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GEORGE C. MARSHALL SPACE FLIGHT CENTER

SYSTEM ANALYSIS OF A PISTON STEAM ENGINE EMPLOYING THE UNIFLOW PRINCIPLE, A STUDY IN OPTIMIZED PERFORMANCE

By Jerry A. Peoples

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16. ABSTRACT

Results are reported which were obtained from a math model of a generalized piston steam engine configuration employing the uniflow principal. The model accounted for the effects of clearance volume, compression work, and release volume. In spite of the thousands of possible combinations of particular values for ten basic parameters, a very simple solution is presented which characterizes optimum performance of the steam engine, based on miles per gallon. Development of the math model is presented. Also, the relationship between efficiency and miles per gallon is developed. Most important, an approach to steam car analysis and design is presented which has purpose rather than lucky hopefulness. Finally, a practical engine design is proposed which correlates to the definition of the type engine used in this study. This engine integrates several system components into the engine structure. All conclusions relate only to the classical Rankine Cycle.

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DEFINITION OF SYMBOLS

A	Vehicle frontal area (ft^2)
BDC	Bottom Dead Center
E	Heat of Combustion (Btu/lb)
F_R	Fuel Flow (gal/hr)
G	R_W (r/R_G) Total Gear Ratio (ft)
ΔH	Enthalpy change as water passes through Steam Generator (Btu/lb)
k	Polytropic Compression Exponent
K_1	Rolling Road Resistance -- 20 pounds per 1000 pound vehicle weight (lb)
K_2	Aerodynamic Constant (for a 4 door sedan $K_2 = 0.00125$)
M_G	Steam generated (lb/min)
M_R	Steam required (lb/min)
N	Engine speed (rpm)
NPSH	Net Positive Suction Head
P	Supply Pressure (psia)
P_A	Exhaust Pressure (psia)
P_C	Compression Pressure (psia)
r	Differential drive pinion radius (in.)
R_G	Differential Drive Gear Radius (in.)
R_W	Wheel (tire radius) (ft)
S_G	= D_G/X_M, Guffin Supple Number

DEFINITION OF SYMBOLS (Concluded)

T	Supply Temperature ($^\circ$R)
TDC	Top Dead Center
T_E	Average engine torque (ft-lb)
T_W	Average wheel axle torque (ft-lb)
V_C	Clearance volume (in.^{3})
V_D	Volume at BDC (in.^{3})
V_M	Vehicle limiting speed (mph)
V_0	Volume at cutoff (in.^{3})
V_T	Volume at release (in.^{3})
γ	Polytropic Expansion Exponent
η_B	Steam Generator Efficiency
η_S	System Efficiency
ρ	Fuel density (lb/gal)
v_0	Specific volume at supply pressure and temperature (ft^3/lb)

$$X = \frac{V_0 - V_C}{V_D - V_C} \qquad \text{Geometric Cutoff (decimal, see Appendix B)}$$

$$X_M = \left[\frac{V_0}{V_T} - \frac{144}{85.8} \frac{P_A v_0}{T} \left(\frac{V_C}{V_T} \right)^{1-k} \right] \qquad \begin{array}{l} \text{Mass Ratio Cutoff (decimal, see} \\ \text{Appendix A)} \end{array}$$

$$D_G = \left\{ \frac{\dfrac{V_0}{V_T} - \left(\dfrac{V_0}{V_T} \right)^{\gamma}}{\gamma - 1} + \frac{V_0}{V_T} - \frac{V_C}{V_T} - \frac{1}{k-1} \left(\frac{P_A}{P} \right) \cdot \left[\left(\frac{V_C}{V_T} \right)^{1-k} - 1 \right] \right\}$$

Guffin Dour Number (decimal, see Appendix B)

SYSTEM ANALYSIS OF A PISTON STEAM ENGINE EMPLOYING THE UNIFLOW PRINCIPLE, A STUDY IN OPTIMIZED PERFORMANCE

SUMMARY

It is possible to generalize upon performance of a piston steam engine device using the uniflow principal. Performance can be depicted by a simple solution characterized by three charts.

      1. Miles per gallon, with pressure as an independent variable and exhaust pressure and cutoff as arguments.

      2. Miles per gallon, with cutoff as an independent variable and temperature as an argument.

      3. Miles per gallon, with clearance as an independent variable and temperature as an argument.

These charts illustrate pressure and temperature sensitivity and the cutoff value which results in optimum performance. Specific operating conditions should be subject to specialized scrutiny, since a wide range of all variables have been considered in making generalizations.

      Caution should be exercised in gross extrapolation of efficiency to miles per gallon. No sensible correlation exists between theoretical thermal efficiency and miles per gallon for the classic Rankine Cycle. Efficiency measured in the laboratory can be meaningfully related to miles per gallon.

I. INTRODUCTION

      The basis for this study has evolved from the author's work in solar energy conversion. The author has been involved with dynamic energy conversion derived from a reciprocating engine, operating on the Rankine Cycle. In the application of a solar power steam generator, it is most important to optimize the engine/collector system.

In this application, ideal performance, as reported in Reference 1, is pre-empted by the need to know the criteria for optimizing a realizable configuration. This criteria has been developed and applied to the steam automobile.

Over the past several years there has been an attempt to develop an alternate power plant for automotive application. Of all the primary candidates, the Rankine Cycle exhibits the lowest theoretical thermal efficiency. Opponents to the Rankine Cycle have used this fact to mean poor miles per gallon. Historically, efficiency has been the basis for judging engines; however, as computed by the Mollier Chart or as measured in the laboratory, this parameter is lacking in sufficient information to be related to miles per gallon. To compute miles per gallon, the system approach must be used. The total vehicle system must be included (mass, gear ratio and engine parameters).

Traditionally, thermodynamic engineering has indicated the belief that higher theoretical thermal efficiency automatically means higher miles per gallon. However, the relationship between theoretical thermal efficiency and miles per gallon leads to meaningless results.

Efficiency, as measured in the laboratory, can be accurately and meaningfully related to miles per gallon, if properly manipulated. However, depending upon vehicle mass, gear ratio, and speed, extremely wide variations in miles per gallon can be realized. It is therefore, improper to degrade the Rankine cycle on the basis of its theoretical thermal efficiency. The internal combustion engine is a Pyrrhic victory.

II. BACKGROUND INFORMATION

This report is an extension of techniques developed in a previous publication, Analytical Description of the Modern Steam Automobile [1]. The effects of clearance volume, compression pressure, and release volume represent additional considerations. The basic objective is to express how these design realities affect miles per gallon. All data presented in the previous publication [1] were based upon the ideal P-V diagram. This is a Third Order Analysis [1], and results are expected to be much more representative of the behavior of a piston steam engine.

This analysis does not include drive train losses or engine friction losses. Since these losses tend to vary over such a wide range, it is difficult to generalize until some choice is made concerning the basic size and design

2

philosophy. The results are therefore optimistic, but pure (free from gross percentages which cannot be applied to all designs). Nevertheless, the analysis approach is clear and other factors can be incorporate _nce basic design decisions are made.

A typical vehicle will be defined. This vehicle is characterized by weight and frontal area. Routine expressions for rolling road resistance and air dynamic drag are utilized. The engine is rigidly defined to distinguish it from the many variations which exist in considering this type problem.

This analysis has provoked some new terms, which will be introduced as appropriate. There are no less than twelve variables to be considered in the analysis. There are nine other parameters which will be held constant, such as fuel density. In order to minimize the number of possible combinations, normalization will be utilized as much as possible. Even with the simplification introduced by normalization, no less than 10 000 computed values of milles per gallon were made. Each represents a specific combination of the twelve variables. These combinations considered temperatures from 700 to 1700°F and pressures from 200 to 2300 psia. Exhaust conditions varied between 5 and 15 psia. The range of other variable combinations will be presented as their need occurs. The primary objective was to establish the relationship between sets of variables which optimize miles per gallon. This objective has been achieved.

A secondary objective was to provide the relationship between miles per gallon and efficiency. There are two basic efficiencies which are in accepted use:

- First is the theoretical thermal efficiency as results from a first order classical analysis using only the Mollier Chart, T-S, or P-H diagram.

- Second is the actual measurement of engine power, divided by the equivalent heat power input.

In the first case, it is possible to have an increase in theoretical thermal efficiency with a decrease in miles per gallon. In the second case, results can be related to miles per gallon if the proper mathematical multiplication is accomplished. Every increase in measured efficiency percentage does not necessarily mean an equal increase in miles per gallon percentage (one-to-one concept).

The author will admit these words will fall harshly on the ears of many experimenters, however, we best yield to the thoughts of Dr. Richard Feynman, Nobel Prize-winning physicist, "The truth is more remarkable." Indeed this fact will be evident.

III. ENGINE DEFINITION

The thermodynamic model developed here was based on a strict definition of a piston engine using the uniflow principle. This definition is necessary so that results cannot be legimately applied to some variation in design, such as engines with compression release valves. For purposes of discussion and to separate this analysis from those achieved for the ideal P-V diagram, this engine model will be characterized by the term "Guffin Engine."

The Guffin Engine is illustrated in Figure 1. The admission mass compresses the trapped residual exhaust steam in the clearance volume, V_C, to pressure, P_C, and admission continues until cutoff occurs at V_0. The steam is allowed to expand polytropically to the release volume, V_T. Admission and release occur instantaneously at a specified displacement. However, the piston continues beyond the release volume to BDC at volume V_D. The condenser suction pressure is P_A. The trapped steam is then compressed polytropically from volume V_T to V_C and results in a compression pressure P_C. There are two major constraints:

(1) Cutoff must be sufficiently large to prevent "looping."

(2) The compression pressure cannot exceed supply pressure, P.

These ideas are further developed in Appendix A. For purposes of presenting wide parameters variation, the concept of a baseline operating condition and fixed geometry were selected. These baseline conditions are:

- Vehicle Weight = 3500 lb

- Supply Pressure = 800 psia

- Supply Temperature = 900° F
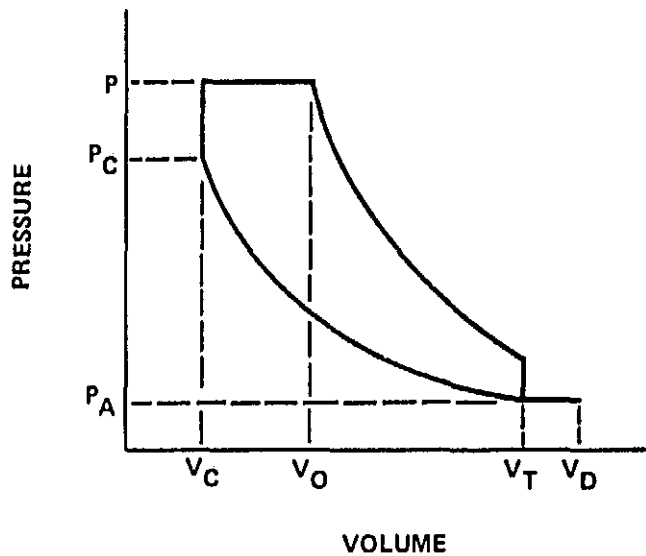
Figure 1. P-V diagram of the Guffin Engine
(specific definition is given in Appendix A).

- Ratio of $V_C/V_T = 0.04$

- Exhaust Pressure = 8 psia

Baseline results will be illustrated, then performance will be shown for derivations from the baseline.

