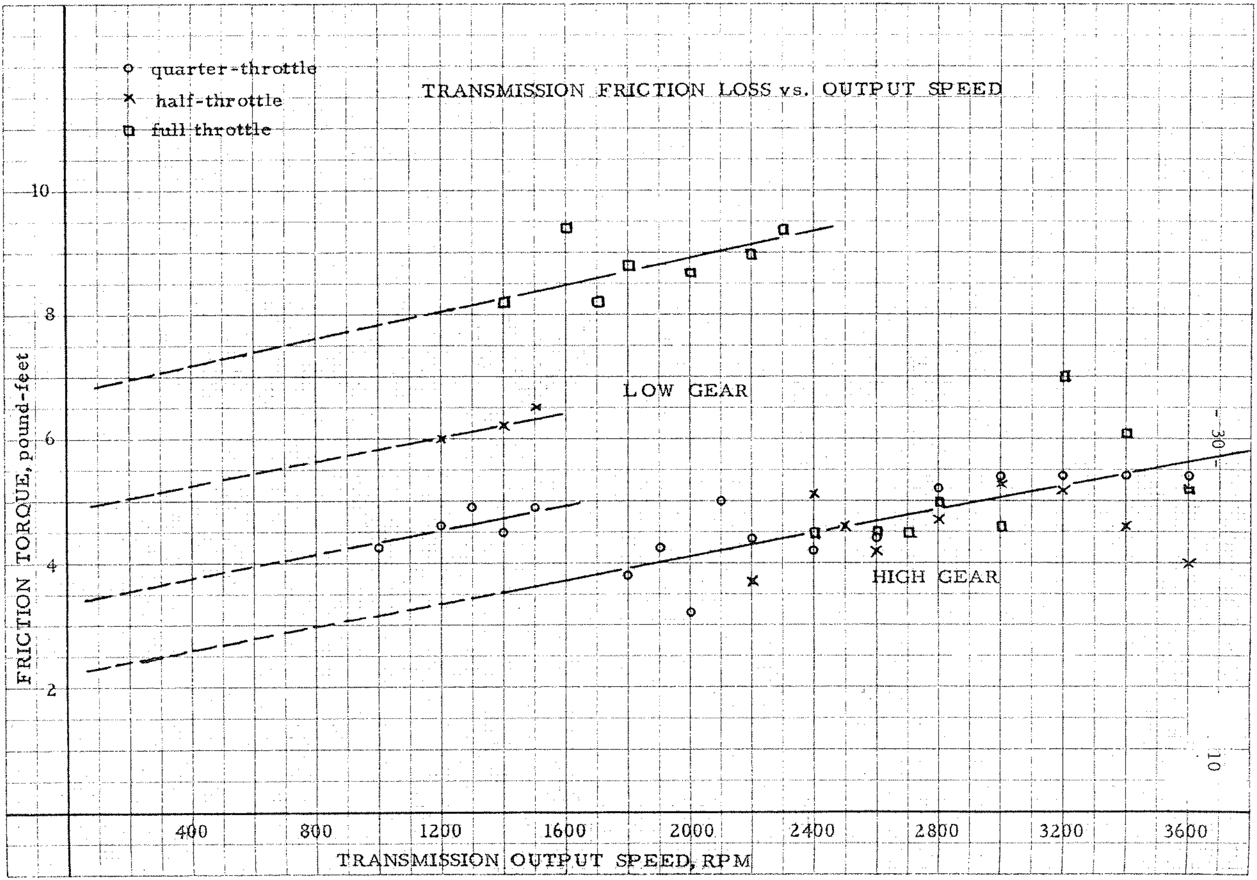

REVISED PERFORMANCE MAP FOR 1956 CHEVROLET V-8 ENGINE
BRAKE TORQUE, pound-feet
160
120
80
40
ENGINE SPEED, R. P. M.
800
1200
1600
2000
2400
2800
3200
3600
.46
.47
.48
.50
.55
.6
.7
.8
.9
1.0
1.2
1.5
2.0
.46
.47
.48
.50
.55
.60
.7
.8
.9
6


FUEL ECONOMY
as a function of
MANIFOLD VACUUM
3% slip
10% slip
FUEL ECONOMY, miles per gallon
4
8
12
16
20
24
28
32
MANIFOLD VACUUM, inches of mercury
4
8
12
16
20
24



SIZE FACTOR vs. SPEED RATIO
260
240
220
200
180
160
140
120
SIZE FACTOR, Ni/√Ti
.2
.4
.6
.8
1.0
SPEED RATIO, No/Ni

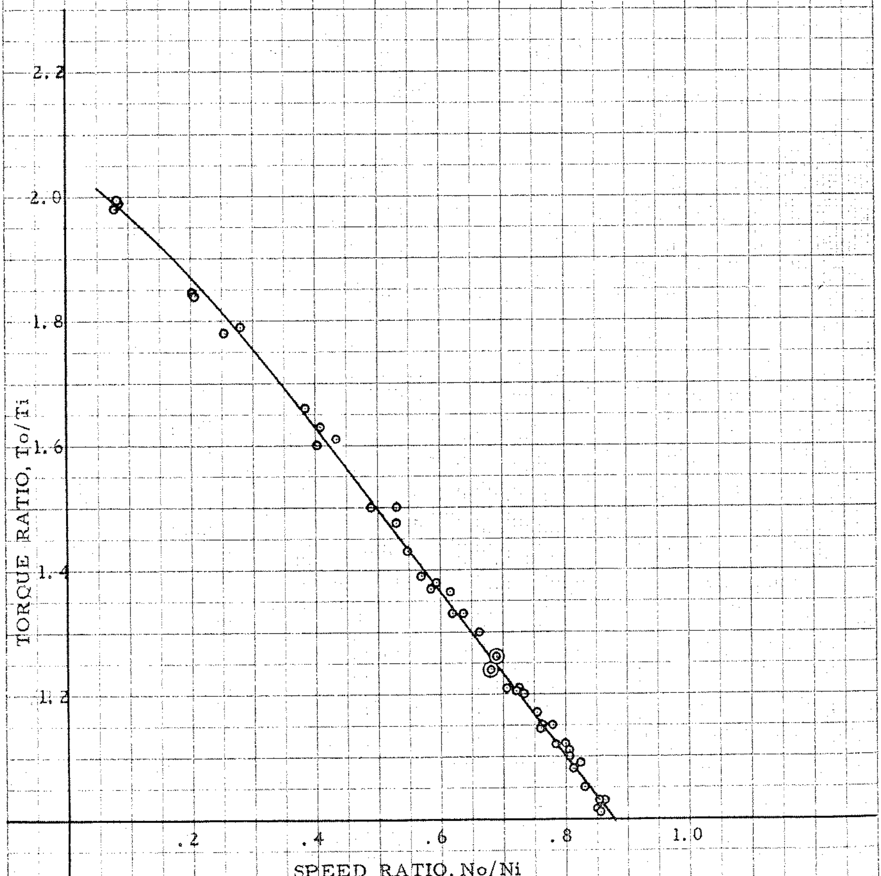

9
SIZE FACTOR vs. SPEED RATIO
in lightly loaded region
SIZE FACTOR, Ni/√Ti
500
480
460
440
420
400
380
360
340
320
300
280
260
240
220
200
180
160
.90
.92
.94
.96
.98
1.00
SPEED RATIO, No/Ni


quarter-throttle
half-throttle
full throttle
TRANSMISSION FRICTION LOSS vs. OUTPUT SPEED
LOW GEAR
HIGH GEAR
FRICTION TORQUE, pound-feet
10
8
6
4
2
400
800
1200
1600
2000
2400
2800
3200
3600
TRANSMISSION OUTPUT SPEED, RPM