As it turns out, the equation for percent clearance volume does not appear discretely in the derivation. The clearance can be calculated by:

$$\text{Clearance (decimal)} = \frac{\dfrac{V_C}{V_T}}{\dfrac{V_D}{V_T} - \dfrac{V_C}{V_T}} \tag{1}$$

The parameters which will be held constant, unless otherwise specified, are given as follows:

- E = heat of combustion = 19 500 Btu/lb

- A = vehicle frontal area = 27 ft^2

- Vehicle weight = 3500 lbs

- ρ = fuel density = 6.7 lb/gal

- η_B = steam generator efficiency = 85 percent

- γ = polytorpic expansion exponent = 1.25

- k = polytropic compression exponent = 1.15

- Vehicle Speed = 60 mpn

- Ratio of V_D/V_T = 1.05

There is one outstanding feature of an engine having clearance which cannot be associated with the ideal P-V diagram. A clearance volume introduces the possibility for zero cutoff. This fact alone is the distinguishing difference between ideal engine behavior and real engines. The idea of zero cutoff is not new, as the principal of the Bash Valve is basically a zero cutoff engine. These type valving systems have been built and demonstrated. Also, a 100 percent cutoff is not possible. The term cutoff, which will be used herein, is based on geometry of the piston displacement,

$$X \text{ (Geometric cutoff, decimal)} = \frac{V_0 - V_C}{V_D - V_C} = \frac{\dfrac{V_0}{V_T} - \dfrac{V_C}{V_T}}{\dfrac{V_D}{V_T} - \dfrac{V_C}{V_T}} \quad , \quad (2)$$

and maximum cutoff occurs when $V_0 - V_C = V_T - V_C$. Therefore, maximum cutoff is:

$$X \text{ (Maximum for baseline)} = \frac{1 - \dfrac{V_C}{V_T}}{\dfrac{V_D}{V_T} - \dfrac{V_C}{V_T}} = \frac{1 - 0.04}{1.05 - 0.04} = 95 \text{ percent} \quad . \quad (3)$$

The difference between ideal engine analysis and the Guffin Engine can be summarized as:

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- Miles per gallon of the Guffin Engine will always be less than the ideal engine for the same operating conditions.

- The Guffin Engine can be optimized for mpg, whereas the author knows of no optimizing parameter (s) for the ideal engine.

- For the ideal P-V diagram, mpg continues to increase as cutoff decreases and this does not occur in the Guffin Engine.

IV. FUNDAMENTAL BEHAVIOR OF THE GUFFIN ENGINE

At the outset of this task, the author outlined a gigantic matric involving all of the many variations. As already mentioned, over 10 000 combination values of miles per gallon would need to be computed. The matrix was managed through Fortran Programming on a digital computer. This methodical procedure would allow a systematic approach analysis in a search for finding the optimizing parameter, if such existed. The equations which were programmed are given in Appendix A and B. All 10 000 answers were computed and printed within a few short minutes.

At first the author was under the impression that a large volume would be required to bind all Guffin Engine characteristics and properly represent performance variations. And indeed, if it did, the data was available. Then a surprise happened. Within a few minute of charting first choice parameters, it became obvious that there were only one parameter which would optimize the Guffin Engine. And most important to the designer, the specific value of the optimizing parameter was practically independent of all other parameters.

The optimizing parameter is the geometric cutoff defined in Appendix A as:

$$X = \frac{\dfrac{V_0}{V_T} - \dfrac{V_C}{V_T}}{\dfrac{V_D}{V_T} - \dfrac{V_C}{V_T}} \quad . \tag{4}$$

Maximum miles per gallon, for any combination of the variables considered will occur at about, $x = 0.07$. It was noted that each set of combinations produced

a different maximum but the maximum always occurred at about $x = 0.07$. For purposes of documentation, the variables included in the matrix were:

P = supply pressure (200 to 2300 psia, in increments of 300 psia)

T = supply temperature (700 to 1700°F, in increments of 200°F)

V_C/V_T = 0.02, 0.04, 0.08, 0.16

V_0/V_T = calculated values, based upon X (Appendix A)

V_D/V_T = 1.05 and 1.10

γ = 1.25 or 1.2

k = 1.15 or 1.25

P_A = Condenser pressure of 5, 8, and 15 psia

A = frontal area = 27, 30, and 31 ft^2

weight = 3500, 4500, and 5500 lbs

V_M = maximum vehicle speeds of 40, 50, and 60 mph

Three combinations of γ and k were used:

(1) γ = 1.25

k = 1.15

(2) γ = 1.25

k = 1.25

(3) γ = 1.20

k = 1.15

All of the possible combinations illustrated above are not valid because some are outside the constraints imposed by Appendix A. However, for valid combinations, maximum miles per gallon is always achieved near $x = 0.07$.

Analysis of the Guffin Engine can, therefore, be represented by a very simple presentation. Only three charts are required, and the simplicity involved is one of those delightful situations where an apparently very difficult data reduction analysis can be reduced to an extremely simple solution.

The purpose of this section is to present this solution. Other sections will expand the data to show how this solution is completed. A physical explanation of behavior will not be attempted in this section, however, other sections in this report will develop the physics of the results. Only mathematical results are given below.

The first chart is presented in Figure 2. This chart illustrates the sensitivity of miles per gallon to supply pressure. Four curves are shown, two for a relatively low cutoff of 5 percent and two for an intermediate cutoff of 20 percent. Associated with each cutoff were the extreme exhaust conditions established for this analysis. As indicated, they were 5 and 15 psia. These two pressures represent the two extreme exhaust pressures considered. There are two extraordinary facts about this chart. First, there is nothing unique about supply pressure. In fact, the higher pressure yields lower miles per gallon. The second unique fact is the added advantage of the higher exhaust pressure, which occurs at the higher pressures. This is contrary to classical analysis on the T-S diagram. However, on the basis of this chart, the generalization can be made that miles per gallon is a weak function of supply pressure and exhaust conditions. Performance must, therefore, be governed by cutoff, temperature, and maybe clearance volume.

In order to study the influence of cutoff and temperature, Figure 3 is presented. This figure was charted on the basis that $P_A = 8$ psia and $P = 800$ psia, which were the baseline conditions. However, as shown in Figure 2, the results are indicated for all pressures and all exhausts conditions. The unique characteristics about this chart is the optimizing characteristics which occur at about 7 percent cutoff. It may be noted that the 7 percent value is not a rigid calculated peak, but results from a visual inspection of the chart.

As already stated, this peak always occurs near 7 percent cutoff, regardless of the combination of variables required to describe the Guffin Engine. This fact alone is the single characteristic which collapsed the usefulness of the extensive matrix developed by the author.

Finally, Figure 4 illustrates the influence of clearance volume at the optimized cutoff. It is most desirable to have the least possible clearance without exceeding the constraints on the Guffin Engine. Each of the curves in this section are charted for a fixed speed of 60 mph. In accordance with

TEMPERATURE 900°F

$$\frac{V_c}{V_T} = 0.04$$

MILES PER GALLON @ 60 MPH

CUTOFF 5%

CUTOFF 20%

$P_A = 5$ PSIA

$P_A = 15$ PSIA

24

22

20

18

16

14

200   600   1000   1400   1800   2200

SUPPLY PRESSURE (PSIA)
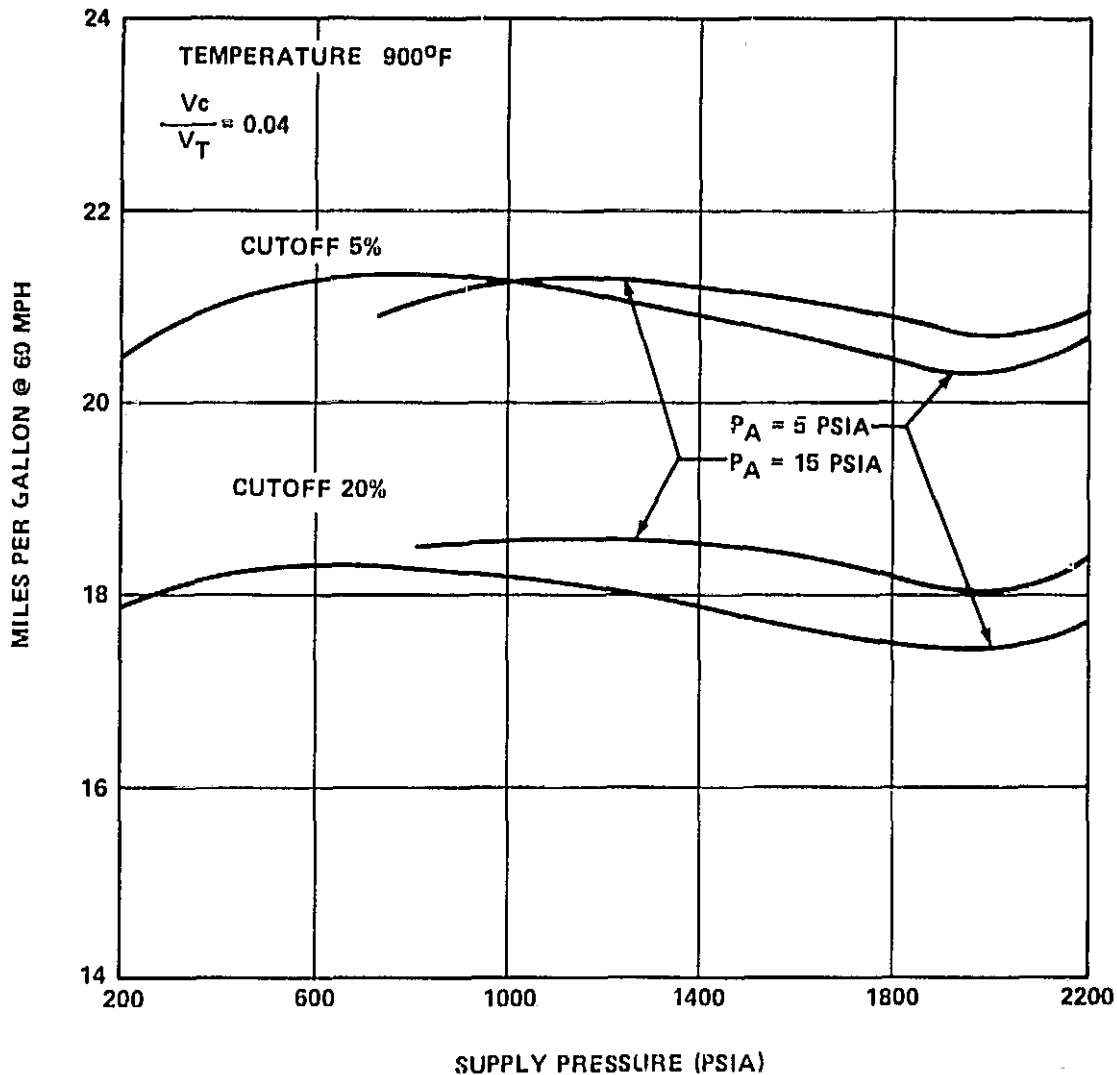
Figure 2.   Sensitivity of the Guffin Engine to supply pressure
(fundamentally, pressure increases theoretical thermal
efficiency but has little practical influence upon mph. A
pressure of 800 psia is a good representative of
performance over a wide pressure range).

Appendix B, particular values of the Abatement Number and the Guffin Dour
Number are necessary to achieve 60 mph.   Since the values of these two num-
bers are most important at the optimizing cutoff, these values are given below.

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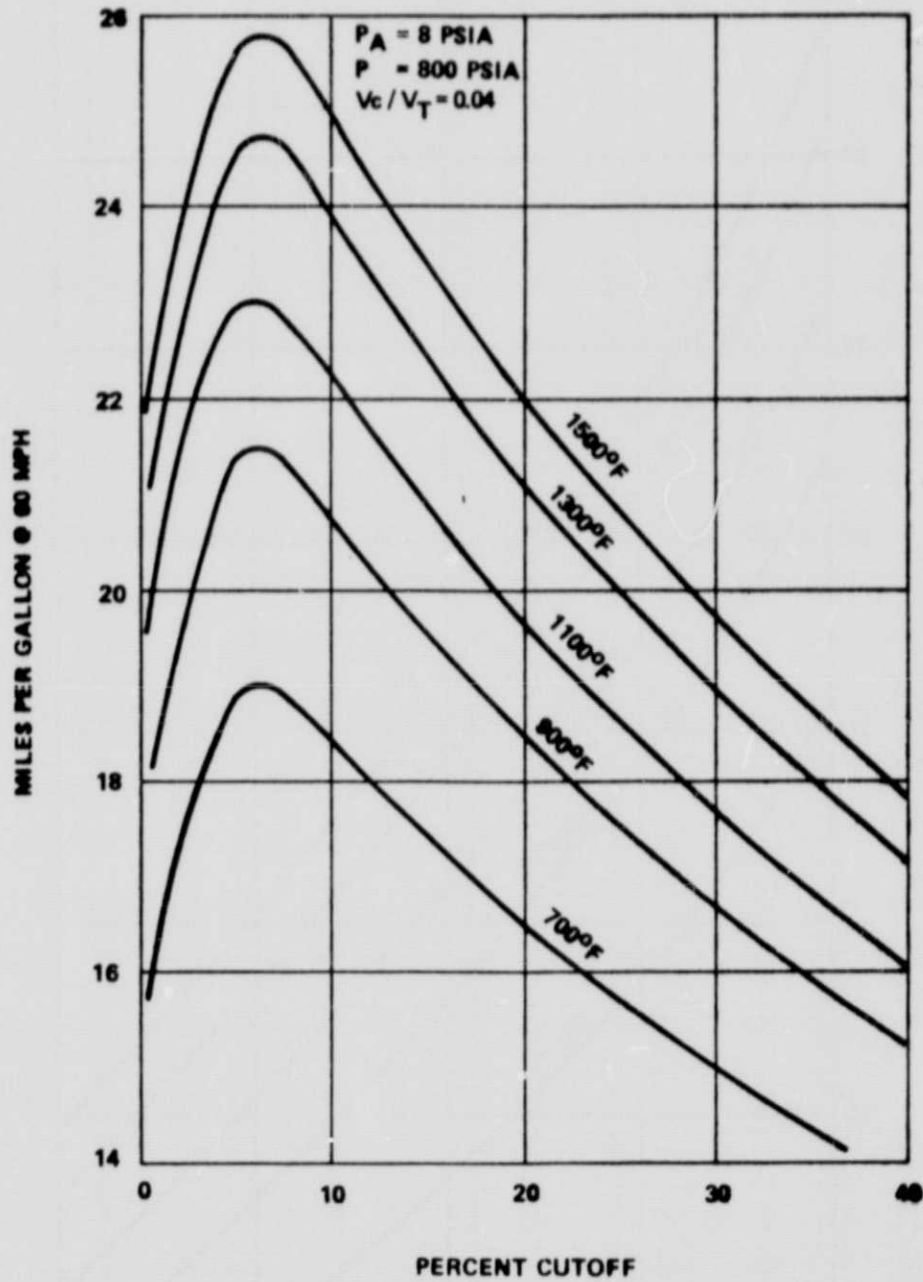
Figure 3. The optimizing nature of cutoff (optimum
performance always occurs within a narrow cutoff
band between about 5 and 7 percent).

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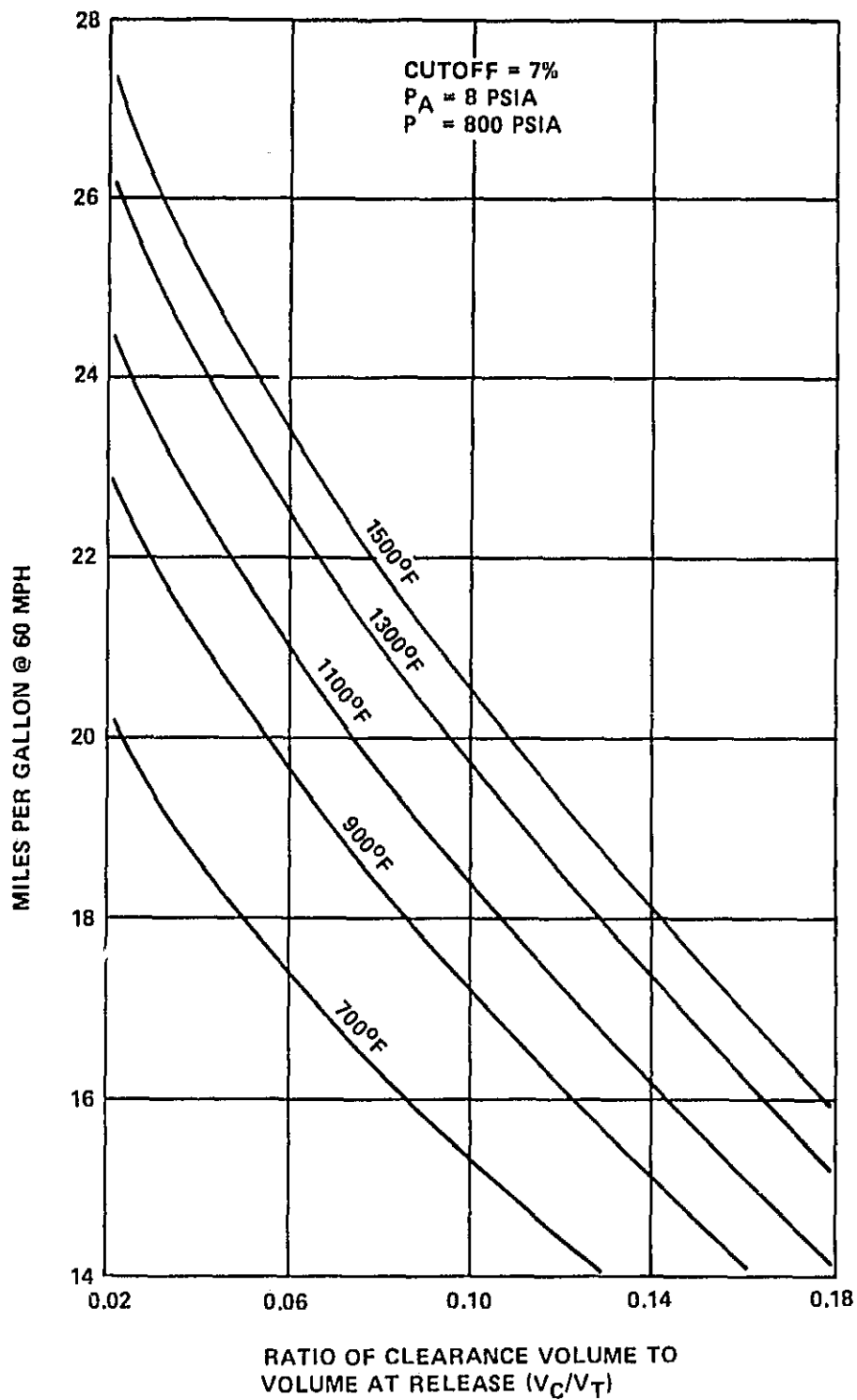
Figure 4. Sensitivity of optimum performance to clearance volume.

Pressure (psia)	Guffin Abatement Number (lb-in./ft)	Guffin Dour Number (Dimensionless)
500	75 209	0.192
800	66 593	0.217
1100	63 279	0.228
1400	61 556	0.234
1700	60 480	0.239
2000	59 748	0.242
2300	59 219	0.244

It should be noted that the term "Guffin" preceding the abatement number and the dour number is used to distinguish these values from those obtained with the ideal P-V diagram, as discussed in Reference 1. Also, expressions for these terms are given in the list of symbols and the Appendices. In the event the reader is interested in making some sample calculations, Appendix C has been added. This appendix has values of the Guffin Dour number and mass ratio cutoff for some specific combination. Be cautious and avoid confusing the geometric cutoff with the mass ratio cutoff. Appendix A gives expressions for both:

- Mass ratio cutoff — ratio between admission mass and the mass required to fill the entire release volume at supply temperature and pressure

- Geometric cutoff — ratio of piston displacement during admission to the total piston displacement. This is the optimizing cutoff value.

Also, a table in Appendix C gives the specific volume and enthalphy at the supply conditions. These values are helpful in computing the Booty Number. In computing the Booty Number, the value of enthalphy at the saturated liquid exhaust pressure must be subtracted from the table value.

The term Mass Ratio Cutoff, X_M, will be demonstrated later for its usefulness and importance in assessing the reasons for these mathematical results.

V. DETAILED DESCRIPTION OF THE GUFFIN ENGINE ANALYSIS

The discussion in Section IV generalized upon the behavior of the Guffin engine. This approach gave an overview of what may be expected from an engine which resembles the Guffin Engine definition. The purpose of this section is to expand the data and bring out some peculiarities. Five subjects will be discussed:

(1) the effects of pressure at very low and very high temperatures,

(2) detailed analysis of the optimum cutoff,

(3) effects of temperature on mpg at the optimum cutoff for different clearance volumes,

(4) effect of compression and expansion polytropic exponent, and

(5) effect of vehicle mass and speed on mpg.

Figure 5 illustrates the effect of pressure at 700° F. The same results are evident as those observed in Section IV at 900° F. The interesting thing about low temperature operation is the increased degradation of performance with pressure. This fact reinforces previous arguments that pressure above 600 or 800 psia does little or nothing to aid miles per gallon. Again, best performance seems to favor the higher exhaust pressure. However, it may be noticed that performance is less for all pressure from that achieved at 900° F.

Now compare the performance at 700 and 900° F with that at 1500° F (Fig. 6). Again, the same trend prevails. Performance does not deteriorate after 800 psia, but the increased pressure seems to offer nothing, even at the higher temperature. As expected, overall performance is better than that achieved at the lower temperatures.

The logic in selecting the baseline operating conditions at 800 psia is understandable. A pressure of 800 psia results in best or equal performance to that obtained for all other pressures, and also is within the constraints of Appendix A. For any temperature and exhaust conditions, the pressure need not be greater than 800 psia. This is the wisdom of the "new steam." If the exhaust conditions are low, this pressure can be reduced to 600 psia.