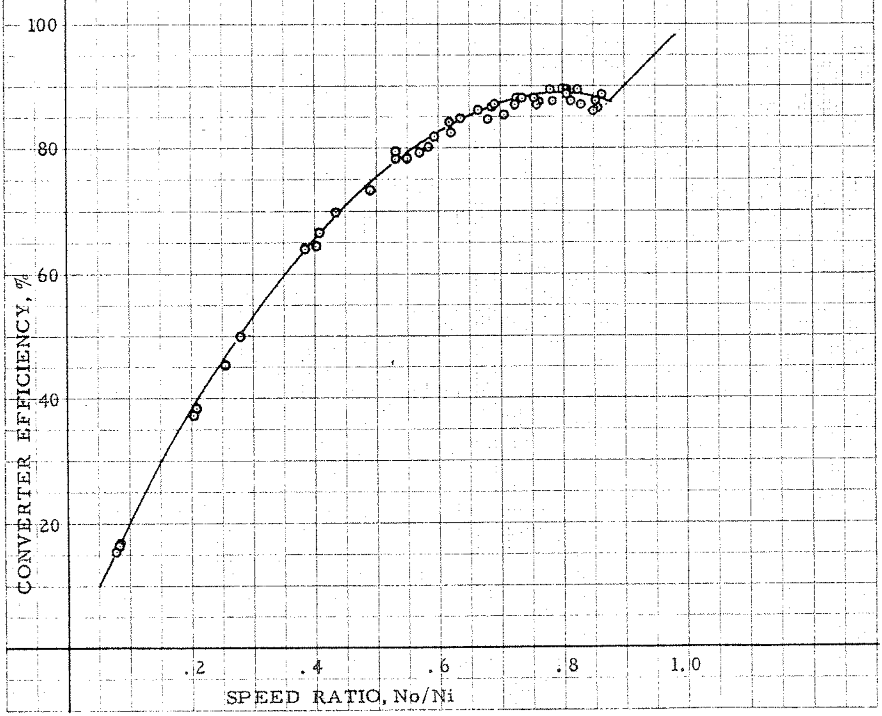
10

TORQUE RATIO vs. SPEED RATIO

POWERGLIDE TORQUE CONVERTER

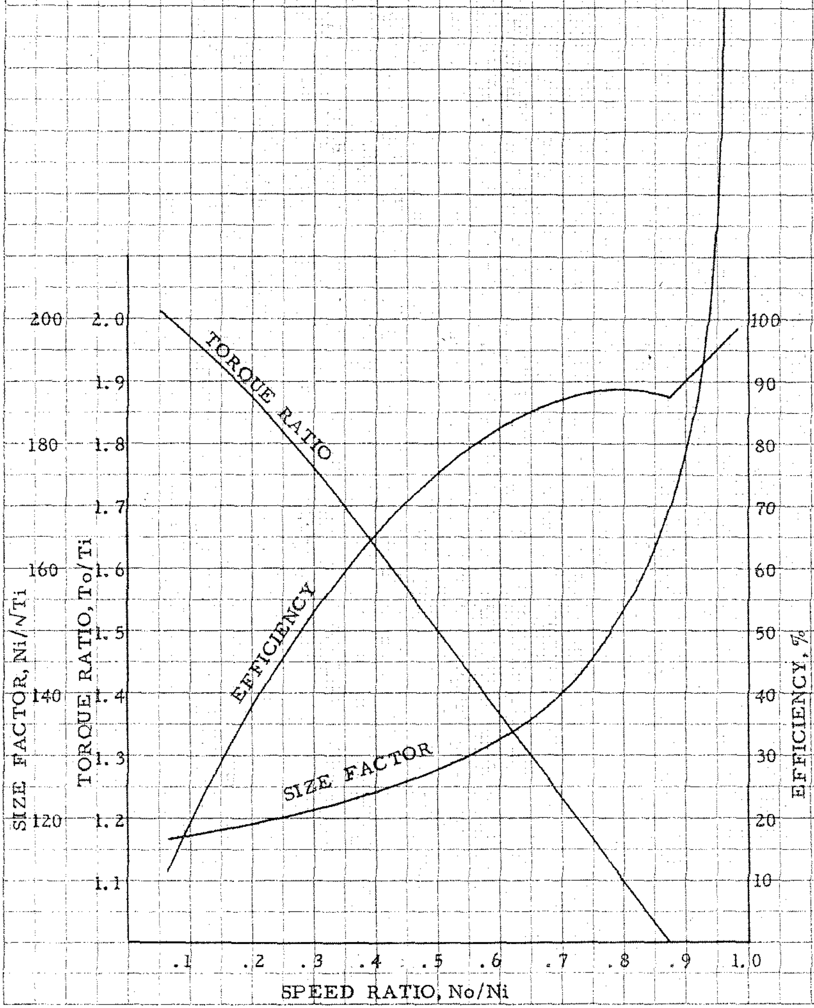


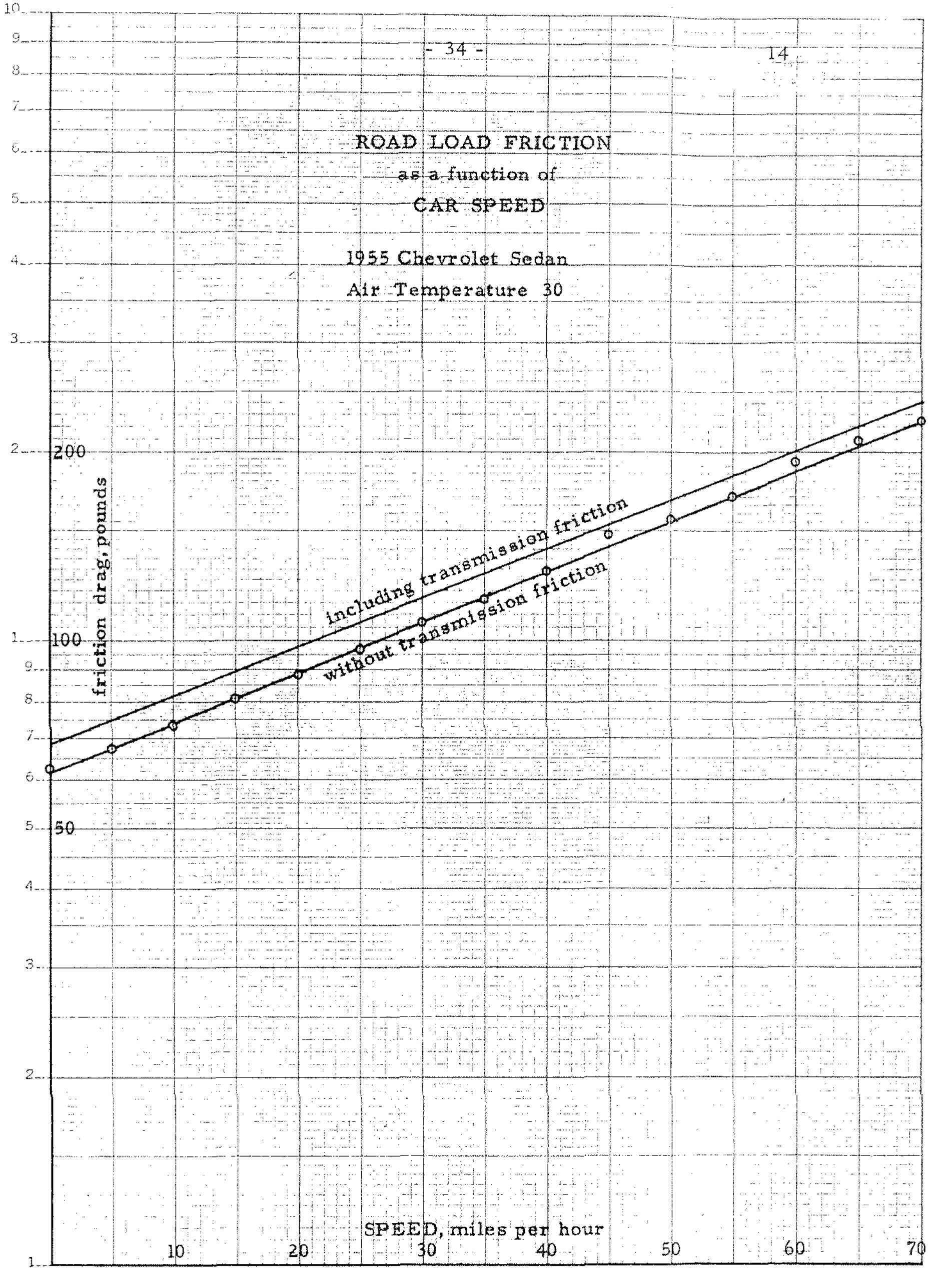
CONVERTER EFFICIENCY vs. SPEED RATIO

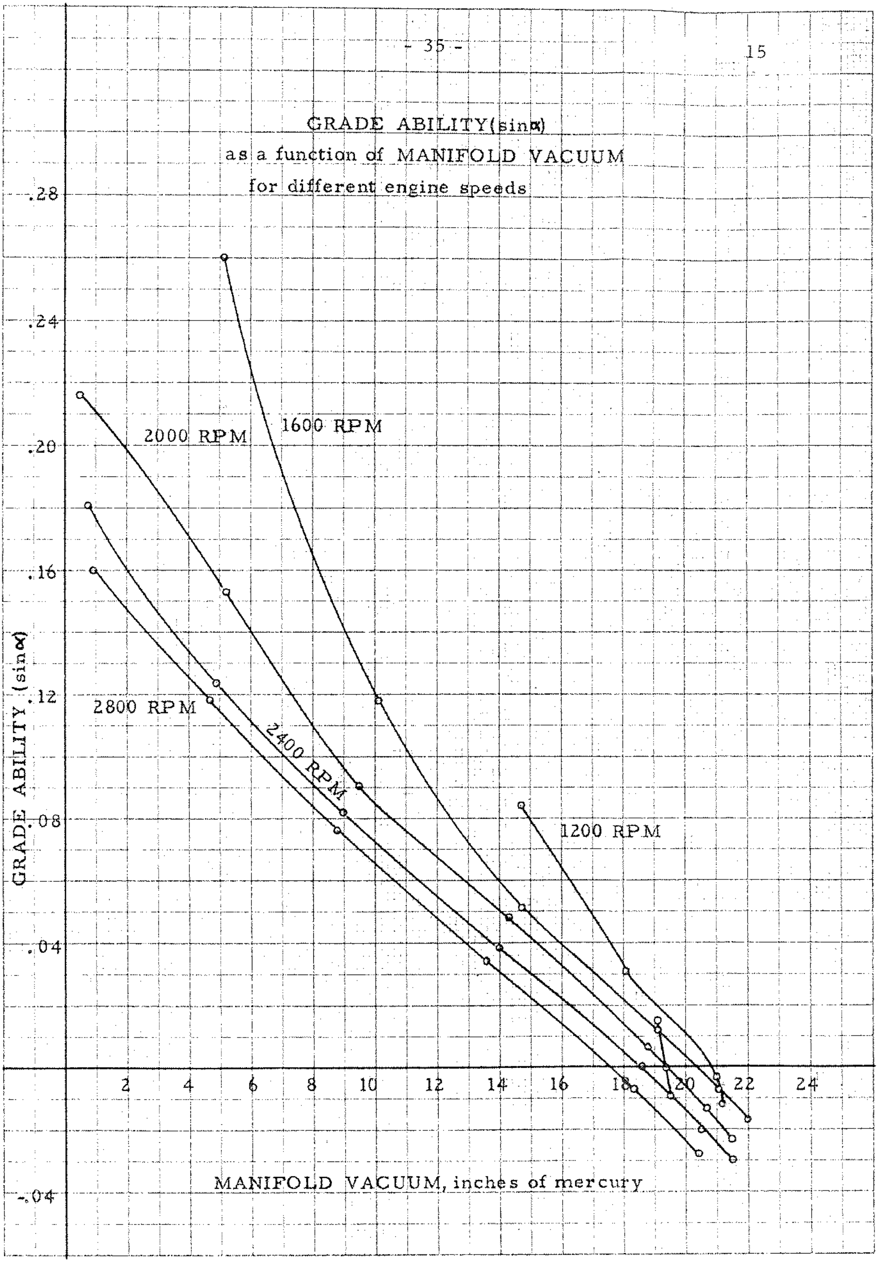


PERFORMANCE CHARACTERISTICS

POWERGLIDE 3-ELEMENT TORQUE CONVERTER

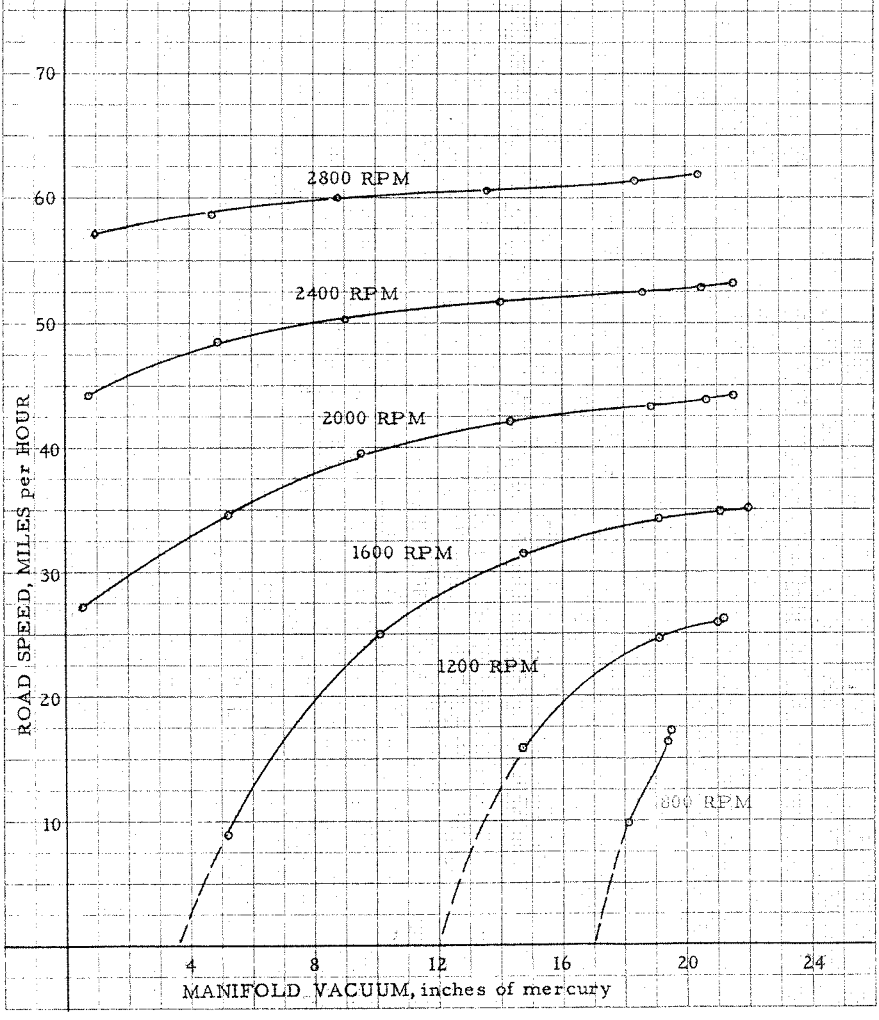



ROAD LOAD FRICTION
as a function of
CAR SPEED
1955 Chevrolet Sedan
Air Temperature 30
friction drag, pounds
200
100
50
including transmission friction
without transmission friction
SPEED, miles per hour
10
20
30
40
50
60
70


GRADE ABILITY(sinα)
as a function of MANIFOLD VACUUM
for different engine speeds
1600 RPM
2000 RPM
2800 RPM
2400 RPM
1200 RPM
GRADE ABILITY (sinα)
MANIFOLD VACUUM, inches of mercury
.28
.24
.20
.16
.12
.08
.04
-.04
2
4
6
8
10
12
14
16
18
20
22
24

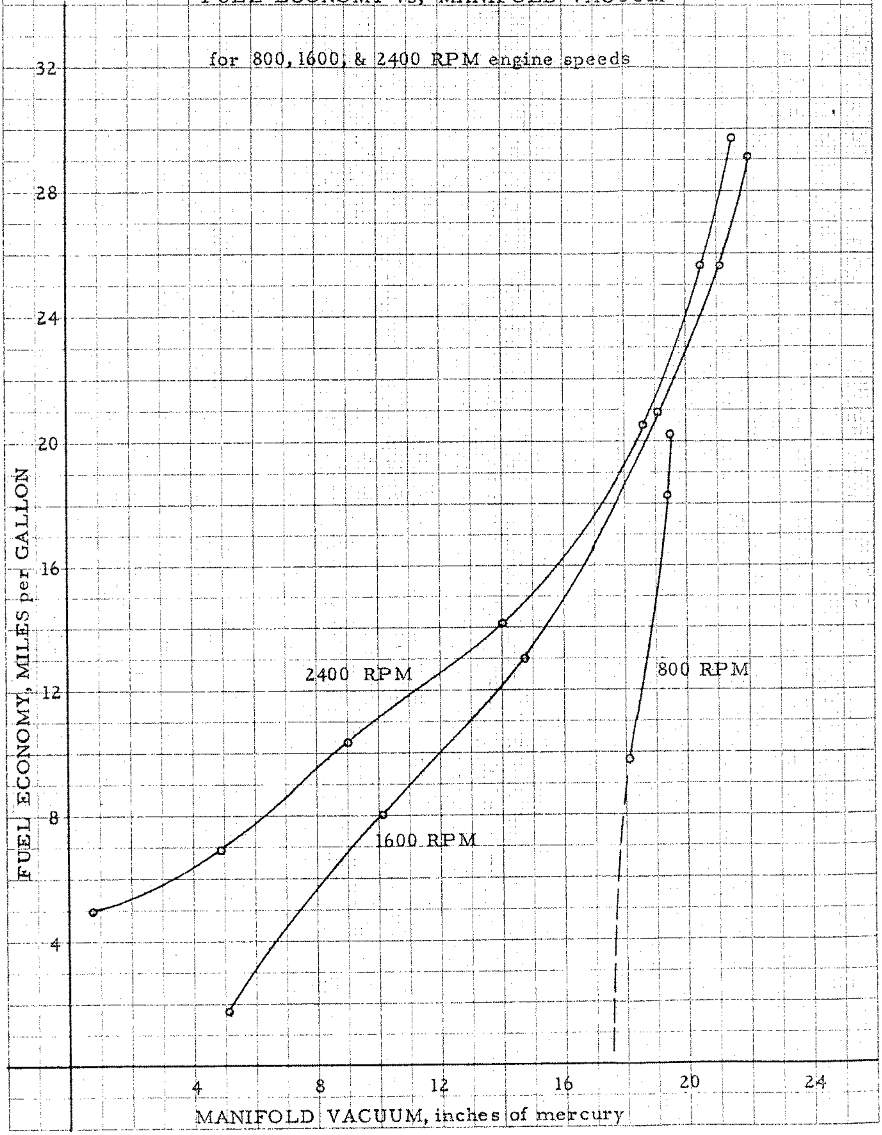
ROAD SPEED as a function of MANIFOLD VACUUM