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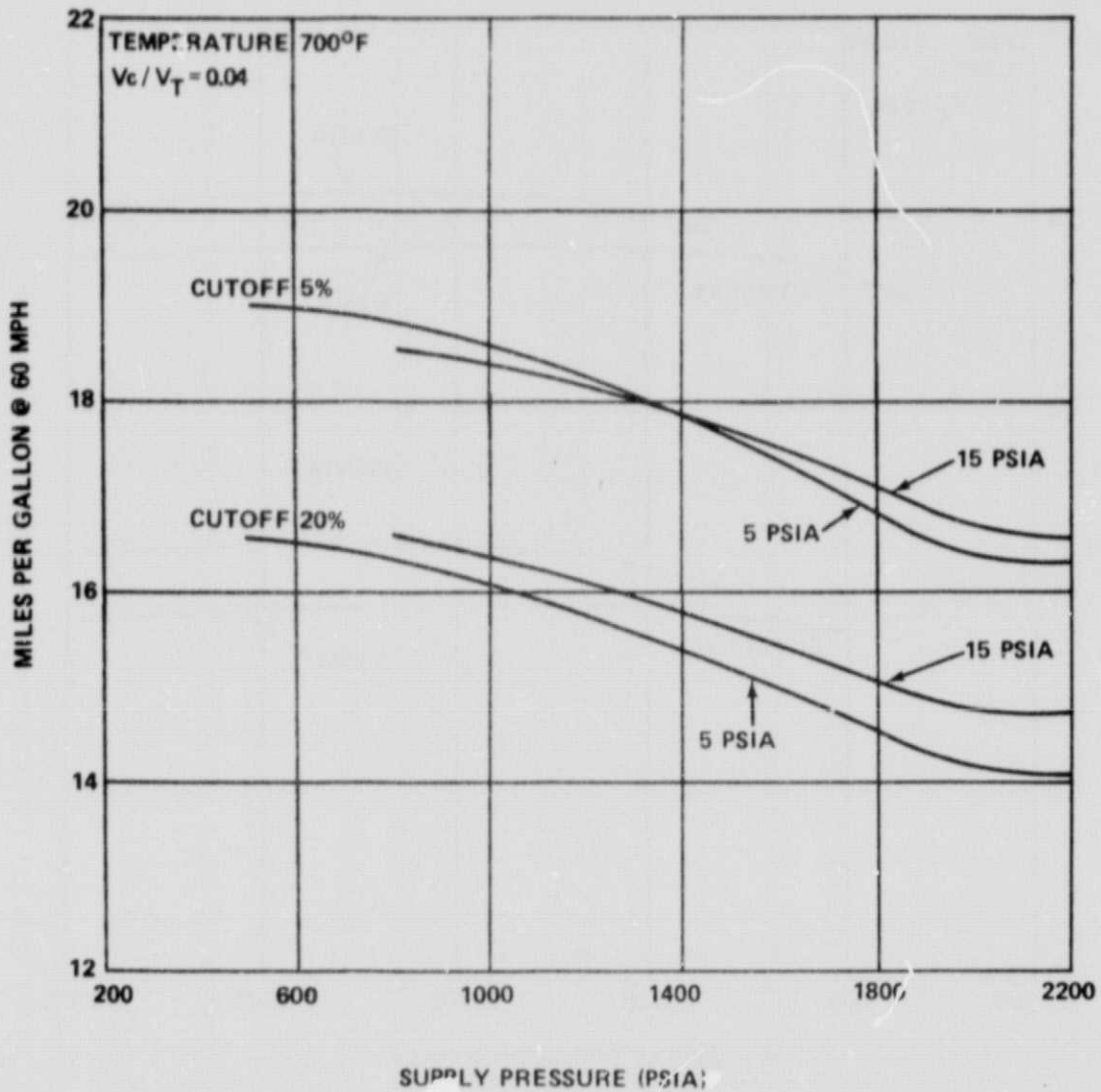
Figure 5. Sensitivity of performance to pressure for low temperature (note that increased pressure causes substantial degradation in miles per gallon. A pressure of 800 psia represents a region of superior performance).

There is one exception to this rule. This situation occurs at zero cutoff, as achieved with the Bash Valve. Figure 7 represents what can be expected from the Bash Valve. In this application, the supply pressure should be no less than 1000 psia. However, as illustrated in Figure 3 of Section IV, Bash Valve performance is inferior. For example, at 900° F, 1000 psia, and 8 psia
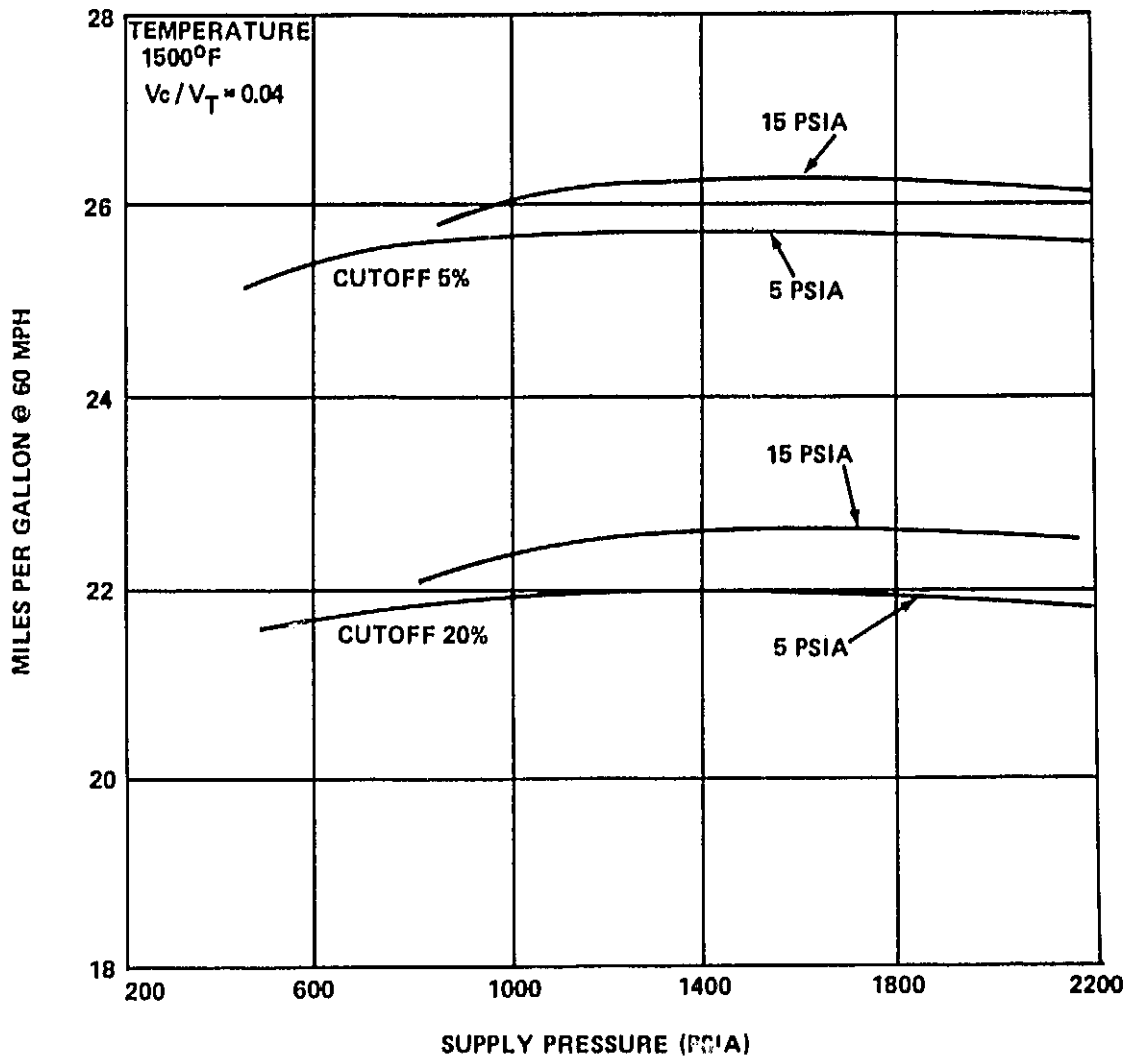
Figure 6. Sensitivity of performance to pressure for high temperature (note that above 800 psia, pressure offers no performance advantage).

exhaust, the Bash Valve approach yields about 18.2 mpg at 60 mph. At 7 percent cutoff, 60 mph at 800 psia, 21 mpg is obtained. This is a 13.3 percent increase, an improvement which cannot be ignored.

In Section IV, the optimizing cutoff was referred to as "about 7 percent." The reason for this is the fact that over the wide range of variables being considered, the optimum cutoff can vary from 5 percent to 7.5 percent. This range

CUTOFF
— ZERO%
$V_c / V_T = 0.04$

$P_A = 8$ PSIA

24

22

20

18

16

14

12

MILES PER GALLON @ 68 MPH

1500°F

800°F

700°F

200   600   1000   1400   1800   2200

SUPPLY PRESSURE (PSIA)
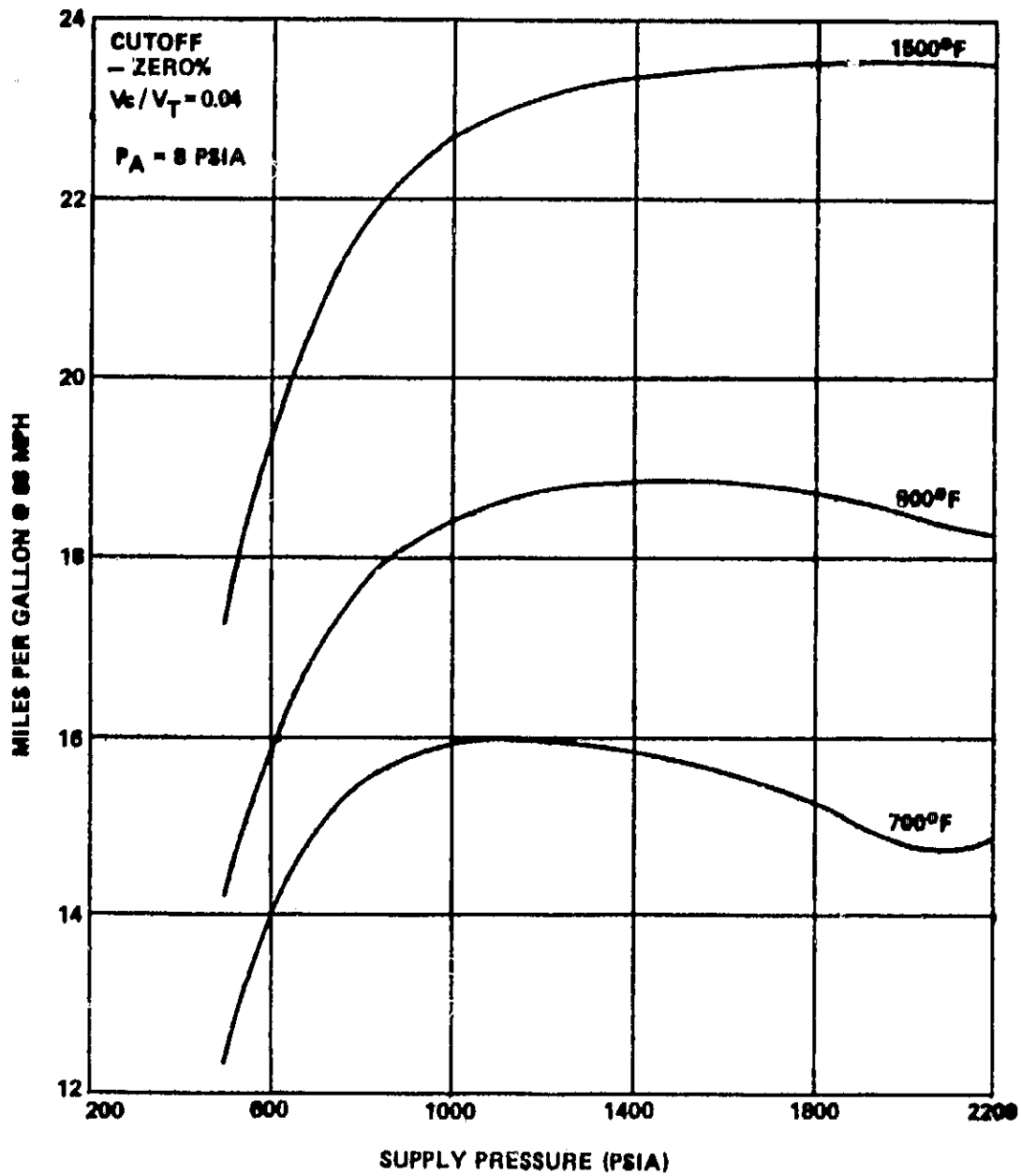
Figure 7. A special case of performance at zero cutoff (in this
situation, pressure should be 1000 psia or greater. However,
as shown in Section IV performance of zero cutoff is inferior).

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is considered to be sufficiently small that the Guffin Engine can be character-
ized by a single cutoff. For the purpose of this report 7 percent was selected.
Figures 8 and 9 illustrate performance between 4.5 percent and 7.5 percent
cutoff.
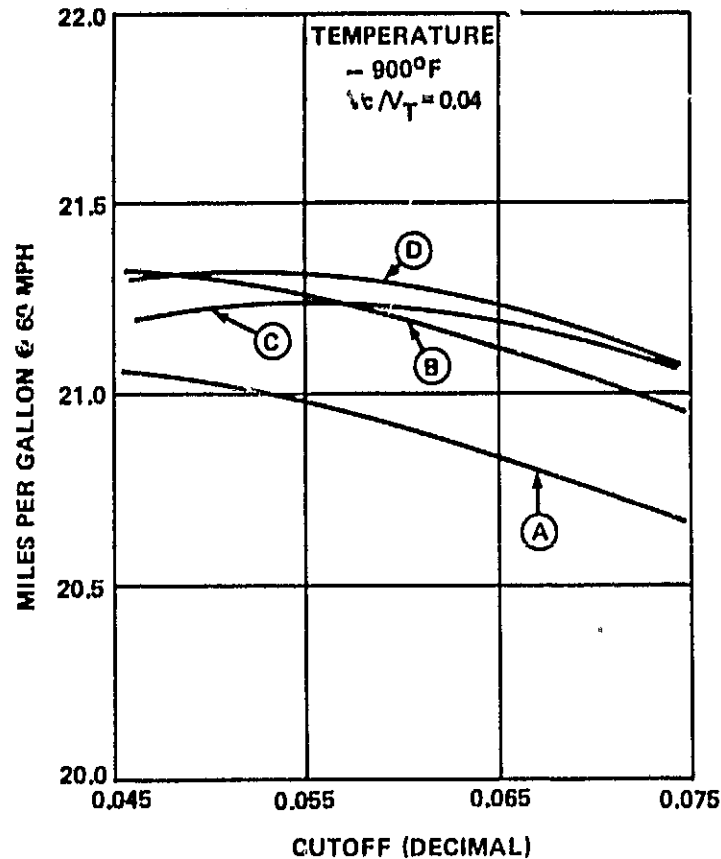
Figure 8. Variation of the optimum cutoff for various
combinations noted in Table 1.

The combination of variables are designated in Tables 1 and 2, respec-
tively. Figure 8 represents what may be characterized as normal operating
conditions. Figure 9 covers end combinations. From these charts, no one
cutoff can be selected to give the maximum value for all ranges of the variables
included. It does seem that 5.5 percent or 6.0 percent may satisfy most con-
ditions. However, the slope of the curve below 4.5 percent drops off steeply,
as shown in Section IV. Past the peak, the curve degrades less. Therefore,
it is best to insure that the design be just a little past the peak rather than
taking an uncertain risk of ending up on the steep side of the curve. For this

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Figure 9. Variation of the optimum cutoff for various combinations noted in Table 2.
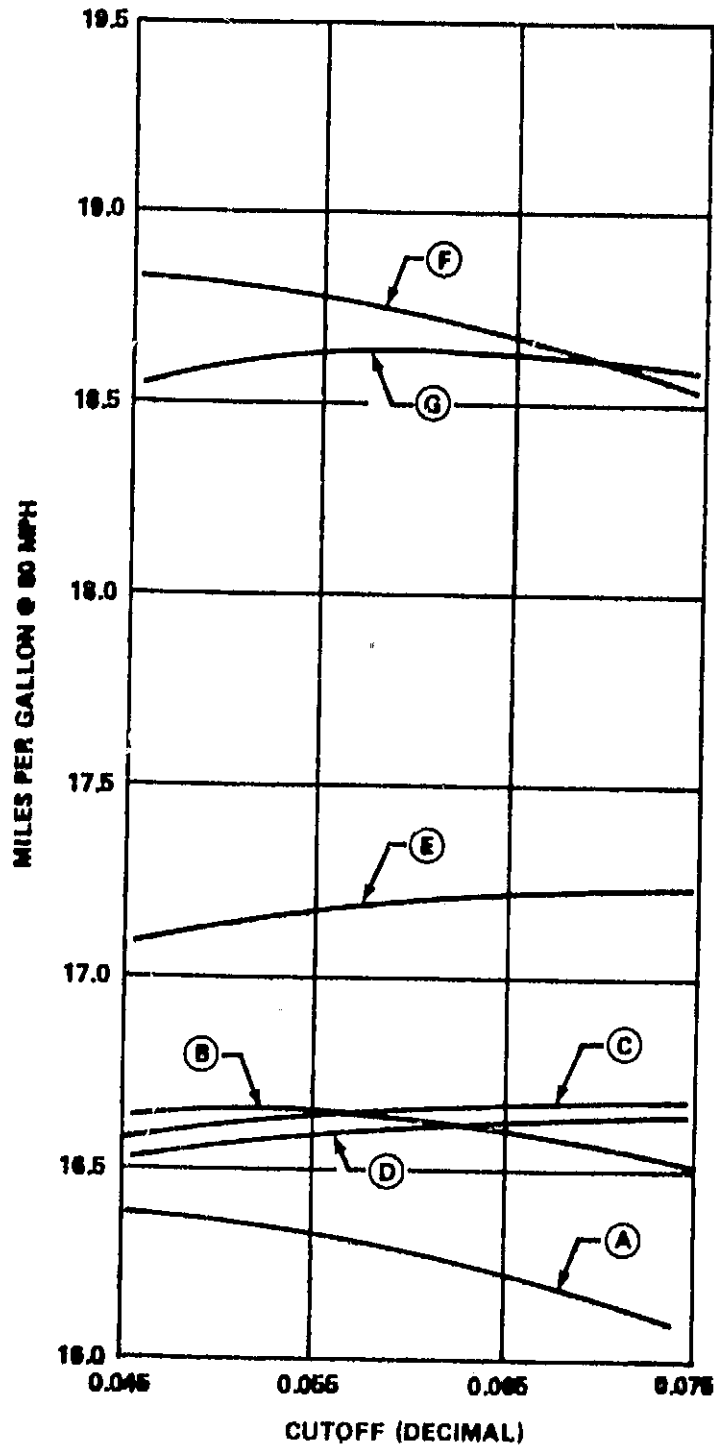
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TABLE 1. COMBINATIONS USED FOR FIGURE 8.

Curve	Pressure (psia)	Temp (°F)	P_A	V_C/V_T
A	1100	900	5	0.04
B	500	900	5	0.04
C	800	900	15	0.04
D	1100	900	15	0.04

TABLE 2. COMBINATIONS USED FOR FIGURE 9.

Curve	Pressure (psia)	Temp (°F)	P_A	V_C/V_T
A	2000	700	5	0.04
B	2000	700	15	0.04
C	800	1500	5	0.16
D	2000	1500	5	0.16
E	2000*	1500	15	0.16
F	800	700	5	0.04
G	800	700	15	0.04

*Also 800 is valid for this combination

reason 7 percent was selected for documentation. Note that these comments are based on graphic results. An exact mathematical optimum could be found by:

$$\frac{dD_G}{dX_M} = 0 \quad (\text{This expression will be discussed in Section VI}) \quad , (5)$$

and solving for the geometric cutoff. The author has attempted this, but failed due to the extreme complexities involved.

The designer has very little control over the expansion and compression polytropic exponent. In the range of values considered, the exponents seem to have little practical effects, except at zero cutoff. Table 3 is a compilation of miles per gallon achieved at different combinations of γ and k. As can be observed, a decrease in γ will increase miles per gallon, but a decrease in k will also increase miles per gallon.

TABLE 3. VARIATION OF MPG WITH DIFFERENT
COMBINATIONS OF POLYTROPIC EXPONENTS FOR
EXPANSION AND COMPRESSION PROCESSES.*

Cutoff (decimal)	Combinations of Polytropic Exponents for Expansion and Compression Processes		
	$\gamma = 1.25$ $k = 1.15$	$\gamma = 1.25$ $k = 1.25$	$\gamma = 1.20$ $k = 1.15$
0	17.86	19.52	20.33
0.05	21.36	22.11	22.50
0.10	20.56	20.95	21.31
0.20	18.42	18.58	18.83
0.40	14.92	14.95	15.06

*The respective mpg for each cutoff is valid for 900°F, 800 psia, P_A = 8 psia and for a vehicle speed of 60 mph.

Figure 10 illustrates the effect of vehicle speed and weight upon performance. This speed variation upon mpg is a clue to the variation in system efficiency, which will be discussed in Section VII.
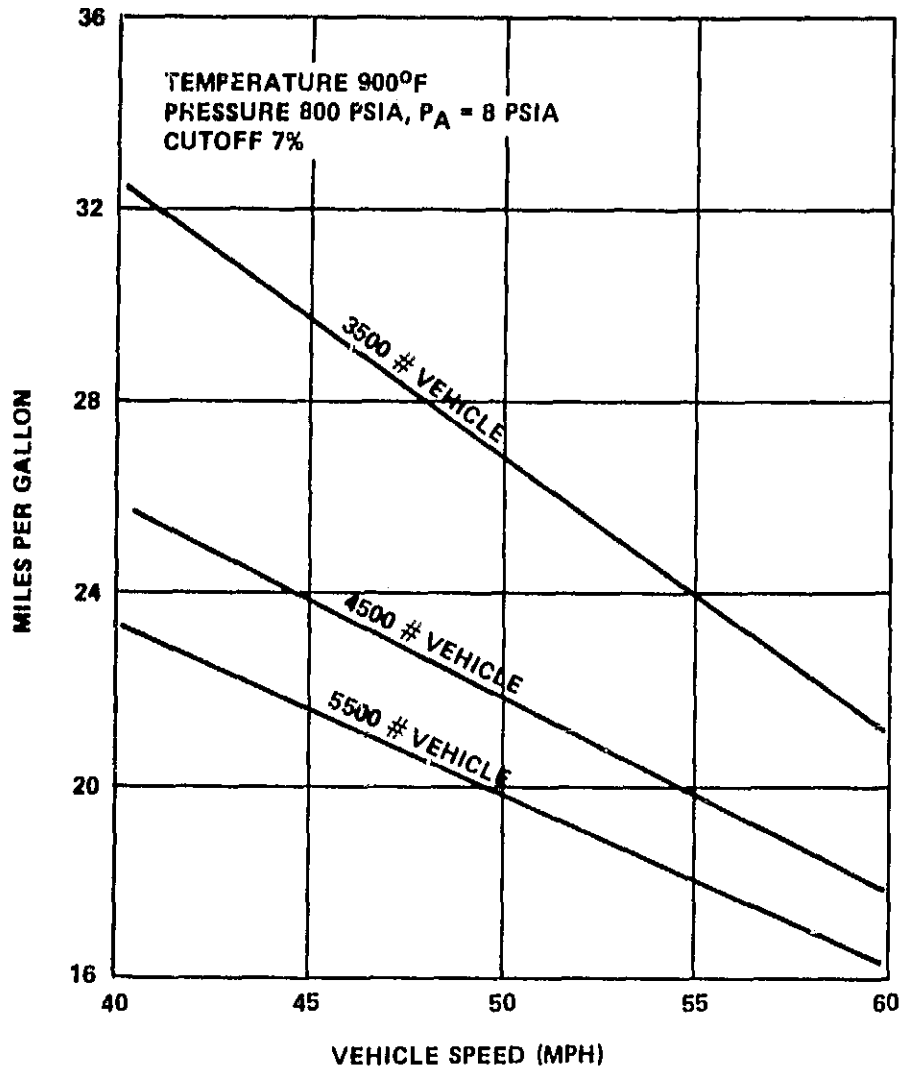
Figure 10. Relationship between performance and vehicle speed.