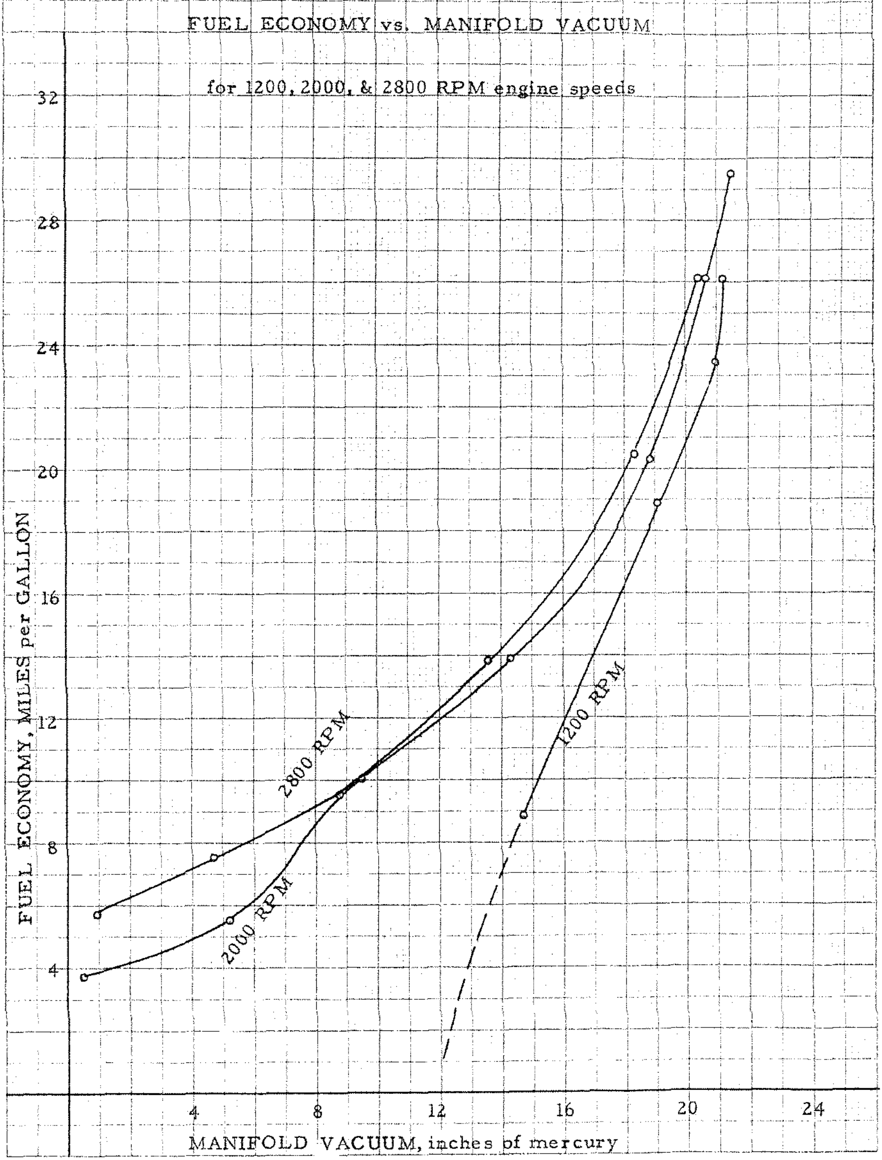
For different engine speeds



FUEL ECONOMY vs, MANIFOLD VACUUM

for 800, 1600, & 2400 RPM engine speeds

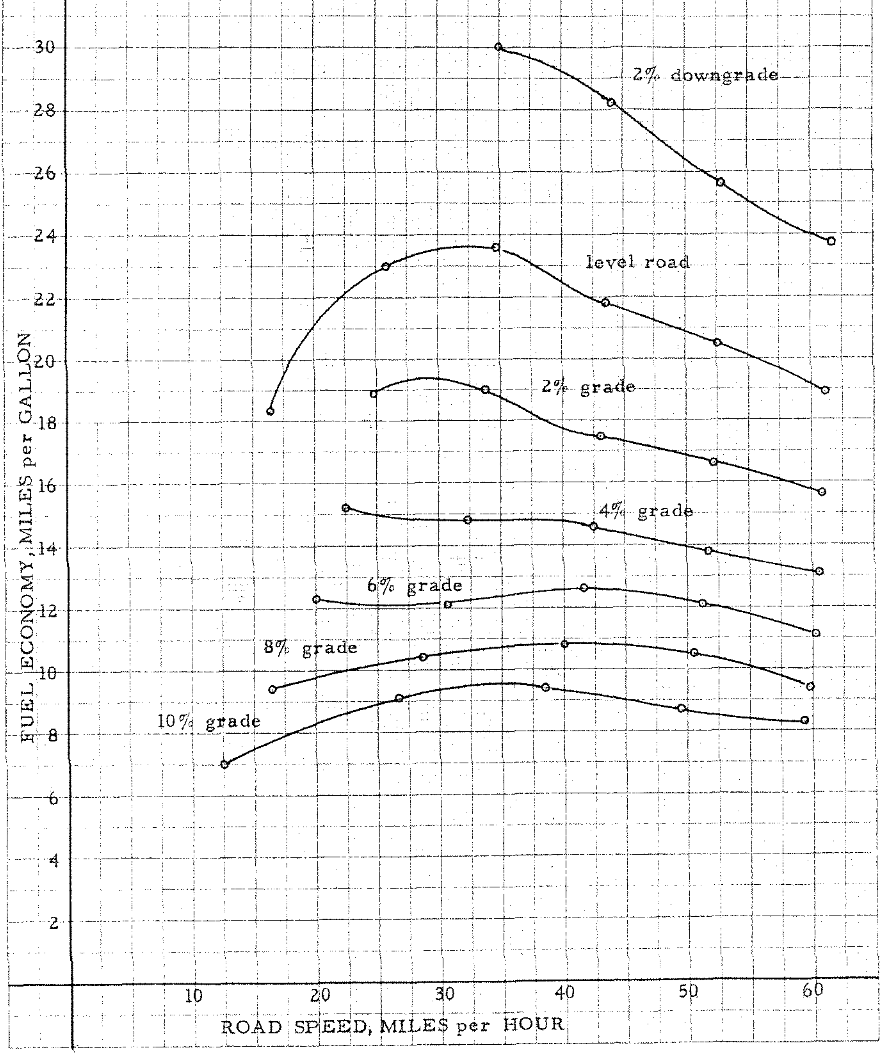




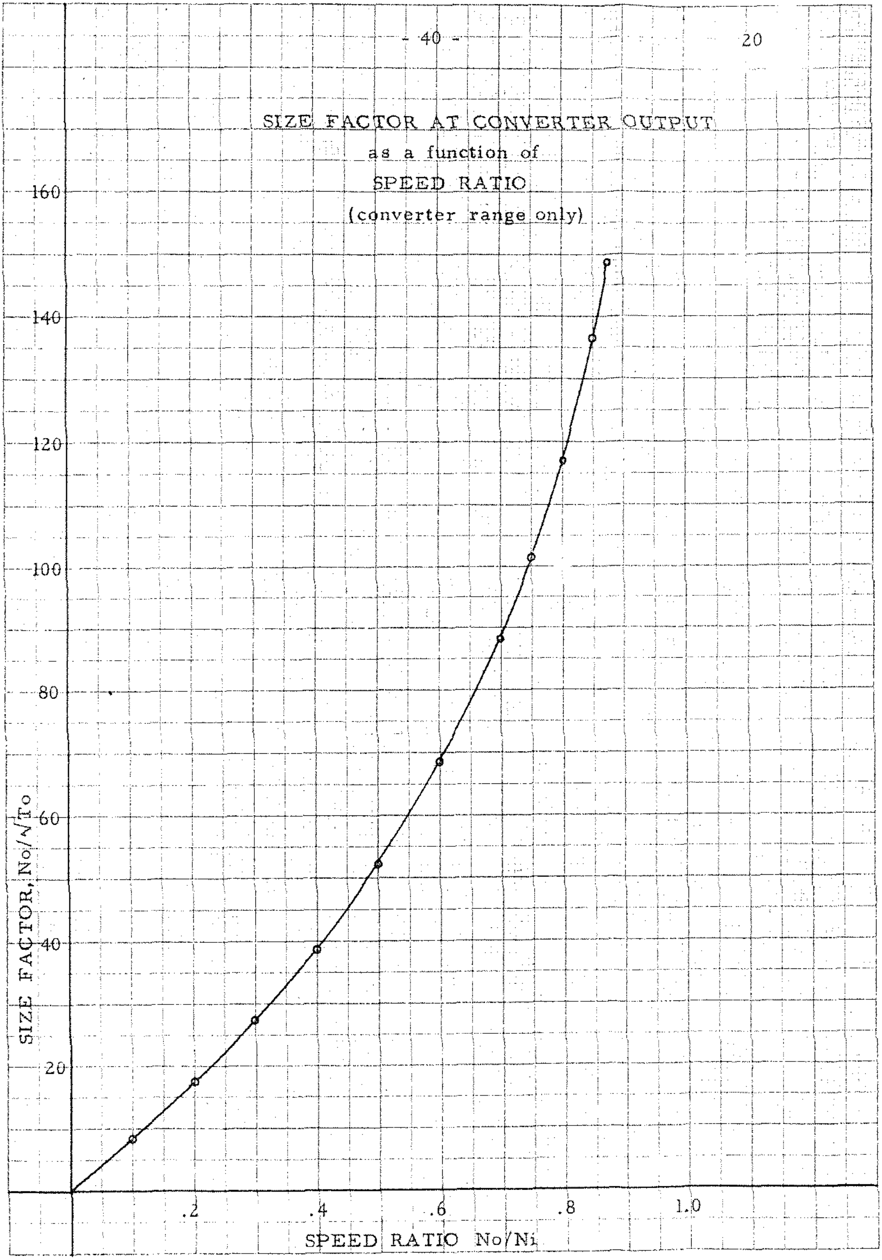
FUEL ECONOMY vs. MANIFOLD VACUUM
for 1200, 2000, & 2800 RPM engine speeds
FUEL ECONOMY, MILES per GALLON
32
28
24
20
16
12
8
4
2800 RPM
2000 RPM
1200 RPM
4
8
12
16
20
24
MANIFOLD VACUUM, inches of mercury