VI. PHYSICS OF THE GUFFIN ENGINE

Now that the basic performance characteristics of the Guffin Engine have been presented, a question of "why" naturally arises. The discussions in this section are not intended to represent an exhaustive analysis of the physics involved in the engine behavior.

There appears to be two major questions which require explanations. The purpose of this section is to answer these two questions:

(1) In general, why does a higher exhaust pressure result in improved performance? This characteristic is most pronounced for the higher pressures and moderate cutoff.

(2) Why does the optimizing value of cutoff always occur within a small range (0.05 to 0.07) for a wide variation of engine geometry and operating conditions?

To begin with, the expression for miles per gallon in Appendix A will be combined with the limiting speed equation developed in Appendix B. The idea is to substitute the Guffin Abatement Number, PV_T/G, of equation (B-7) in equation (A-17). Solving for the Guffin Abatement Number.

$$\frac{PV_T}{G} = \frac{24\,\pi K_2 A}{D_G}\left(V_M^2 + \frac{K_1}{K_2 A}\right) \quad .$$ (6)

Substituting equation (6) into (A-17) gives

$$mph = 0.0142\,\eta_B E\rho\,\frac{Z_0 RT}{\Delta H}\,\frac{1}{\left(V_M^2 + \frac{K_1}{K_2 A}\right)24\,\pi K_2 A}\,\frac{D_G}{X_M} \quad .$$ (7)

Where, D_G is the Guffin Dour Number,

$$D_G = \left\{\frac{\frac{V_0}{V_T} - \left(\frac{V_C}{V_T}\right)^\gamma}{\gamma - 1} + \frac{V_0}{V_T} - \frac{V_C}{V_T} - \frac{1}{k-1}\left(\frac{P_A}{P}\right)\left[\left(\frac{V_C}{V_T}\right)^{1-k} - 1\right]\right\} \quad ,$$ (8)

and X_M is the Mass Ratio Cutoff,

$$X_M = \left[\frac{V_0}{V_T} - \frac{144}{85.8} \frac{P_A v_0}{T} \left(\frac{V_C}{V_T} \right)^{1-k} \right] . \qquad (9)$$

Both of the expressions are derived from Appendix A and B, respectively. By studying equation (7), there are three variables which must account for the characteristics already presented. These are the Booty Number,

$$\frac{Z_0 RT}{\Delta H} , \qquad (10)$$

the Guffin Dour Number, D_G, and the Mass ratio cutoff, X_M. More exactly, the ratio of D_G/X_M is crucial.

At this point it is worth nothing that D_G/X_M is analogous to the supple number developed from the ideal P-V diagram. Therefore, the ratio D_G/X_M will be referred to as the Guffin Supple Number. Also, note that the Booty Number is a characteristic of the working medium only, whereas, D_G and X_M involves combination of the supply conditions and engine geometry.

First consider the affect of the exhaust pressure upon the Booty Number. At saturated liquid conditions, with $P_A = 5$ psia and $h = 130.13$ Btu/lb, or $P_A = 15$ psia and $h = 181.11$ Btu/lg; then, the value of the Booty Number will be greater at $P_A = 15$ psia than at $P_A = 5$ psia, since ΔH will be smaller at $P_A = 15$ psia.

This statement applies for a given supply pressure and temperature. Thus, regardless of the value of the Guffin Supple Number, the greater exhaust pressure will tend to increase mpg. At first, the small difference in ΔH may appear insignificant. This insignificance becomes important when the Supple Number, as a function of exhaust pressure, is investigated.

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As the variation of D_G and X_M is considered with changes in the geometric cutoff, X, two counter situations develop:

(1) For low cutoff, the compression work will comprise a large portion of the net work. As cutoff increases, the percentage of compression work will decrease.

(2) At low cutoff, the required mass flow rate is relatively small. As cutoff increases, the mass flow rate will increase.

It may be noted that D_G and X_M are a direct measurement of the net work and the required mass flow rate, respectively. It must be considered that as the exhaust pressure increases, the characteristic mass increases, thus reducing the required mass flow rate.

The above arguments can be demonstrated by comparing the percent degradation of D_G as the exhaust pressure is increased from 5 psia to 15 psia. At the same time, the percent reduction in the required mass flow rate can be compared as the exhaust pressure is increased from 5 to 15 psia. Although several numbers and calculated data were required to illustrate this, only the final result will be shown. Table 4 illustrates specific results.

TABLE 4. PERCENT REDUCTION IN THE GUFFIN DOUR
NUMBER, D_G, AND MASS RATIO CUTOFF AS THE
EXHAUST PRESSURE IS CHANGED FROM 5 TO 15 PSIA.*

	X – Cutoff (Percent)				
	0	5	10	20	40
D_G	68%	17.0%	10.9%	6.48%	4.22%
X_M	45%	13.33%	8.8%	4.86%	2.57%

*A decrease in D_G is undesirable while a decrease in X_M is very desirable. Thus, there is a tradeoff as cutoff increases. The optimum cutoff is governed by D_G/X_M, the Guffin Supple Number as illustrated in Figure 11.

The following example will illustrate how to intepret Table 4. At zero cutoff, D_G is decreased by 68 percent as a result of compression work change between $P_A = 5$ and 15 psia. By utilizing the characteristic mass in the compression procession, there is a 45 percent reduction in the required mass flow rate as the exhaust pressure changes from 5 to 15 psia. The combined effect can be seen as the percentage change with increasing cutoff. As cutoff increases, the percent degradation of D_G decreases. Also, the saving of the required mass flow rate decreases. The major question is, at what cutoff does the best trade off result?

To answer this question, it can be solved mathematically by: $dD_G/dX_M = 0$. It would be beautiful if this equation could be solved for an explicit value of cutoff. However, such is impossible. However, a graphic solution can be easily obtained by plotting the Guffin Supple Number as a function of cutoff. Figure 11 represents such a chart. Note that the optimized values occur at about 7 percent, as presented in Section IV. The Guffin Supple Number is the factor which drives the basic characteristic. Note that the difference between the two curves is less at cutoff values that are greater than 10 percent. Also, the 5 psia curve is only very slightly greater than the 15 psia line. However, the slightly larger Booty Number at the higher exhaust pressure results in more mpg for cutoff values between 5 and 40 percent.

VII. SYNTHESIS OF EFFICIENCY

In recent years, a hesitant attitude toward engine efficiency has developed. This applies first to theoretical thermal efficiency. As hardware is developed, efficiency begins to imply laboratory type data. The idea is that more efficient machines result in more miles per gallon. At first, this appears so basic that any deviation from this would be impossible and in violation of all common sense, if not the hard core laws. As an example of the hysteria over efficiency, consider the following example which is representative of some of the reasoning that appears in the literature:

A designer of Brand-X steam engine announces that his engine is 12.81 percent efficient and gets 22.5 miles per gallon. Usually no other information is given. Now, a designer of Brand-Y steam engine knows that his engine is 32.42 percent efficient. This represents an increase of 153.08 percent. The builder of Brand-Y immediately concludes that this gas mileage should be increased by 153.08 percent. He performs the calculations and becomes convinced that his engine will give 56.43 miles per gallon. The designer of Brand-Y hastens to announce a break-through.
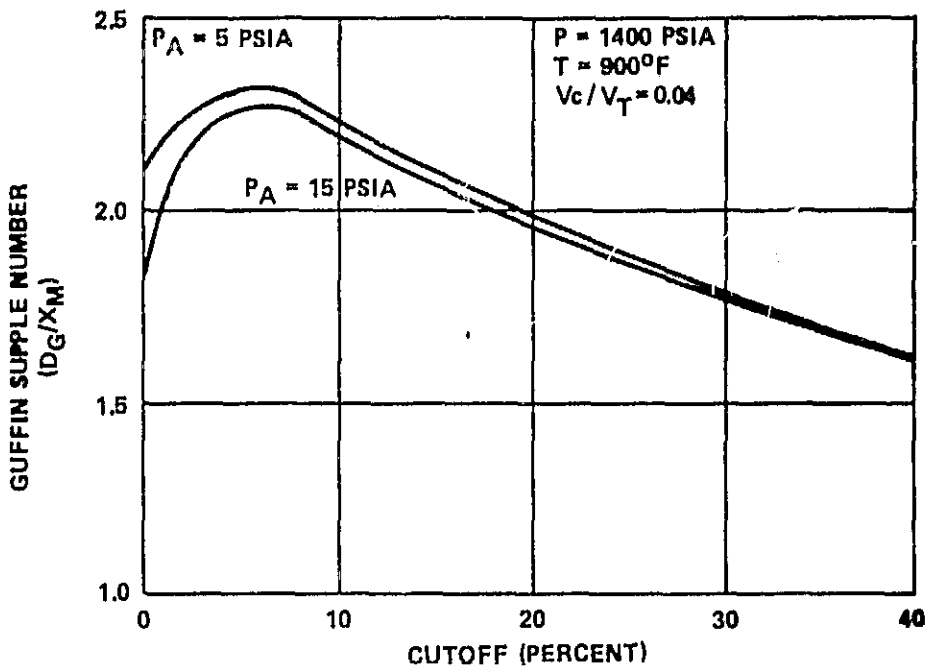
Figure 11. An illustration of how the Guffin Supple Number
drives the position of optimum cutoff (since the Booty
Number is greater at 15 psia than 5 psia, generally
better results are obtained at the higher exhaust pressures).

The purpose of this section is to provide an analytical basis for relating mpg to both theoretical thermal efficiency and laboratory measured efficiency. Also, the type of reasoning represented by the above example is technically inadequately described and the resulting conclusions are folly.

To introduce this discussion, consider the increase in Carnot efficiency as a function of percent increase in source temperature. For a sink temperature of 70° F and a supply temperature of 700° F, the Carnot thermal efficiency is 55.17 percent. The increase in Carnot efficiency above 55.17 percent for a given percent increase in the supply temperature, is illustrated in Figure 12. Note that a 50 percent increase in temperature (1050° F) results in only a 20 percent increase in efficiency above 55.17 percent. In this classical example, there is a clue that the "one-to-one" percentage increase may not exist. However, the primary object is to illustrate, through an equivalent type chart, the relationship between miles per gallon and "efficiency." Before the development of these relationships, consider the effect of temperature change upon miles per gallon. The effect of pressure will not be investigated since it has already been demonstrated that miles per gallon is very insensitive to pressure.
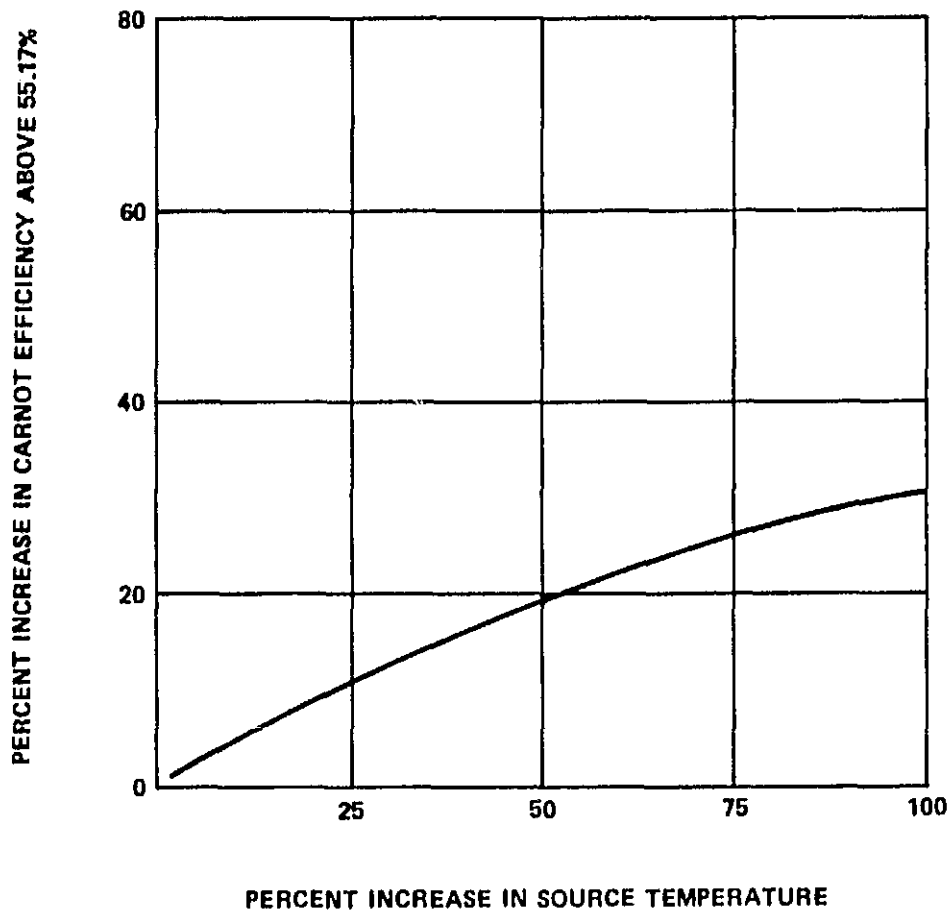
Figure 12. Percent increase in Carnot efficiency with percent
increase in supply temperature above 700° F (large percentage
increase in source temperature results in only mediocre
increase in theoretical thermal efficiency).

For this illustration, the percent increase in mpg above that which
occurs at 700° F and 7 percent cutoff will be selected. The specific points
plotted will be taken for those applicable to 800 psia and with V_C/V_T equal to
0.04. Exhaust pressure is 8 psia.

Under these conditions 18.75 mpg is achieved at 60 mph. This value
can be correlated with Figure 4. The resulting percentage increase in mpg with
corresponding increases in temperature, is shown in Figure 13.
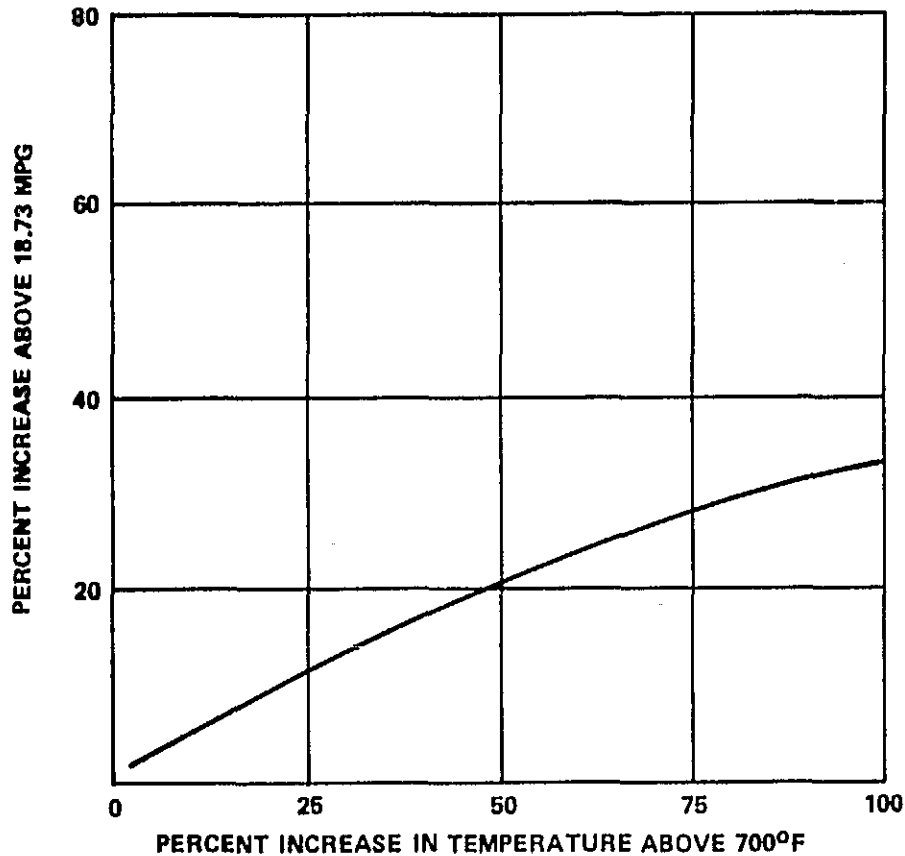
Figure 13.   Percent increase in mpg of the Guffin engine above that
which is achieved at the optimum cutoff at 700° F and 800 psia.

As for the efficiency of the Carnot cycle, Figure 12 shows that relative
large percentage increases in temperature produce only mediocre increases
in mpg. A close comparison of these two figures reveal a surprising result.
The two curves are identical. Figure 12 relates to increases in efficiency of
the Carnot cycle whereas Figure 13 relates to increases in mph of the Guffin
Engine. The reason for this exact correlation is unknown. There may be an
algorithm between performance of the Carnot Cycle and the Guffin Engine. If
such be the case, there would be a simple rule to convert Carnot efficiency
into mpg capability of the Guffin Engine. Although this is a most interesting
problem, its solution is not within the scope of the objective at hand. It will
not be persued any further in this report.

First Objective

Now that some fundamentals events and characteristics have been established, the first objective will be attempted. This objective is to illustrate the relationship between theoretical thermal efficiency of the Rankine Cycle and mpg capability of the Guffin Engine. This relationship will be demonstrated by cross-plotting optimized performance of the Guffin Engine to correspond to respective pressure and temperature which establish the theoretical thermal efficiency.

Equations governing the performance of the Guffin Engine have already been established. A chart giving the theoretical thermal efficiency as a function of pressure and temperature is given in Figure 14. Before a cross plot like this is attempted, some ground rules must be established. The specific results are therefore limited to the ground rules, but trends remain. In the case of theoretical thermal efficiency, steam expands isentropically from the specified pressure and temperature to a temperature of 220° F. For the Guffin Engine, the following conditions apply:

Speed = 60 mph,

Weight = 3500 lbs,

Cutoff = 7 percent, and

$V_C/V_T = 0.04.$

Results of the cross plot are given in Figure 15. It is emphasized that the specific values of mpg will change drastically with cutoff, speed, and clearance volume. However, it may be argued that the characteristics of the correlation remain. Thus, at the outset, for any given value of theoretical thermal efficiency (with its associated pressure and temperature), the corresponding mpg can be changed over a very wide range. Thus, it is impossible to generalize between theoretical thermal efficiency and mpg. The only way a correlation can be demonstrated is to use specified conditions as was used in the construction of Figure 15. A study of Figure 15 will reveal difficulties in trying to establish sensible rules, even for the specified conditions. For example, take the low temperature range between 800 and 2000 psia. The mpg changes -14 percent for a +13 percent change in efficiency. This is directly opposite of the expectations of the one-to-one rule. However, visualize the effects of pressure at the higher temperatures. Actual increases in efficiency resulting from increases in pressure result in a decrease in mpg. This trend
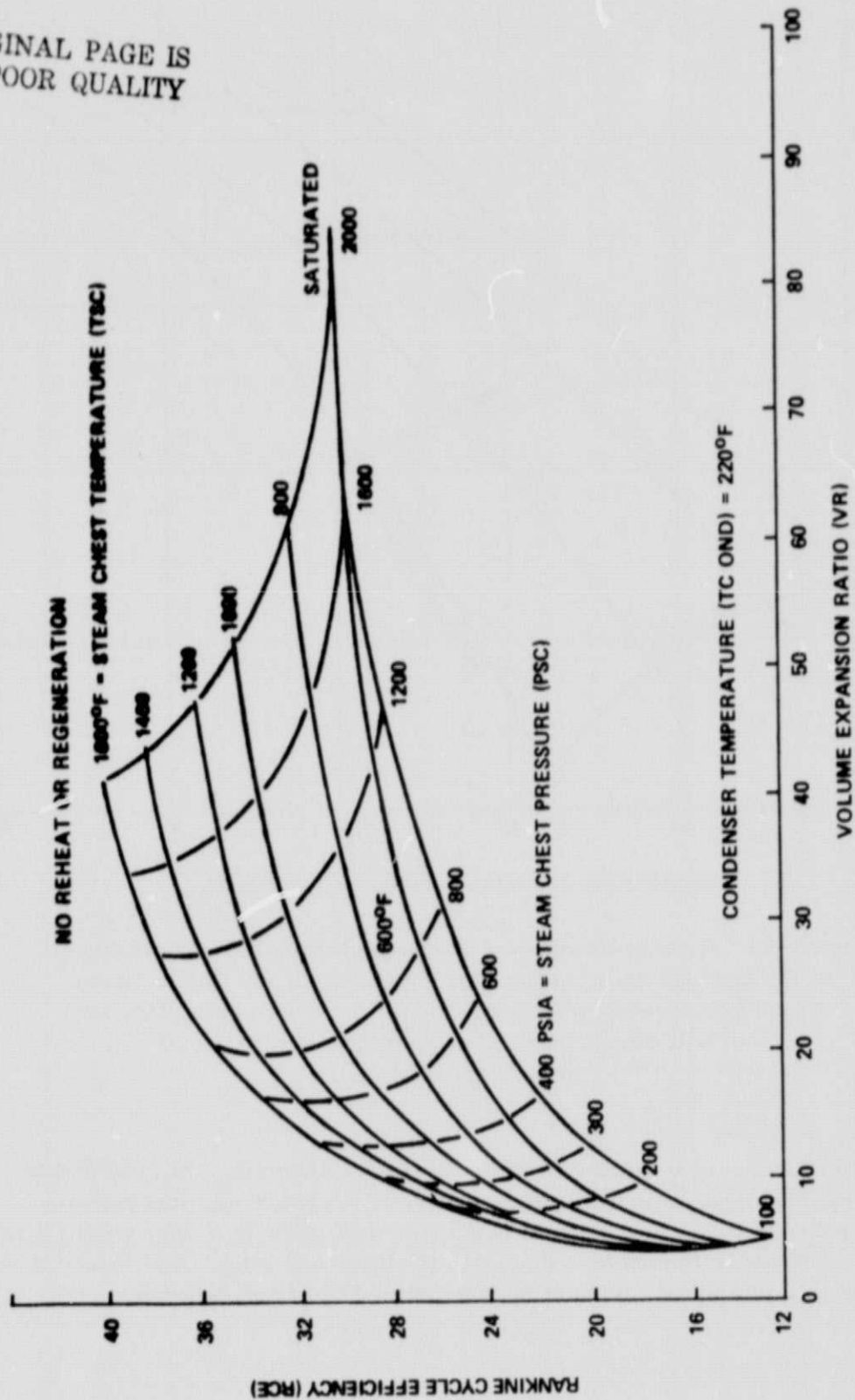
Figure 14. Effect of expansion ratio on Rankine cycle efficiency for various steam chest temperatures and pressures.