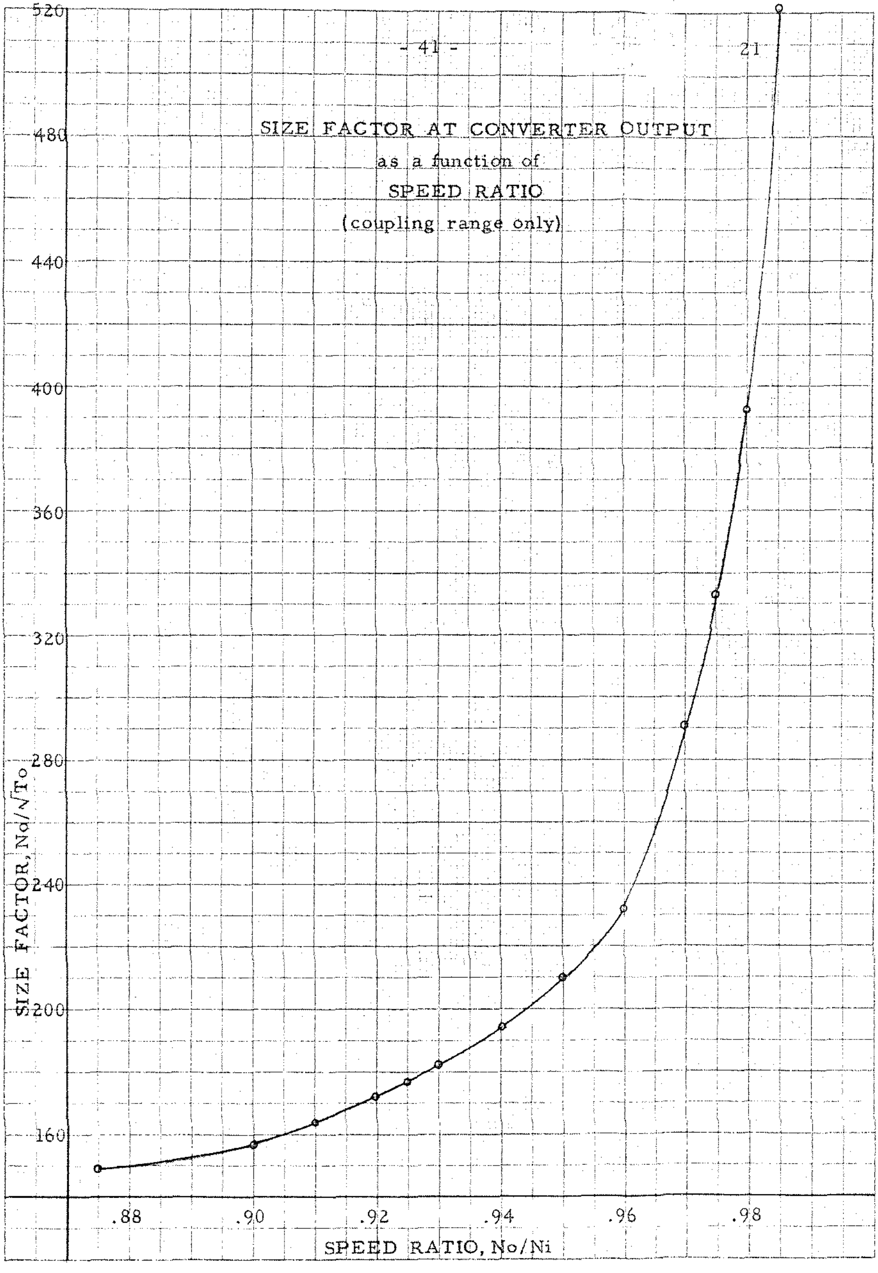
FUEL ECONOMY vs. ROAD SPEED

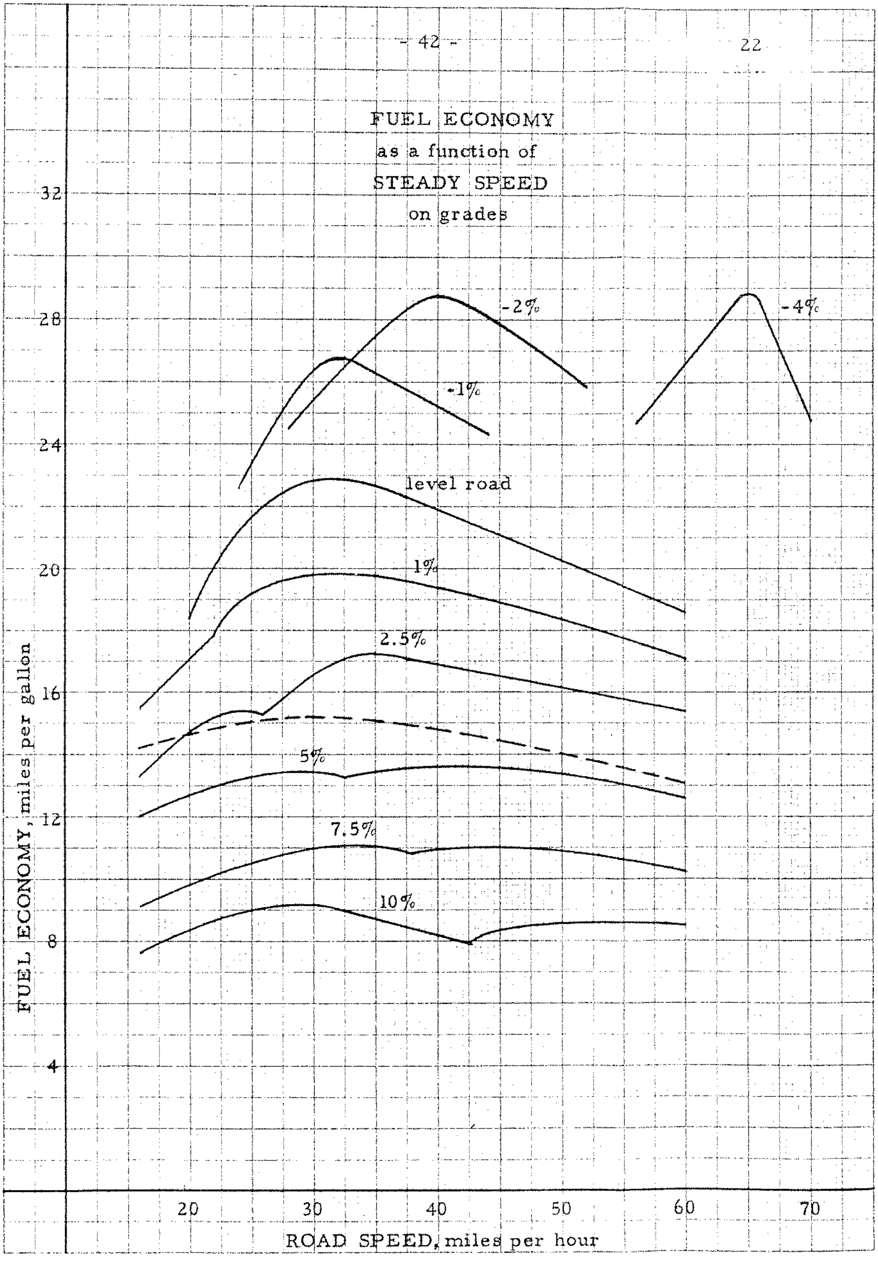
for given per cent grade



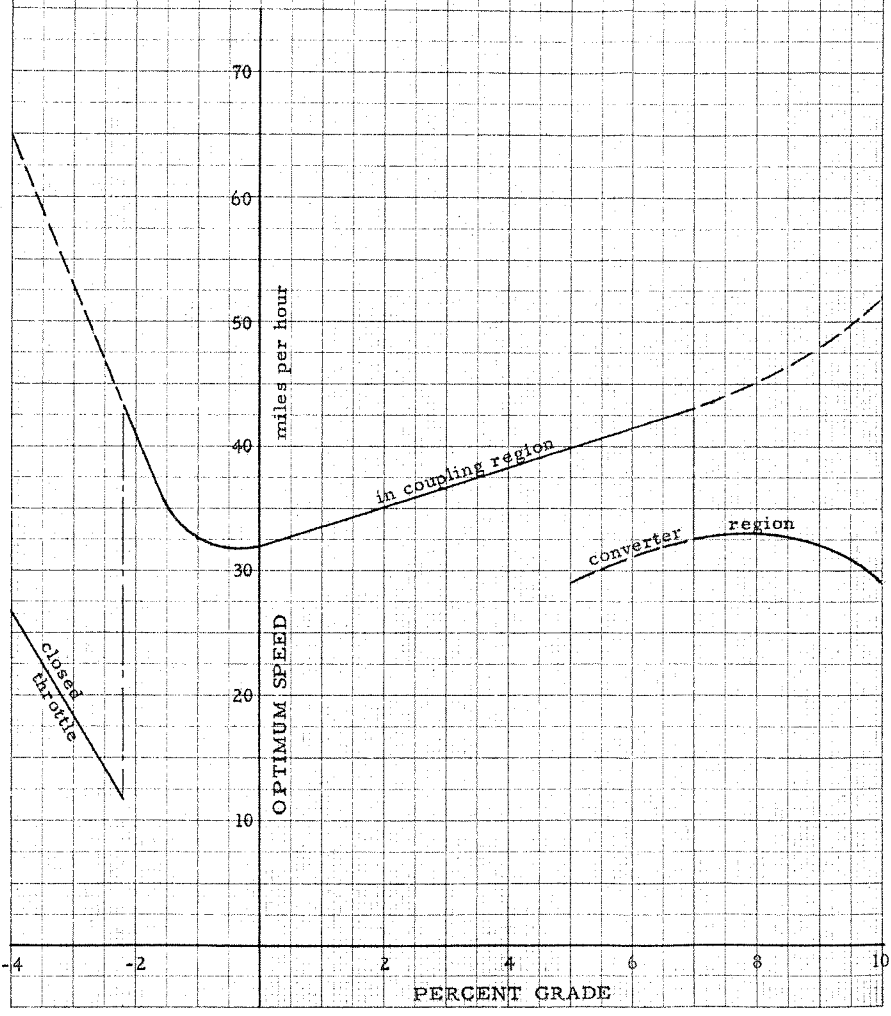
20


SIZE FACTOR AT CONVERTER OUTPUT
as a function of
SPEED RATIO
(converter range only)
160
140
120
100
80
60
40
20
SIZE FACTOR, No/√To
.2
.4
.6
.8
1.0
SPEED RATIO No/Ni

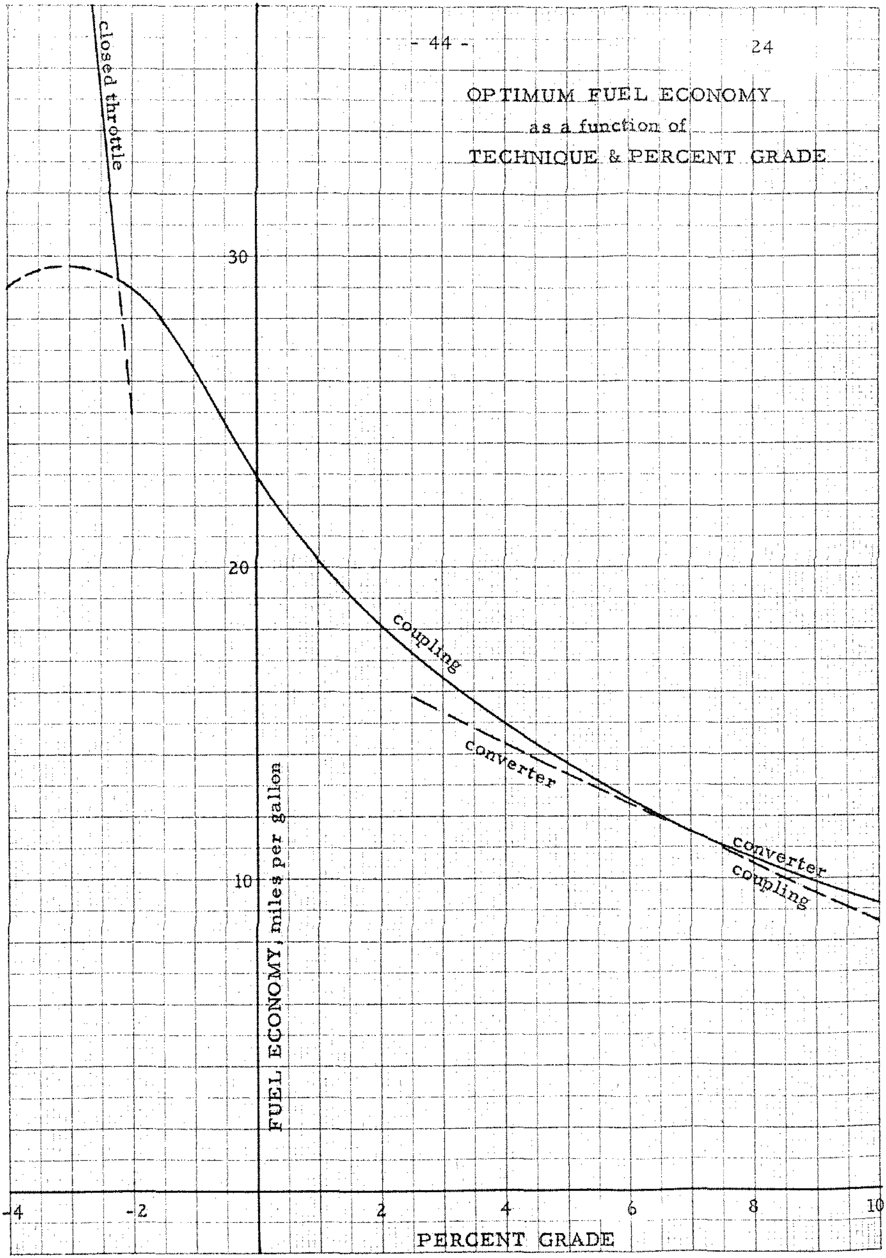

SIZE FACTOR AT CONVERTER OUTPUT
as a function of
SPEED RATIO
(coupling range only)
520
480
440
400
360
320
280
240
200
160
SIZE FACTOR, Nd/√To
.88
.90
.92
.94
.96
.98
SPEED RATIO, No/Ni

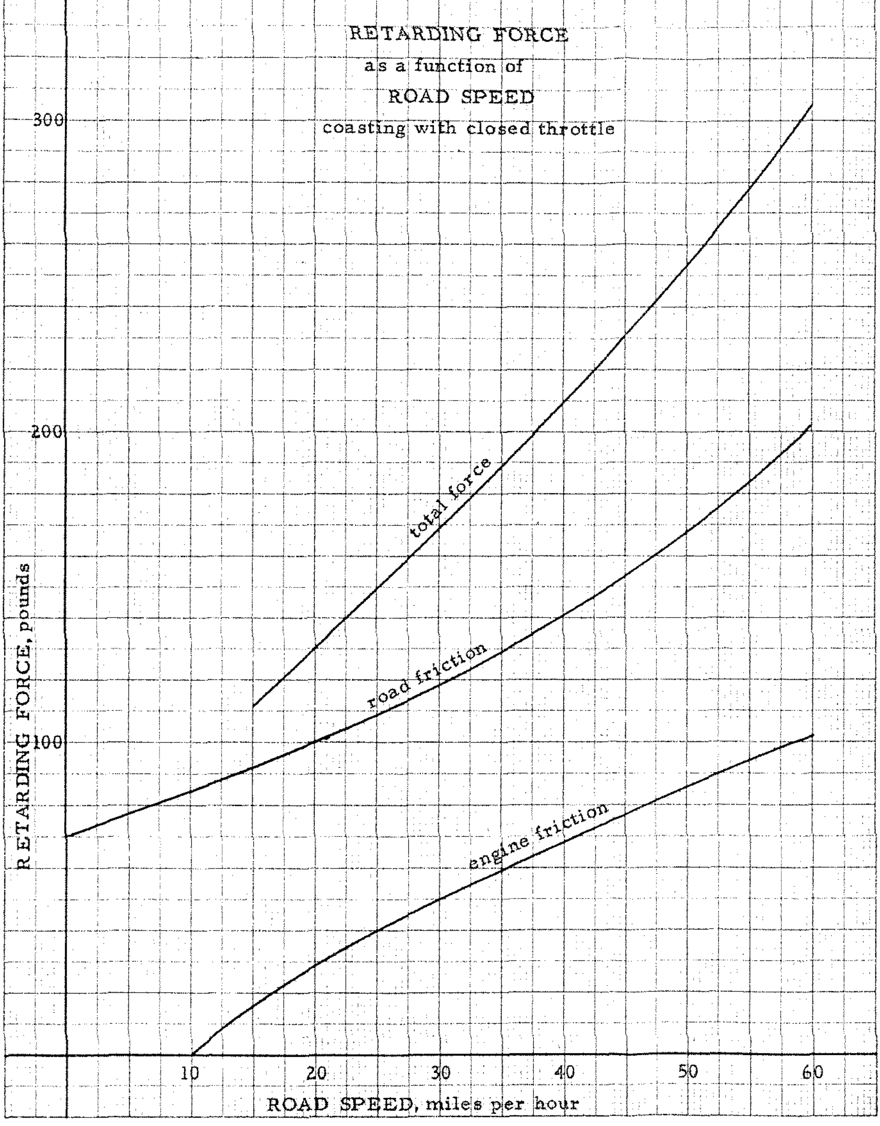

FUEL ECONOMY
as a function of
STEADY SPEED
on grades
-2%
-4%
-1%
level road
1%
2.5%
5%
7.5%
10%
32
28
24
20
16
12
8
4
FUEL ECONOMY, miles per gallon
20
30
40
50
60
70
ROAD SPEED, miles per hour

SPEED FOR OPTIMUM ECONOMY
as a function of
TECHNIQUE & PERCENT GRADE

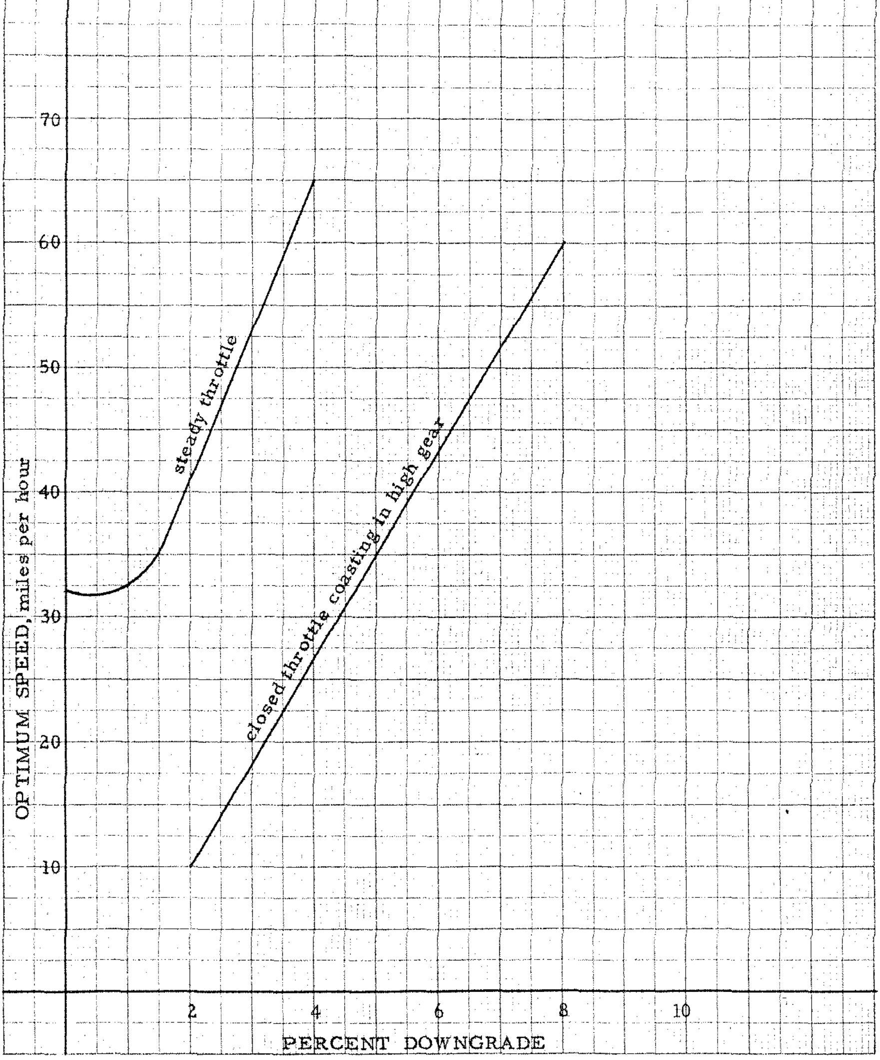


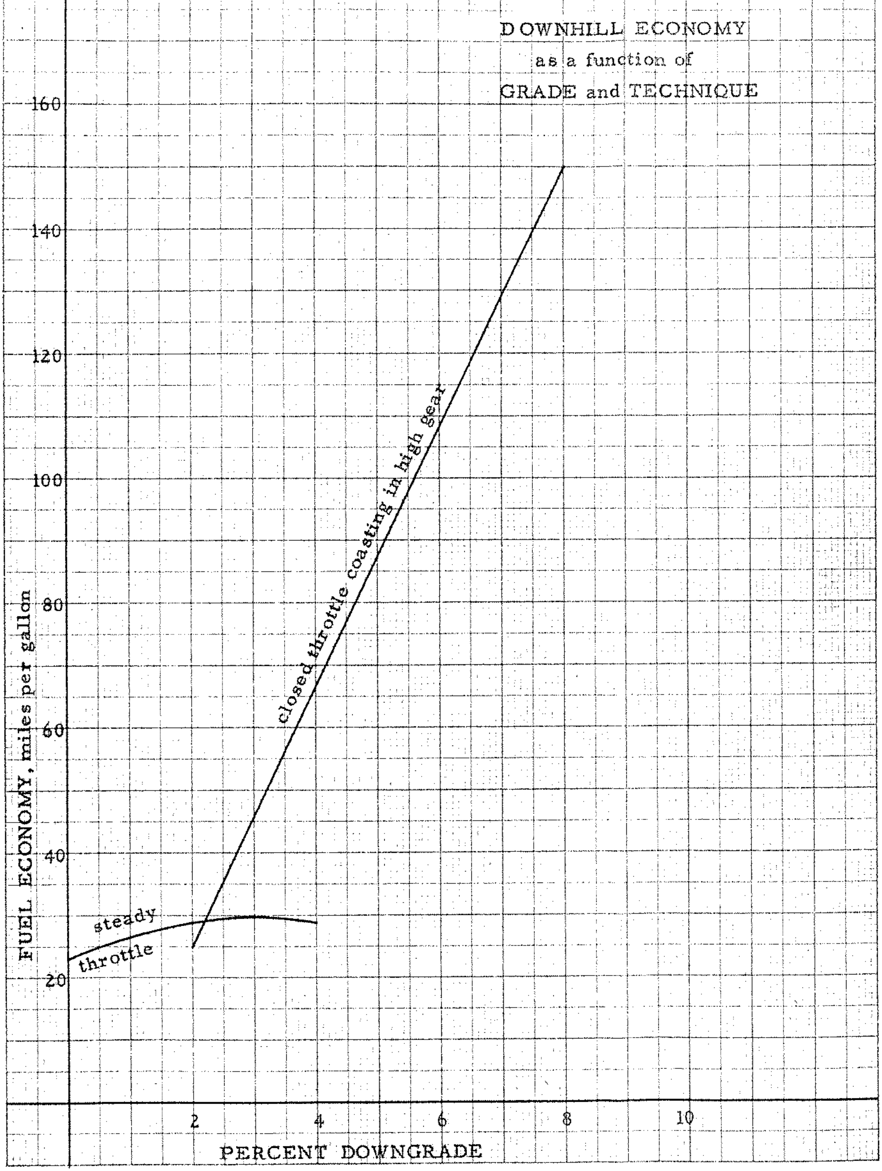
24

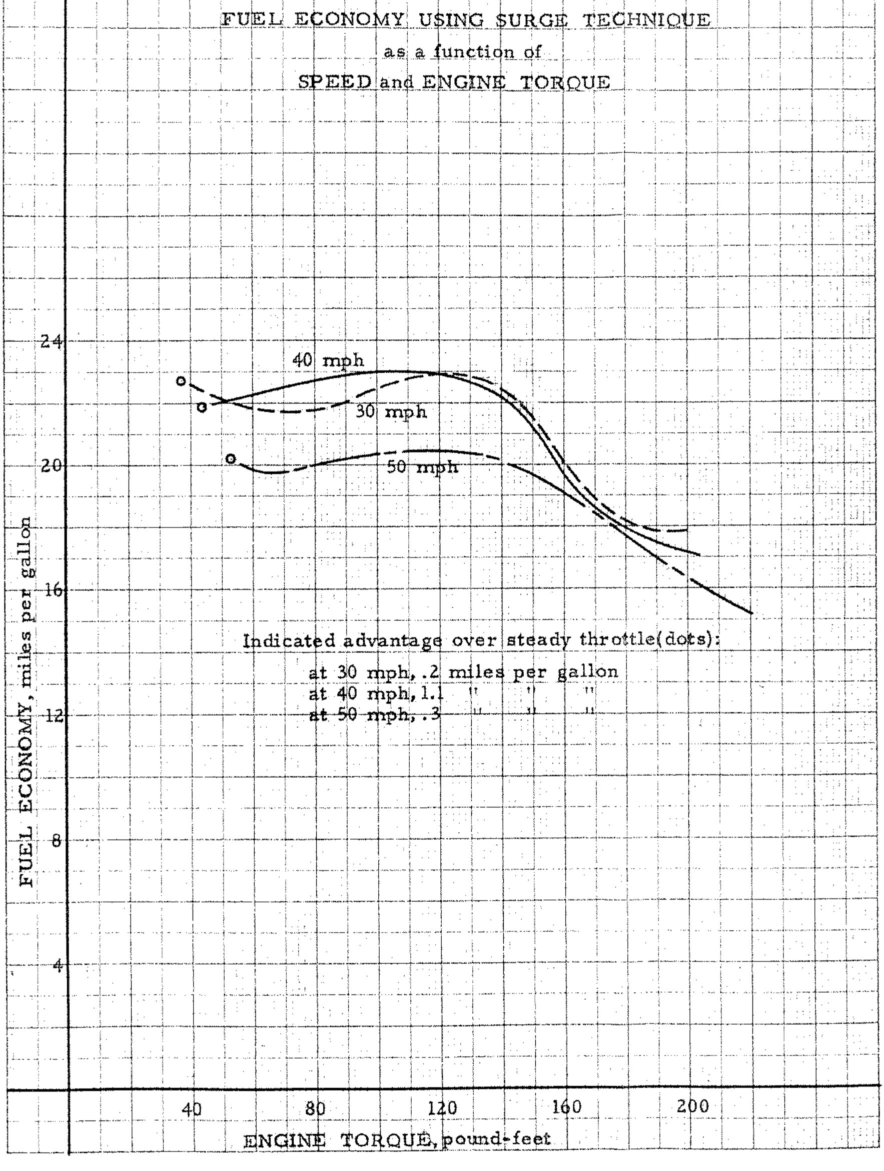

OPTIMUM FUEL ECONOMY
as a function of
TECHNIQUE & PERCENT GRADE
closed throttle
coupling
converter
converter
coupling
30
20
10
FUEL ECONOMY, miles per gallon
-4
-2
2
4
6
8
10
PERCENT GRADE

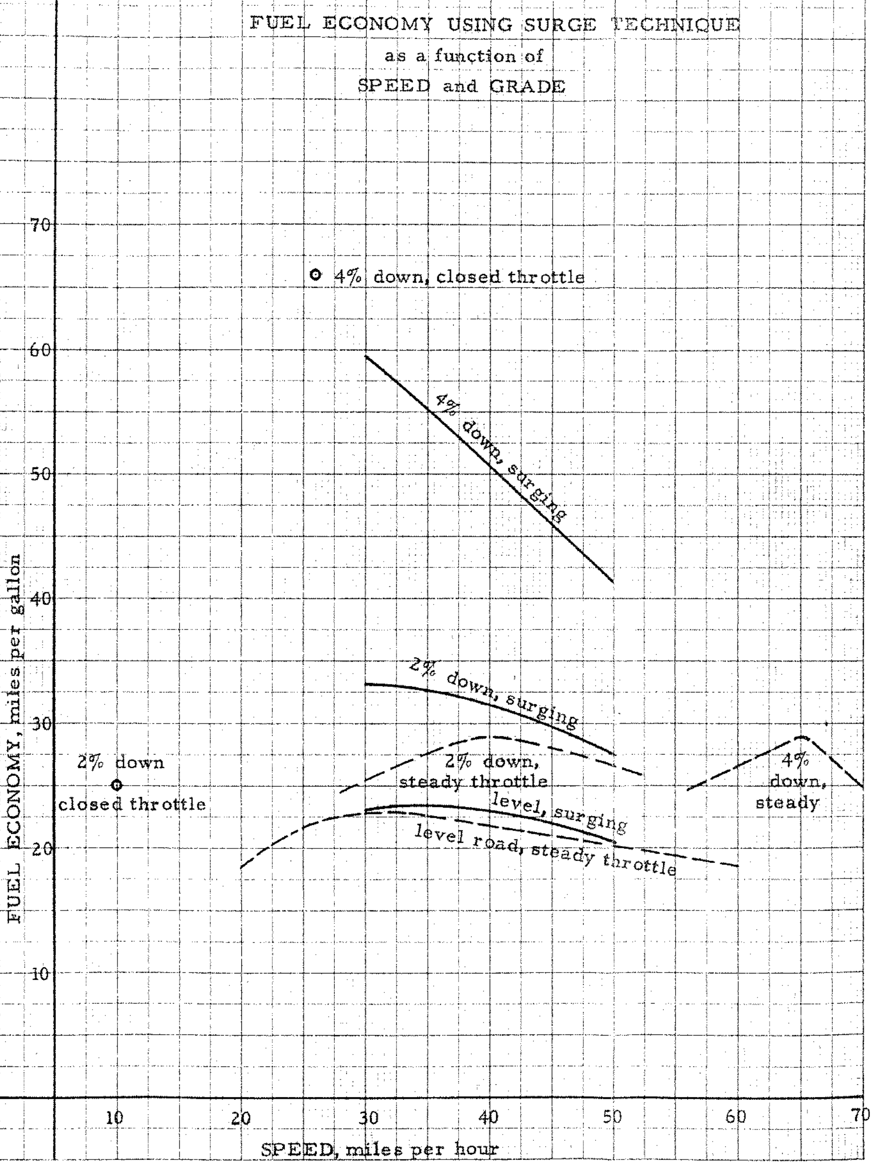

RETARDING FORCE
as a function of
ROAD SPEED
coasting with closed throttle
300
200
100
RETARDING FORCE, pounds
total force
road friction
engine friction
10
20
30
40
50
60
ROAD SPEED, miles per hour

OPTIMUM DOWNHILL SPEED
as a function of
GRADE and TECHNIQUE




DOWNHILL ECONOMY
as a function of
GRADE and TECHNIQUE
FUEL ECONOMY, miles per gallon
PERCENT DOWNGRADE
closed throttle coasting in high gear
steady throttle
160
140
120
100
80
60
40
20
2
4
6
8
10


FUEL ECONOMY USING SURGE TECHNIQUE
as a function of
SPEED and ENGINE TORQUE
40 mph
30 mph
50 mph
Indicated advantage over steady throttle(dots):
at 30 mph, .2 miles per gallon
at 40 mph, 1.1 " " "
at 50 mph, .3 " " "
FUEL ECONOMY, miles per gallon
24
20
16
12
8
4
40
80
120
160
200
ENGINE TORQUE, pound-feet


FUEL ECONOMY USING SURGE TECHNIQUE
as a function of
SPEED and GRADE
4% down, closed throttle
4% down, surging
2% down, surging
2% down, steady throttle
4% down, steady
2% down
closed throttle
level, surging
level road, steady throttle
FUEL ECONOMY, miles per gallon
SPEED, miles per hour
70
60
50
40
30
20
10
10
20
30
40
50
60
70

APPENDIX A

METHODS OF COMPUTATION

FIRST METHOD

Given: Engine speed, manifold depression, input torque, indicated miles per gallon (latter derived from fuel flow). To compute size factor, divide N_i by $\sqrt{T_i}$. From converter curves find N_o/N_i and T_o/T_i corresponding to this size factor, compute car speed by using relation between V, N_o, and N_o/N_i, such that $V = N_o/N_i \times N_i \ (\frac{N_o}{V})$ where $\frac{N_o}{V} = 44.5$ for axle ratio and tire size of subject vehicle. Then compute gross thrust by calculation of axle shaft torque divided by wheel radius, giving

$$P = \frac{\text{shaft torque x axle ratio}}{\text{rolling radius}} = \frac{T_o \times 3.55}{1.12 \text{ (feet)}} = 3.18\ T_o .$$

Road load thrust is obtained for each car speed and subtracted from the gross thrust to obtain new thrust available for grade ascension. The grade is then the angle whose sine is $\frac{\text{net thrust}}{\text{car weight}}$.

The actual miles per gallon is simply indicated MPG times speed ratio, N_o/N_i.

SECOND METHOD

Given: Car speed, percent grade ($\sin\alpha$). Find road friction drag from plot (Fig. 14) and add $W \sin\alpha$ to give total thrust necessary to ascend given grade at given speed. Divide by 3.18 to obtain torque at converter output. Take square root. Find converter output speed by $\frac{N_o}{V}$ relation, and divide N_o by $\sqrt{T_o}$ to give K_o. Use Figures 20 and 21 to determine the speed ratio, then Figure 13 to find torque ratio. Divide N_o by N_o/N_i to find engine speed (if desired) and divide T_o by T_o/T_i to find engine torque. Use these to find S. F. C. (use torque alone if working with Figure 4). Then

$$MPG = \frac{2390\ E_c}{P \times BSFC} \qquad \text{as previously noted.}$$

SURGE TECHNIQUE

A. With variable torque; level road.

At any speed, total decelerating force with closed throttle is given by Figure 25. Then deceleration is given by $d = \frac{P \times g}{W + 50}$ where the gravity constant is in mph/sec., and 50 pounds has been added to car weight to account for the moment of inertia of wheels and tires. Then d will also be given in mph/sec (pound-feet-second units could be used if desired). Assumption of a constant d makes it possible to use the simple equation $(V_1^2 - V_o^2) = 2$ ds to find the distance moved during deceleration. The average speed during deceleration was V, so S also = Vt, and $Q_d = .4\,t = .4\frac{S}{V}$.

For the acceleration period, the miles per gallon is computed for each torque value, as in second method above, but net thrust at speed V is also obtained as in method one. Then $a = \frac{P \times g}{W + 50}$ as above, and accelerating distance is

$$\frac{V_1^2 - V_o^2}{2\,a}$$

From this we find fuel used, $Q_a = S_a \div (MPG)_a$, and add the fuel and distance results to get totals for one cycle of operation, and divide the distance total by the gallonage total to find the economy.

B. With fixed torque, variable grade.

Computations are identical with above, except that only one line is necessary for each speed and grade.

MISCELLANEOUS COMPUTATIONS

$$\frac{dQ}{dt} = .00062N\,(29 - \Delta p)$$

$$T = 10\,(23.5 - \Delta p)\ ;\ \Delta p = 23.5 - .1\,T$$

$$BSFC = \frac{\text{pounds of fuel}}{BHP - HR} = \frac{\text{lb fuel/hr}}{BHP}$$

$$= \frac{.00062N\,\left[29 - (23.5 - .1\,T)\right]}{\dfrac{2\pi\ TN}{33000}} = \frac{33{,}000 \times .00062\,(5.5 + .1\,T)}{6.28\,T}$$

$$= \frac{.179 + .326\,T}{T} = .326 + \frac{.179}{T}$$

$$\frac{dQ}{dt} = .00062N\,(29 - \Delta p)\ \text{lb/hr} = \frac{.00062}{6.375}\,N\,(29 - \Delta p)\ \text{gallons/hr.}$$

$$\text{car speed} = \frac{N}{44.5}\ (1 - \text{slip})\ MPH$$

$$MPG = \frac{\dfrac{N}{44.5}\ (1 - \text{slip})}{\dfrac{.00062N\,(29 - \Delta p)}{6.372}} = \frac{36.2\,(1 - \text{slip})\,6.375}{(29 - \Delta p)}$$

$$= \frac{231\,(1 - \text{slip})}{29 - \Delta p}$$

$$3\%\ \text{slip:}\quad MPG = \frac{.97 \times 231}{29 - \Delta p} = \frac{224}{29 - \Delta p}$$

$$10\%\ \text{slip:}\quad MPG = \frac{.90 \times 231}{29 - \Delta p} = \frac{208}{29 - \Delta p}$$

$$\text{Efficiency of Converter} = \frac{\text{output}}{\text{input}} = \frac{N_o \times T_o}{N_i \times T_i} = \frac{N_o}{N_i} \times \frac{T_o}{T_i}$$

$$K_o = \frac{N_o}{\sqrt{T_o}} = \frac{N_i \times \dfrac{N_o}{N_i}}{\sqrt{T_i \dfrac{T_o}{T_i}}} = \frac{N_i}{\sqrt{T_i}} \times \frac{\dfrac{N_o}{N_i}}{\sqrt{\dfrac{T_o}{T_i}}} = K \frac{\dfrac{N_o}{N_i}}{\sqrt{\dfrac{T_o}{T_i}}}$$

But $\dfrac{T_o}{T_i}$ is a unique function of $\dfrac{N_o}{N_i}$, as is K

$\therefore$ K_o is also a unique function of $\dfrac{N_o}{N_i}$

$$\text{Road HP} = P \times \frac{88}{60}\,\frac{V}{550} = \frac{88PV}{33000}$$

$$\text{Engine HP} = \frac{\text{Road HP}}{\text{converter efficiency}} = \frac{88\ PV}{33{,}000\ E_c}$$

$$\text{Fuel Flow} = \text{BSFC} \times \text{HP} = \text{BSFC} \times \frac{88\ PV}{33{,}000\ E_c} \quad \text{(pounds/hr)}$$

$$\text{gal/hr} = \frac{\#/\text{hr}}{\#/\text{gal}} = \frac{\text{BSFC}}{6.375}\ \frac{88\ PV}{33{,}000\ E_c} \quad \text{(gallons/hr)}$$

$$\text{miles per gallon} = \frac{\text{miles}}{\text{gallons}} = \frac{\text{miles/hr}}{\text{gallons/hr}}$$

$$mpg = \frac{V}{\frac{BSFC}{6.375} \quad \frac{88\,PV}{33,000\,E_c}} = \frac{33,000 \times 6.375\,E_c}{88\,P \times BSFC}$$

$$= \frac{2390\,E_c}{P \times BSFC}$$

Alternate: $$MPG = \frac{2390\,E_c}{P \times BSFC} = \frac{2390\,E_c}{3.19\,T_o \times SFC} = \frac{748\,E_c}{T_o \times SFC}$$

but $E_c = N_o/N_i \times T_o/T_i$

$$MPG = \frac{748 \times N_o/N_i \times T_o/T_i}{T_o \times SFC} = \frac{748\,N_o/N_i}{T_i \times SFC}$$

Acceleration and Deceleration Distance:

$$S_a = \frac{V_2^{\,2} - V_1^{\,2}}{2\,a} \;; \quad \text{units are} \quad \frac{\frac{miles}{hours}}{\frac{miles}{hours\text{-}sec.}} = \frac{miles\text{-}seconds}{hours}$$

To get distance in miles, multiply by hours/second $\left(\frac{1}{3600}\right)$

$$S_a = \frac{V_2^{\,2} - V_1^{\,2}}{7200\,a}$$: V_1 & V_2 in mph

a in mph/sec.

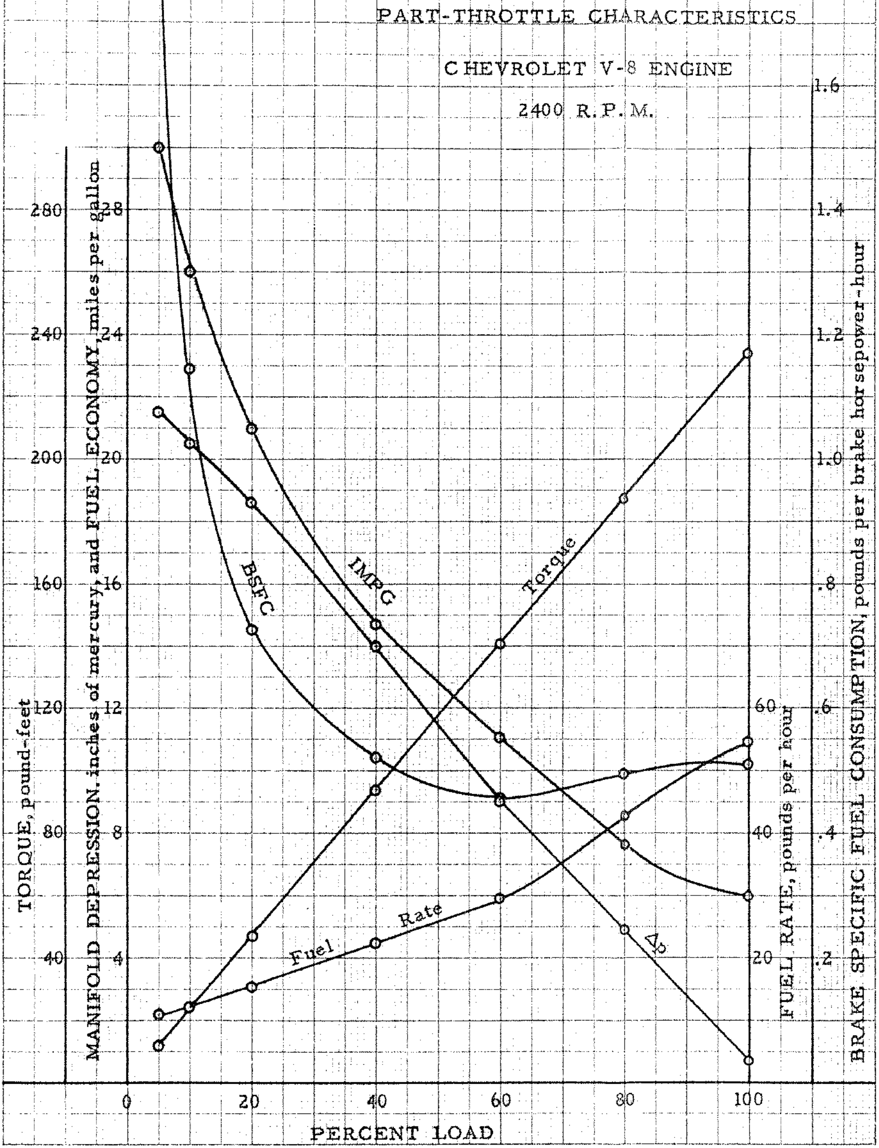
For deceleration, same equation holds

Example: at 40 mph, $S_a = \frac{45^2 - 35^2}{7200\,a} = \frac{2025 - 1225}{7200\,a} = \frac{1}{9\,a}$

Acceleration and Deceleration:

$$a = \frac{force}{mass} = \frac{net\ thrust}{\frac{w + 50}{g}} = \frac{P_n \times 32.2}{3800} \times \frac{60}{88} \quad \frac{mph}{ft/sec}$$

$$= .00578\,P_n\ (mphps)$$


PART-THROTTLE CHARACTERISTICS
CHEVROLET V-8 ENGINE
2400 R.P.M.
TORQUE, pound-feet
MANIFOLD DEPRESSION, inches of mercury, and FUEL ECONOMY, miles per gallon
FUEL RATE, pounds per hour
BRAKE SPECIFIC FUEL CONSUMPTION, pounds per brake horsepower-hour
PERCENT LOAD
BSFC
MPG
Torque
Fuel Rate
Δp
0
20
40
60
80
100
280
240
200
160
120
80
40
28
24
20
16
12
8
4
60
40
20
1.6
1.4
1.2
1.0
.8
.6
.4
.2

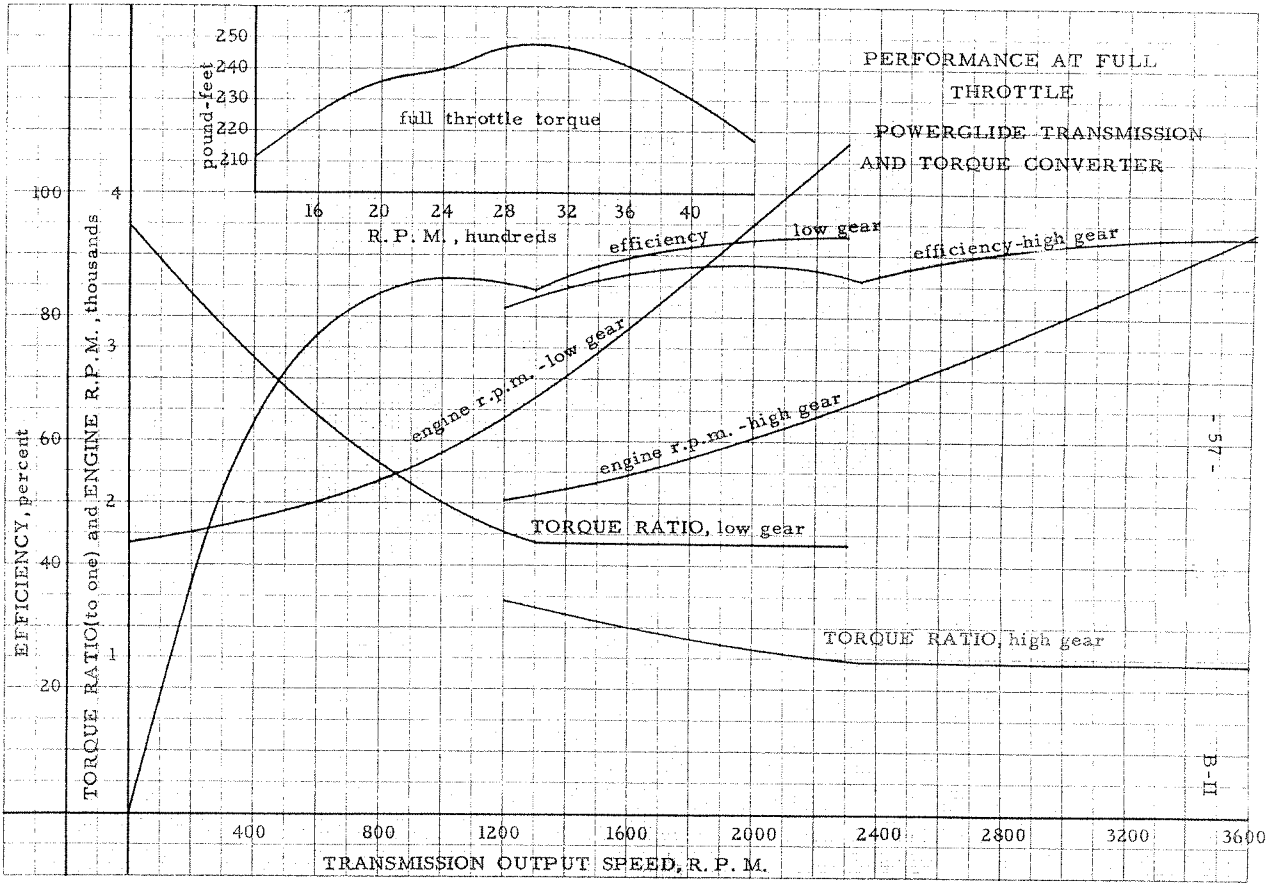

PERFORMANCE AT FULL
THROTTLE
POWERGLIDE TRANSMISSION
AND TORQUE CONVERTER
full throttle torque
pound-feet
250
240
230
220
210
16
20
24
28
32
36
40
R. P. M. , hundreds
efficiency
low gear
efficiency-high gear
engine r.p.m. -low gear
engine r.p.m. -high gear
TORQUE RATIO, low gear
TORQUE RATIO, high gear
EFFICIENCY, percent
100
80
60
40
20
TORQUE RATIO(to one) and ENGINE R.P.M., thousands
4
3
2
1
400
800
1200
1600
2000
2400
2800
3200
3600
TRANSMISSION OUTPUT SPEED, R. P. M.

TABLE I

COMPUTATION OF TORQUE CONVERTER CHARACTERISTICS

FULL THROTTLE

Col. No.	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
Quantity	Gear Ratio	Output Speed	Input Speed	Input Torque	Indicated Efficiency	Indicated Torque Ratio		Size Factor	Converter Output Speed	Speed Ratio	Total Losses	Coupling Loss	Trans. Loss	Torque Loss	Indicated Output Torque	Loss Plus Output	Converter Output Torque	Torque Ratio	Efficiency of Converter
Symbol		N_o	N_i	T_i	(%)		$\sqrt{T_i}$	K	N_{co}	N_{co}/N_i	(%)	(%)	(%)			T_o	T_{co}	T_{co}/T_i	E_c
Source	← From Transmission Curves →						$\sqrt{(4)}$	(3)÷(7)	(2)÷(1)	(9)÷(3)	1.00-(5)	1.00 - (10)	(11) - (12)	(13) x (4)	(4) x (6)	(14) + (15)	(16)÷(1)	(17)÷(4)	(18) x (10)
	1.82	80	1765	231	16	3.60	15.2	116	146	.083				6.8 (from Fig. 10)	831	838	460	1.99	16.5
		200	1800	232	37	3.33	15.25	118	364	.202				6.95	773	780	428	1.845	37.2
		400	1890	234	63.3	3.00	15.3	123.5	728	.385				7.2	702	709	389	1.66	64.0
		600	1985	236	77.5	2.57	15.38	129	1090	.548				7.4	607	614	337	1.43	78.3
		700	2055	237	81.4	2.39	15.4	133.5	1275	.62				7.5	566	574	315	1.33	82.4
		800	2130	238	83.7	2.225	15.44	138	1456	.682				7.6	529	537	295	1.24	84.6
		1000	2320	239	86.0	2.00	15.48	150	1820	.784				7.85	478	486	267	1.12	87.7
		1200	2550	243	85.6	1.81	15.6	163.5	2185	.856				8.05	440	448	246	1.01	86.5
		1400	2820	247	86.9	1.75	15.72	179.5	2545	.902	13.1	9.8	3.3	8.2					
		1600	3125	248	89.4	1.75	15.75	198.5	2910	.932	10.6	6.8	3.8	9.4					
		1700	3290	246	90.6	1.75	15.70	209.5	3090	.939	9.4	6.1	3.3	8.2					
		1800	3445	244	91.5	1.75	15.63	220.5	3275	.951	8.5	4.9	3.6	8.8					
		2000	3790	236	92.3	1.75	15.36	247	3640	.96	7.7	4.0	3.7	8.7					
		2200	4140	226	92.6	1.745	15.04	275	4000	.966	7.4	3.4	4.0	9.0					
	↓	2300	4320	219	92.7	1.745	14.81	291.5	4190	.97	7.3	3.0	4.3	9.4			-	1.38	81.9
	1.00	1200	2020	236	81.5	1.37	15.38	131.5	same as N_o	.594				3.35 (from Fig. 10)	323	326	-	1.3	86.1
		1400	2110	237	85.0	1.28	15.4	137		.663				3.55	303	307	-	1.21	88
		1600	2200	238	87.0	1.2	15.44	142.5		.727				3.7	285	289	-	1.15	89.5
		1800	2310	239	87.8	1.13	15.48	149		.779				3.9	270	274	-	1.12	89.6
		1900	2370	240	88.5	1.10	15.50	153		.801				4.0	264	268	-	1.09	89.7
		2000	2430	241	88.5	1.07	15.53	156.5		.823				4.1	258	262	-	1.03	88.8
		2200	2550	243	87.5	1.01	15.6	163.5		.863				4.3	246	250			
		2400	2705	246	87.0	.98	15.7	172		.888	13	11.2	1.8	4.5					
		2600	2860	248	89.1	.98	15.75	181.5		.909	10.9	9.1	1.8	4.5					
		2700	2940	248	90	.98	15.75	186.5		.918	10	8.2	1.8	4.5					
		2800	3025	248	90.6	.98	15.75	192		.926	9.4	7.4	2.0	5					
		3000	3200	247	91.9	.97	15.72	203.5		.9375	8.1	6.25	1.85	4.6					
		3200	3370	245	92.2	.97	15.67	215		.95	7.8	5.0	2.8	7					
		3400	3560	242	93.0	.97	15.56	229		.955	7	4.5	2.5	6.1					
	↓	3600	3740	237	94.0	.97	15.4	243		.962	6	3.8	2.2	5.2					

HALF THROTTLE

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
1.82	0	1740	217	0	3.75	14.73	118	0	0									
	75	1745	217	15	3.59	14.73	118.5	136.5	.078			from Fig. 10	4.9	778	783	430	1.98	15.5
	200	1760	217	37.8	3.33	14.73	119.5	364	.208				5.05	722	727	399	1.84	38.3
	400	1810	216	64	2.89	14.70	123	728	.402				5.25	624	629	345	1.6	64.3
	600	1910	215	78.6	2.50	14.66	130	1090	.57				5.45	538	543	298	1.39	79.2
	800	2060	213	84.5	2.18	14.60	141	1456	.705				5.65	464	470	258	1.21	85.3
	900	2150	212	86.5	2.06	14.56	148	1640	.762				5.75	437	443	243	1.15	87.6
	1000	2240	210	86.6	1.94	14.50	154.5	1820	.813				5.85	407	413	227	1.08	87.8
	1100	2350	208	85.5	1.82	14.43	163	2000	.85				5.90	378	384	211	1.015	86.1
	1200	2465	206	85.7	1.76	14.35	172	2185	.886	14.3	11.4	2.9	6					
	1400	2745	200	89.6	1.755	14.14	194	2545	.927	10.4	7.3	3.1	6.2					
	1500	2900	196	90.7	1.75	14.0	207	2730	.940	9.3	6	3.3	6.5					
1.00	800	1845	216	69.3	1.6	14.70	125.5		.434			from Fig. 10	3	345	348	-	1.61	69.8
	1000	1885	215	77.6	1.46	14.66	128.5		.531				3.15	314	317	-	1.475	78.2
	1200	1945	214	82.9	1.35	14.63	133		.617				3.35	289	292	-	1.365	84.2
	1400	2030	213	85.8	1.24	14.60	139		.69				3.55	264	268	-	1.26	87
	1500	2075	213	86.1	1.19	14.60	142		.722				3.6	253	257	-	1.205	87
	1600	2120	212	87	1.15	14.56	146		.754				3.7	244	248	-	1.17	88
	1800	2230	210	87.8	1.09	14.5	154		.807				3.9	229	233	-	1.11	89.5
	2000	2345	208	86.7	1.01	14.43	162.5		.853				4.1	210	214	-	1.03	87.8
	2200	2475	206	87.1	.98	14.35	172.5	same as N_o	.889	12.9	11.1	1.8	3.7					
	2400	2625	203	89	.97	14.25	184		.915	11	8.5	2.5	5.1					
	2500	2710	201	90	.97	14.18	191		.923	10	7.7	2.3	4.6					
	2600	2790	199	91	.97	14.10	198		.931	9	6.9	2.1	4.2					
	2800	2965	195	92	.97	13.96	212.5		.944	8	5.6	2.4	4.7					
	3000	3145	190	92.6	.97	13.78	228		.954	7.4	4.6	2.8	5.3					
	3200	3325	184	93.4	.97	13.57	245		.962	6.6	3.8	2.8	5.15					
	3400	3520	178	94.0	.97	13.33	264		.966	6	3.4	2.6	4.6					
	3600	3720	172	94.5	.97	13.12	283.5		.968	5.5	3.2	2.3	4					

QUARTER THROTTLE

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
1.82	0	1375	144	3.77	114.5	12.0	0	0	0									
	65	1390	143	3.6	116	11.96	16	118	.085				from Fig. 10 3.4	515	518	285	1.99	16.9
	200	1430	142	3.22	120	11.92	45.6	364	.255				3.55	457	461	253	1.78	45.3
	400	1490	139	2.69	126.5	11.78	72.3	728	.488				3.75	374	378	208	1.5	73.2
	600	1600	134	2.235	138.5	11.57	83.6	1090	.681				3.95	299	303	166.5	1.24	84.5
	700	1675	129.5	2.05	147	11.38	85.2	1275	.76				4.05	265	269	148	1.145	87
	800	1750	124	1.88	157.5	11.13	85.6	1456	.83				4.15	233	237	130	1.05	87
	1000	1985	112	1.74	187.5	10.58	87.8	1820	.916	12.2	8.4	3.8	4.25					
	1200	2290	98	1.73	231.5	9.9	90.7	2185	.954	9.3	4.6	4.7	4.6					
	1300	2460	87	1.72	263.5	9.34	90.5	2365	.961	9.5	3.9	5.6	4.9					
	1400	2630	78.5	1.71	297	8.87	91.0	2545	.967	9.0	3.3	5.7	4.5					
	1500	2810	70	1.69	335	8.38	90.1	2730	.971	9.9	2.9	7.0	4.9					
1.00	400	1435	141	1.77	121	11.87	49.3	same as N_o	.279				from Fig. 10 2.6	249	252	-	1.79	50
	600	1470	140	1.61	124	11.84	66.1		.408				2.8	225	228	-	1.63	66.5
	800	1510	138	1.475	128.5	11.75	78.1		.53				3.0	204	207	-	1.5	79.5
	900	1540	137	1.35	131.5	11.71	79		.585				3.1	185	188	-	1.37	80.1
	1000	1570	135.5	1.31	135	11.64	83.6		.637				3.15	177	180	-	1.33	84.8
	1100	1600	134	1.24	138	11.57	85.4		.6875				3.25	166	169	-	1.26	86.6
	1200	1635	132	1.175	142	11.49	86.3		.734				3.35	155	158	-	1.2	88
	1400	1735	126	1.07	154.5	11.23	86.1		.807				3.55	135	139	-	1.1	88.8
	1600	1825	120	.97	166.5	10.95	85.0		.876	15.0	12.4	2.6	3.10					
	1800	1965	112.5	.965	185	10.6	88.2		.916	11.8	8.4	3.4	3.8					
	1900	2045	109	.96	195.5	10.44	89.1		.930	10.9	7	3.9	4.25					
	2000	2130	104	.965	209	10.2	90.8		.939	9.2	6.1	3.1	3.2					
	2100	2210	100	.95	221	10	90.0		.95	10	5	5	5.0					
	2200	2300	95	.95	235.5	9.76	91		.956	9	4.4	4.6	4.4					
	2400	2490	85.5	.95	269	9.26	91.5		.964	8.5	3.6	4.9	4.2					
	2600	2680	76	.94	307	8.73	91.2		.97	8.8	3	5.8	4.4					
	2800	2870	67	.92	350	8.2	89.9		.976	10.1	2.4	7.7	5.2					
	3000	3060	59	.91	398	7.69	89.0		.98	11	2	9	5.4					
	3200	3250	51	.89	455	7.15	87.8		.984	12.2	1.6	10.6	5.4					
	3400	3450	44	.885	519	6.64	86.2		.985	13.8	1.5	12.3	5.4					
	3600	3640	38	.85	590	6.17	84.6		.989	15.4	1.1	14.3	5.4					

TABLE II

FIRST METHOD OF COMPUTATION

Col. No.	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
Item	Percent Load	Engine Speed	Manifold Depression	Input Torque	Indicated M. P. G.		Size Factor	Speed Ratio	Torque Ratio	Car Speed	Gross Thrust	Road Load Friction	Net Thrust	Grade	Actual M. P. G.
Symbol		N_i	Δp	T_i	IMPG	$\sqrt{T_i}$	K	N_o/N_i	T_o/T_i	V	P	P_f	P_n	$\sin\alpha$	M. P. G.
Source	← From Engine Data →					$\sqrt{(4)}$	$(2)\div(6)$	Fig. 13	Fig. 13	$\frac{N_o}{N_i} \times \frac{N_i}{445}$	$3.18\ T_i \times \frac{T_o}{T_i}$	Fig. 14	(11) - (12)	$(13)\div W$	(5) x (8)
	5	800	19.5	9.5	20.9	3.08	259.5	.965	1.00	17.3	30.3	66	-35.7	-.0095	20.2
	10		19.4	19.1	19.9	4.37	183	.918	1.00	16.4	61	65	-4	-.0001	18.25
	20		18.1	38.2	17.7	6.18	129.5	.55	1.43	9.85	174	59	115	.0307	9.75
	5	1200	21.2	10.4	26.7	3.23	372	.978	1.00	26.2	33.2	79.5	-46.3	-.0123	26.1
	10		21.0	20.8	24.3	4.57	262.5	.965	1.00	25.9	66.4	79	-12.6	-.0033	23.45
	20		19.1	41.6	20.55	6.45	186	.919	1.00	24.6	133	77	56	.015	18.9
	40		14.7	83.2	15.0	9.13	131.5	.59	1.38	15.8	366	64.5	301.5	.084	8.85
	5	1600	22.0	11.05	29.6	3.33	480	.984	1.00	35.2	35.2	98	-63	-.0168	29.1
	10		21.1	22.1	26.3	4.7	340	.975	1.00	34.9	70.5	97.5	-27	-.0072	25.6
	20		19.1	44.2	21.75	6.65	241	.960	1.00	34.3	141	96	45	.0120	20.9
	40		14.75	88.4	14.75	9.41	170	.88	1.00	31.5	282	90	192	.0512	13.0
	60		10.1	132.6	11.4	11.53	139	.70	1.235	25	522	77.5	444.5	.1185	8.0
	80		5.15	176.8	6.95	13.32	120	.25	1.815	8.9	1025	49.5	975	.2600	1.75
	5	2000	21.5	11.4	29.85	3.38	592	.989	1.00	44.2	36.4	122	-85.6	-.0228	29.5
	10		20.65	22.8	26.6	4.78	418	.982	1.00	43.9	72.7	121	-48.3	-.0129	26.1
	20		18.85	45.5	20.95	6.75	296	.969	1.00	43.3	145	119	26	+.0069	20.3
	40		14.3	91	14.75	9.55	209.5	.942	1.00	42.1	290	110	180	.0480	13.9
	60		9.5	136.5	11.35	11.69	171	.885	1.00	39.5	435	96	339	.0905	10.05
	80		5.2	182	7.1	13.5	148	.775	1.135	34.6	659	85	574	.1532	5.5
	100		.5	227.5	6.15	15.1	132.5	.61	1.35	27.2	980	71	809	.216	3.75
	5	2400	21.5	11.7	30.0	3.42	702	.99	1.00	53.2	37.3	150	-112.7	-.0300	29.7
	10		20.5	23.4	26.0	4.84	496	.985	1.00	52.9	74.6	149	-74.4	-.0198	25.6
	20		18.6	46.8	20.95	6.85	351	.976	1.00	52.4	149.5	147.5	2	0	20.5
	40		14.0	93.6	14.65	9.67	248	.962	1.00	51.7	299	145	144	+.0384	14.1
	60		9.0	140.5	11.05	11.86	202.5	.936	1.00	50.3	448	141	307	+.0819	10.35
	80		4.9	187.5	7.65	13.7	175	.903	1.00	48.5	598	135	463	.1235	6.9
	100		.75	234	6.0	15.32	156.5	.825	1.07	44.3	798	122	676	.1804	4.95
	10	2800	20.4	23.9	26.5	4.89	573	.988	1.00	61.9	76.2	180	-103.8	-.0277	26.15
	20		18.35	47.8	20.85	6.92	405	.980	1.00	61.4	152.5	178	-25.5	-.0068	20.45
	40		13.6	95.6	14.3	9.78	286	.968	1.00	60.6	305	176	129	.0344	13.85
	60		8.8	143.4	9.85	11.97	234	.957	1.00	60.0	457	173	284	.0757	9.45
	80		4.7	191.2	8.05	13.84	202	.936	1.00	58.6	610	168	442	.1179	7.55
	100		.95	239	6.25	15.48	181	.912	1.00	57.1	762	163	599	.1600	5.7

TABLE II A

	N_i	Δp	MPG	V
Level Road:	800	19.4	18.3	16.3
	1200	20.8	23.0	25.8
	1600	20.3	23.6	34.6
	2000	19.4	21.8	43.5
	2400	18.6	20.5	52.5
	2800	17.5	18.9	61.1
2% Grade:	1200	19.1	18.9	24.7
	1600	18.2	19.0	33.7
	2000	17.4	17.5	43.0
	2400	16.3	16.6	52.2
	2800	15.3	15.6	60.8
4% Grade:	1200	17.5	15.2	22.5
	1600	15.9	14.8	32.3
	2000	15.1	14.6	42.4
	2400	13.7	13.8	51.7
	2800	12.9	13.1	60.5
6% Grade:	1200	16.2	12.3	20.0
	1600	14.0	12.1	30.5
	2000	12.9	12.6	41.5
	2400	11.4	12.1	51.2
	2800	10.6	11.1	60.2
8% Grade:	1200	14.9	9.4	16.5
	1600	12.4	10.4	28.6
	2000	10.5	10.8	40.0
	2400	9.2	10.5	50.5
	2800	8.4	9.4	59.8
10% Grade:	1200	14±	7±	12.5±
	1600	11.1	9.1	26.7
	2000	8.7	9.4	38.5
	2400	7.1	8.7	49.6
	2800	6.4	8.3	59.4
2% Down-Grade	1600	22.2	30±	35
	2000	21.2	28.2	44.1
	2400	20.5	25.6	52.9
	2800	19.6	23.7	61.7

TABLE III

SECOND METHOD OF COMPUTATION

Col. No.	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
Item	Car Speed	Shaft Speed	Friction Drag	Grade Load	Total Load	Shaft Torque		Size Factor	Speed Ratio	Torque Ratio	Engine Torque	Engine Speed	Specific Fuel Consumption	Converter Efficiency	Miles per Gallon
Symbol	V	N_o	P_f	P_g	P	T_o	$\sqrt{T_o}$	K_o	N_o/N_i	T_o/T_i	T_i	N_i	S. F. C.	E_c	M. P. G.
Source		(1) x 44. 5	Fig. 14	W sin α	(3) + (4)	(5) ÷ 3. 18	$\sqrt{(6)}$	(2) ÷ (7)	Fig. 13	Fig. 13	(6) ÷ (10)	(2) ÷ (9)	*	Fig. 13	**
Level Road	20	890	98	none	98	30. 7	5. 55	160	.905	1. 00	same as (6)	985	1. 20		18. 4
	24	1070	105		105	32. 9	5. 74	186	.934			1150	1. 0		21. 2
	28	1250	113		113	35. 5	5. 97	209	.950			1320	.89	same	22. 6
	32	1425	122		122	38. 3	6. 20	230	.959			1490	.82		22. 9
	36	1605	131		131	41. 1	6. 92	250	.964			above	.78	as	22. 55
	40	1780	141		141	44. 2	6. 66	267	.967			1600	.75		21. 85
	44	1960	151		151	47. 3	6. 89	284	.969				.72	(9)	21. 3
	48	2135	162		162	50. 8	7. 13	299	.971				.70		20. 5
	52	2315	175		175	54. 9	7. 42	312	.973				.67		19. 8
	56	2490	188		188	59. 0	7. 69	324	.974				.64		19. 35
	60	2670	202		202	63. 4	7. 98	335	.975				.62		18. 6
1% Grade	16	710	91	38	129	40. 6	6. 37	111	.78	1. 125	36	910	1.05	.877	15. 8
	20	890	98		136	42. 8	6. 54	136	.85	1. 03	31. 5	1050	.90	.875	17. 1
	24	1070	105		143	45	6. 71	160	.905	1. 00	45	1180	.80	.905	18. 9
	28	1250	113		151	47. 5	6. 89	181	.929		47. 5	1350	.75	.929	19. 6
	32	1425	122		160	50. 3	7. 09	201	.945		50. 3	1510	.71	.945	19. 8
	36	1605	131		169	53. 2	7. 29	220	.955		53. 2	over 1600	.69	.955	19. 7
	40	1780	141		179	56. 3	7. 5	237	.961		56. 3		.66	.961	19. 2
	44	1960	151		189	59. 4	7. 71	254	.964		59. 4		.64	.964	19
	48	2135	162		200	62. 9	7. 93	270	.967		62. 9		.62	.967	18. 6
	52	2315	175		213	67	8. 19	283	.969		67		.60	.969	18. 1
	56	2490	188		226	71. 1	8. 43	296	.971		71. 1		.58	.971	17. 7
	60	2670	202		240	75. 5	8. 69	307	.972		75. 5		.565	.972	17. 1

2 1/2% Grade	16	710	91	94	185	58.2	7.63	93	.72	1.21	48	985	.85	.871	13.25
	20	890	98		192	60.4	7.77	115	.79	1.11	54.5	1130	.74	.877	14.75
	21	1070	105		199	62.6	7.91	135	.845	1.04	60	1270	.68	.879	15.5
	28	1250	113		207	65.1	8.07	155	.896	1.00	65.1	1400	.65	.896	15.95
	32	1425	122		216	67.9	8.24	173	.921		67.9	1550	.6	.921	17.0
	36	1605	131		225	70.8	8.41	191	.938		70.8	above 1600	.58	.938	17.2
	40	1780	141		235	73.9	8.6	207	.948		73.9		.57	.948	16.9
	44	1960	151		245	77.1	8.78	223	.957		77.1		.56	.957	16.7
	48	2135	162		256	80.5	8.97	238	.961		80.5		.55	.961	16.3
	52	2315	175		269	84.6	9.2	252	.964		84.6		.535	.964	16.0
	56	2490	188		282	88.7	9.42	265	.966		88.7		.52	.966	15.75
	60	2670	202		296	93.2	9.65	277	.968		93.2		.51	.968	15.3
5% Grade	16	710	91	187	278	87.0	9.35	76	.64	1.31	66.5	1110	.60	.838	12.0
	20	890	98		285	89.4	9.47	94	.72	1.205	74.0	1235	.57	.867	12.7
	24	1070	105		292	91.5	9.58	112	.785	1.12	81.5	1360	.545	.879	13.2
	28	1250	113		300	94	9.71	129	.83	1.06	88.7	1510	.525	.880	13.4
	32	1425	122		309	97	9.86	145	.865	1.015	95.5	above 1600	.51	.877	13.3
	36	1605	131		318	99.7	9.98	161	.906	1.00	99.7		.50	.906	13.6
	40	1780	141		328	102.7	10.13	176	.924		102.7		.493	.924	13.65
	44	1960	151		338	106.0	10.30	190	.937		106.0		.486	.937	13.6
	48	2135	162		349	109.4	10.46	204	.947		109.4		.482	.947	13.45
	52	2315	175		362	113.4	10.64	217	.954		113.4		.476	.954	13.2
	56	2490	188		375	117.5	10.84	230	.959		117.5		.47	.959	13.0
	60	2670	202		389	122.0	11.04	242	.962		122.0		.467	.962	12.6
7 1/2% Grade	16	710	91	281	374	117	10.8	66	.58	1.39	84	1220	.57	.806	9.1
	20	890	98		379	119	10.9	82	.67	1.275	93	1330	.54	.854	9.95
	24	1070	105		386	121	11.0	97	.73	1.19	102	1470	.52	.868	10.35
	28	1250	113		394	124	11.14	112	.785	1.12	111	1590	.49	.88	10.9
	32	1425	122		403	127	11.27	127	.852	1.065	119	1730	.47	.878	11.1
	36	1605	131		412	129.5	11.38	141	.86	1.02	127		.462	.876	11.0
	40	1780	141		422	133	11.53	154.5	.895	1.00	133	above 1600	.46	.895	11.0
	44	1960	151		432	136	11.66	168	.915		136		.458	.915	11.05
	48	2135	162		443	139	11.79	181	.929		139		.458	.929	10.95
	52	2315	175		456	143	11.96	194	.940		143		.458	.940	10.75
	56	2490	188		469	147.5	12.14	205	.947		147.5		.458	.947	10.55
	60	2670	202		483	152	12.33	216	.952		152		.458	.952	10.3

10% Grade	16	710	91	375	466	147	12.12	58.5	.54	1.45	101	1310	.525	.783	7.65
	20	890	98		473	149	12.21	73	.625	1.33	112	1425	.50	.831	8.4
	24	1070	105		480	151	12.29	87	.69	1.245	121	1550	.485	.859	8.8
	28	1250	113		488	153	12.37	101	.75	1.165	131	over 1600	.465	.874	9.2
	32	1425	122		497	156	12.49	114	.79	1.11	141		.465	.877	9.1
	36	1605	131		506	159	12.61	127	.825	1.065	149		.48	.879	8.65
	40	1780	141		516	162	12.73	140	.86	1.02	159		.50	.877	8.1
	44	1960	151		526	165	12.85	153	.89	1.00	165		.49	.89	8.25
	48	2135	162		537	169	13.0	164	.91		169		.475	.91	8.55
	52	2315	175		550	173	13.15	176	.924		173		.465	.924	8.6
	56	2490	188		563	177	13.3	187	.934		177		.46	.934	8.6
	60	2670	202		577	181	13.45	198	.943		181		.46	.943	8.5
-1% Down-Grade	24	1070	105	-37.5	67.5	21.2	4.61	232	.96	1.0	21.2	1115	1.5	.96	22.7
	28	1250	113		75.5	23.7	4.88	256	.965		23.7	1300	1.2	.965	25.5
	32	1425	122		84.5	26.5	5.15	277	.968		26.5	1470	1.02	.968	26.9
	36	1605	131		93.5	29.4	5.43	295	.971		29.4	1650	.95	.971	26.1
	40	1780	141		103.5	32.6	5.72	311	.973		32.6	over 1600	.89	.973	25.2
	44	1960	151		113.5	35.7	5.99	327	.975		35.7		.84	.975	24.4
-2% Down-Grade	28	1250	113	-75	38	12	3.47	360	.978	1.0	12		2.5±	.978	24.6
	32	1425	122		47	14.8	3.85	370	.9785		14.8		1.9	.9785	26.2
	36	1605	131		56	17.6	4.2	380	.979		17.6		1.5	.979	27.9
	40	1780	141		66	20.7	4.56	390	.980		20.7		1.23	.980	28.9
	44	1960	151		76	23.9	4.9	400	.9805		23.9		1.09	.9805	28.3
	48	2135	162		87	27.3	5.23	408	.981		27.3		.99	.981	27.3
	52	2315	175		100	31.8	5.65	410	.981		31.8		.90	.981	26.0
-4% Down-Grade	56	2490	188	-150	38	12.0					same	above	2.5±		24.7
	60	2670	202		52	16.3					as	1600	1.7	about	26.5
	65	2890	220		70	22.0					(6)		1.16	.97	28.9
	70	3120	241		91	28.6							.96		26.9

* Below 1600 rpm Engine Speed, use Fig. 6, above 1600 Fig. 4.

** $\frac{2390 \times (14)}{(5) \times (13)}$

TABLE IV

SURGE TECHNIQUE--VARIABLE TORQUE

Col. No.	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)
Item	Shaft Torque		Size Factor	Speed Ratio	Torque Ratio	Engine Torque	Engine Speed	S. F. C.	M. P. G.	Gross Thrust	Net Thrust	Acceleration	Accel. Distance	Accel. Fuel	Total Distance	Total Fuel	M. P. G.
Symbol	T_o	$\sqrt{T}$	K_o	N_o/N_i	T_o/T_i	T_e	N_e	S. F. C.	M. P. G.	P	P_n	a	S_a	Q_a	S	Q	M. P. G.
Source		$\sqrt{(1)}$	$N_o \div (2)$	Fig. 13	Fig. 13	$(1) \div (5)$	$N_o \div (4)$	Fig. 4	**	3.18 x (1)	(10)-P_d	.00578 x (10)	*	$(13) \div (9)$	$S_a + S_d$	$Q_a + Q_d$	$(15) \div (16)$
30 mph	49	7	191	.938	1.00	49	1425	.74	19.3	156	39	.485	.172	.00890	.210	.0094	22.3
N_o = 1340	64	8	167	.914	1.00	64	1460	.66	16.2	204	87	1.08	.077	.00475	.115	.00526	21.9
P_d = 169	81	9	148	.87	1.00	81	1535	.58	13.85	258	141	1.75	.0476	.00344	.086	.00395	21.8
d = 1.00	100	10	133.5	.84	1.045	95.5	1590	.53	12.4	319	202	2.51	.0332	.00268	.0716	.00319	22.4
S_d = .0384	121	11	121	.81	1.085	112	over	.49	11.0	386	269	3.34	.0249	.00226	.0633	.00277	22.9
Q_d = .00051	144	12	111	.78	1.125	128	1600	.47	9.7	459	342	4.25	.0196	.00202	.0580	.00253	22.9
	169	13	103	.755	1.16	146		.48	8.1	539	422	5.25	.0159	.00196	.0543	.00247	22.0
	196	14	95	.725	1.20	163		.53	6.3	625	508	6.3	.0132	.00209	.0516	.00260	19.8
	225	15	89	.70	1.23	183		.57	5.0	717	600	7.45	.0112	.00224	.0496	.00275	18.0
	256	16	83	.675	1.265	202		.57	4.4	816	699	8.7	.0095	.00217	.0480	.00268	17.9
40 mph	64	8	222	.956	1.00	64	over	.615	18.2	204	63	.364	.3049	.0168	.3974	.01773	22.4
N_o = 1780	81	9	197.5	.9425	1.00	81	1600	.545	15.9	258	117	.676	.1645	.0103	.2570	.01123	22.9
P_d = 208	100	10	178	.926	1.00	100		.5	13.8	319	178	1.03	.1079	.0078	.2004	.00873	23.0
d = 1.20	121	11	162	.908	1.00	121		.47	12.0	386	245	1.42	.0781	.0065	.1706	.00743	23.0
S_d = .0925	144	12	148	.875	1.00	144		.46	9.9	459	318	1.84	.0603	.0061	.1528	.00700	21.8
Q_d = .000925	169	13	137	.850	1.035	163		.51	7.6	539	398	2.30	.0483	.00636	.1408	.00729	19.3
	196	14	127	.825	1.065	184		.54	6.2	625	484	2.80	.0397	.00640	.1322	.00733	18.0
	225	15	118.5	.800	1.10	205		.56	5.2	718	577	3.34	.0332	.00638	.1257	.00731	17.2
50 mph	64	8	278	.968	1.0	64	over	.63	17.6	204	36	.209	.665	.0378	.759	.03855	19.7
N_o = 2225	81	9	247	.963		81	1600	.555	15.4	258	90	.522	.249	.0162	.343	.01695	20.2
P_d = 253	100	10	222.5	.957		100		.505	13.6	319	151	.873	.159	.0117	.253	.01245	20.3
d = 1.47	121	11	202	.945		121		.47	11.9	386	218	1.260	.110	.0092	.204	.00995	20.5
S_d = .094	144	12	185	.933		144		.455	10.1	459	291	1.684	.083	.0082	.177	.00895	19.8
Q_d = .00075	169	13	171	.903		169		.50	6.5	539	371	2.145	.065	.0078	.159	.00855	18.6
	225	15	148	.870	1.01	223		.51	5.3	625	457	2.643	.053	.0082	.147	.00895	16.4
										718	550	3.178	.044	.0083	.138	.00905	15.2

$$** \quad MPG = \frac{2390\ E_c}{P \times SFC} = \frac{748\ N_o/N_i}{T_i \times SFC} = \frac{748 \times (4)}{(6) \times (8)}$$

TABLE V

SURGE TECHNIQUE DOWNHILL: T_o = 121 LB-FT

Col. No.	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Item	Speed	Accel. Economy	Gross Thrust	Net Thrust	Acceleration	Accel. Distance	Accel. Fuel	Retarding Force	Deceleration	Decel. Distance	Deceleration Fuel	Total Distance	Total Fuel	Fuel Economy
Symbol	V	MPG	P	P_n	a	S_a	Q_a	P_d	d	S_d	Q_d	S	Q	MPG
Source	-	Table III	3.18 x T	$P-P_f+P_g$	.00578 x (4)	*	(6)÷(2)	$P_f - W \sin\alpha$	.00578 x (8)	*	.4 x (10)÷(1)	(6) + (10)	(7) + (11)	(12)÷(13)
2% Down-Grade	30	11.0	386	344	1.99	.0419	.00381	95	.549	.152	.00202	.194	.00583	33.2
	40	12.0	386	320	1.85	.0600	.0050	135	.781	.142	.00142	.202	.00642	31.5
	50	11.9	386	293	1.695	.0820	.00689	175	1.01	.1375	.0011	.2195	.00799	27.5
4% Down-Grade	30	11.0	386	429	2.48	.0336	.00305	20	.116	.718	.00957	.7516	.01262	59.5
	40	12.0	386	395	2.28	.0487	.00406	60	.347	.320	.00320	.3687	.00726	50.7
	50	11.9	386	368	2.13	.0652	.00548	100	.578	.240	.00192	.3052	.00740	41.2

* See Miscellaneous Computations

APPENDIX D

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