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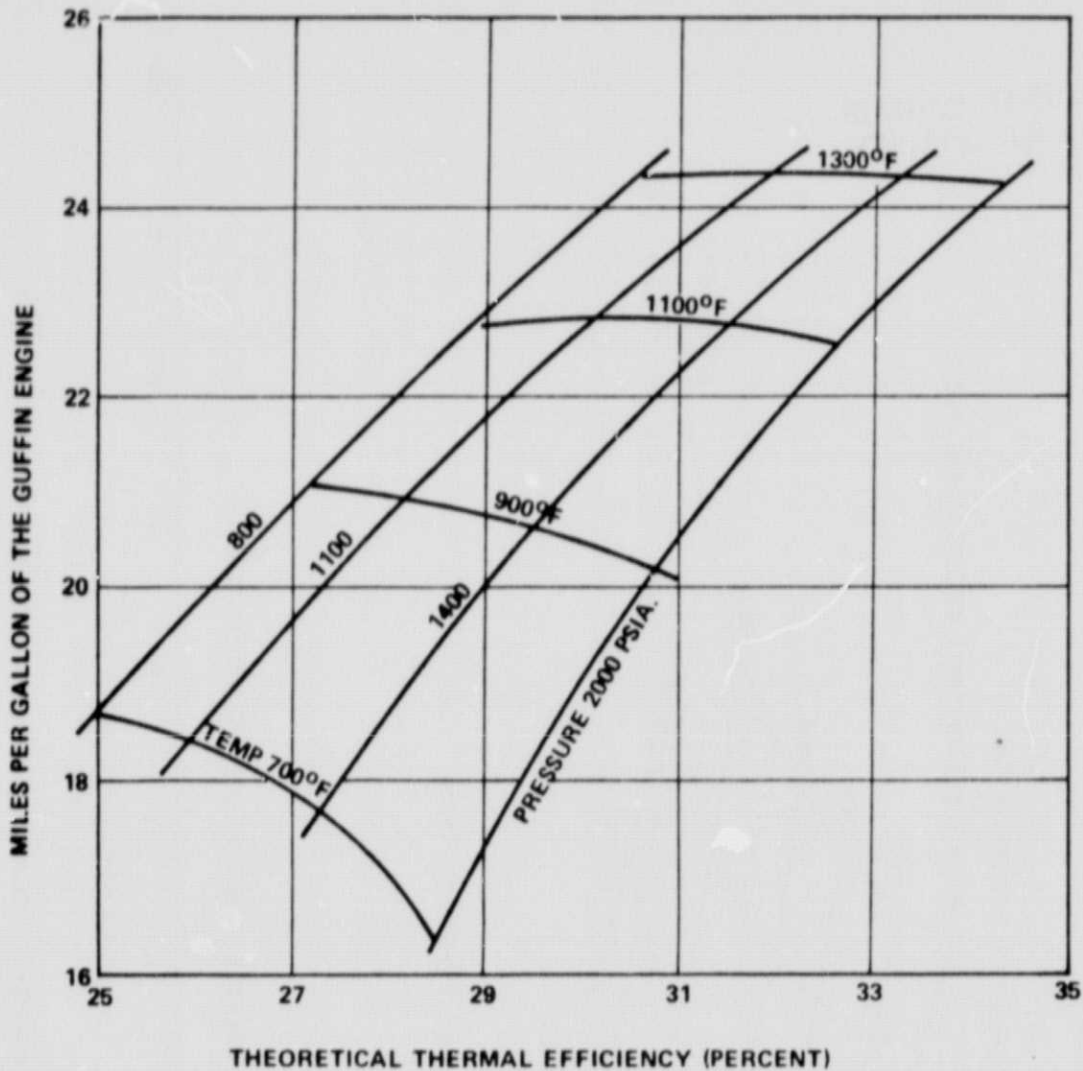
Figure 15. A bold attempt to relate theoretical thermal efficiency
of the Rankine cycle to the mpg capability of the Guffin steam
engine (increase in pressure will increase thermal efficiency,
but will result in either a decrease in mpg or no
improvement at all).

is opposite to what is expected from the classical analysis. At 1000° F and
above, mpg becomes essentially independent of pressure and theoretical
thermal efficiency. It is therefore concluded that there is a very weak (if any)
correlation between theoretical thermal efficiency and mpg. And indeed if any
correlation is attempted, it must be constrained by strict definitions.

Second Objective

So far results have been negative. The second objective attempts to produce positive results by relating laboratory measured system efficiency, η_S, to mpg. Laboratory measured efficiency is simply the engine power output divided by the equivalent heat power input. If this ratio is known along with other test conditions, a meaningful correlation can be made between mpg and efficiency. This relationship is the Thompson Equality, and its derivation is presented in Appendix D. From Appendix D, the Thompson Equality is:

$$mpg = 11.1 \frac{\eta_S E \rho}{\left(\frac{PV_T}{G}\right) D_G}$$ (11)

The term η_S is the laboratory measured system efficiency. The pressure, P, and Guffin Dour Number, D_G, must be comensurate with the measured value of system efficiency. The computed mpg does not allow for transmission and drive losses. It would be technically wrong to substitute arbitrary values of η_S, P, and D_G.

The Thompson Equality is unique in that it does not indicate at what vehicle speed the mpg is applicable. In order to compute the applicable speed, the limiting velocity equation of Appendix B must be utilized.

It was stated above that indescriminate substitution into the Thompson Equality is improper. However, in an analysis of the Guffin Engine, much data are available showing that the computed mpg is compatible with the Guffin Abatement Number and Dour Number. A theoretical system efficiency can therefore be computed by substituting compatible values into the Thompson Equality. This type computation has been accomplished for three conditions:

(1) 60 mph for a 3500 lb vehicle,

(2) 40 mph for a 3500 lb vehicle, and

(3) 40 mph for a 5500 lb vehicle.

Data for each of these three conditions are given in Tables 5, 6, and 7, respectively. Each table includes data for a range of temperatures and cutoffs. For each combination, mpg and calculated engine efficiency is given.

TABLE 5. ENGINE EFFICIENCY DATA FOR STANDARD CONDITIONS (60 MPH AND 3500 LB VEHICLE).*

Temp (°F)	Geometric Cutoff (Decimal)	Guffin Abatement Number	Guffin Dour Number	mpg	System Efficiency (Percents)
700° F	0.05	83625	0.173	18.89	18.83
	0.20	32092	0.450	16.40	16.32
	0.40	20948	0.689	13.30	13.22
900° F	0.05	83625	0.173	21.36	21.29
	0.20	32092	0.450	18.42	18.33
	0.40	20948	0.689	14.92	14.83
1100° F	0.05	83625	0.173	22.97	22.89
	0.20	32092	0.450	19.77	19.67
	0.40	20948	0.689	16.00	15.91

*Theoretical values of engine efficiency based upon utilization of the Thompson equality. Comparison with Tables 6 and 7 show that engine efficiency is independent of vehicle speed or mass. Engine efficiency depends only upon the Guffin Abatement Number and Dour Number for the corresponding temperature and cutoff. These results should be expected and confirm the over validity of the Guffin engine analysis.

These data are unique in that for each respective temperature and cutoff the system efficiency is unchanged. For example, from Table 5, the system efficiency at 60 mph for a 3500 lb vehicle, at 900° F and 20 percent cutoff, is 18.33 percent. This efficiency is exactly identical to the same condition of Table 7, where the speed is 40 mph and the vehicle weight is 5500 lb. However, this characteristic should be expected since the efficiency test results would not be a function of the vehicle weight. In other words, the engine being tested

TABLE 6. ENGINE EFFICIENCY DATA FOR STANDARD CONDITIONS
(40 MPH AND 3500 LB VEHICLE).*

Temp (°F)	Geometric Cutoff (Decimal)	Guffin Abatement Number	Guffin Dour Number	mpg	System Efficiency (Percents)
700	0.05	54149	0.173	29.18	18.83
	0.20	20780	0.450	25.33	16.32
	0.40	13564	0.689	20.54	13.22
900	0.05	54149	0.173	32.99	21.29
	0.20	20780	0.450	28.45	18.33
	0.40	13564	0.689	23.04	14.83
1100	0.05	54149	0.173	35.47	22.89
	0.20	20780	0.450	30.53	19.67
	0.40	13564	0.689	24.71	15.91

*Theoretical values of engine efficiency based upon utilization of the
Thompson equality. Comparison with Tables 5 and 7 show that
engine efficiency is independent of vehicle speed or mass. Engine
efficiency depends only upon the Guffin Abatement Number and Dour
Number for the corresponding temperature and cutoff. These
results should be expected and confirm the over validity of the
Guffin engine analysis.

has no knowledge of the vehicle for which it is intended. However, in every
case, the achievable mpg is different. Thus, it becomes obvious that a given
efficiency cannot be translated into a single value of mpg unless vehicle weight
and speed are specified. The summation of these two tables are given in
graph form in Figure 16. As expected, a single value of efficiency can produce
a wide variation in miles per gallon, depending upon vehicle mass and speed.

TABLE 7. ENGINE EFFICIENCY DATA FOR STANDARD CONDITIONS
(40 MPH AND 5500 LB VEHICLE).*

Temp (°F)	Geometric Cutoff (Decimal)	Guffin Abatement Number	Guffin Dour Number	mpg	System Efficiency (Percents)
700	0.05	108953	0.173	14.50	18.83
	0.20	41812	0.450	12.59	16.32
	0.40	27293	0.689	10.21	13.22
900	0.05	108953	0.173	16.40	21.29
	0.20	41812	0.450	14.04	18.33
	0.40	27293	0.689	11.45	14.83
1100	0.05	108953	0.173	17.63	22.89
	0.20	41812	0.450	15.17	19.67
	0.40	27293	0.689	12.28	15.91

*Theoretical values of engine efficiency based upon utilization of the
Thompson equality. Comparison with Tables 5 and 6 show that
engine efficiency is independent of vehicle speed or mass. Engine
efficiency depends only upon the Guffin Abatement Number and Dour
Number for the corresponding temperature and cutoff. These
results should be expected and confirm the over validity of the
Guffin engine analysis.

If Brand-X steam engine is advertised as achieving a certain fuel consumption
as a result of a measured efficiency, the information is almost meaningless
unless the vehicle speed and weight are specifed for the performance achieved.
If Brand-Y wants to compare his system performance with Brand-X, it is
suggested that the Thompson Equality computation be used and the associated
speed be calculated, as given in Appendix B. It is certainly obvious that the
one-to-one relationship between mpg and efficiency is not valid. Many possi-
bilities exist depending upon specific vehicle definition.
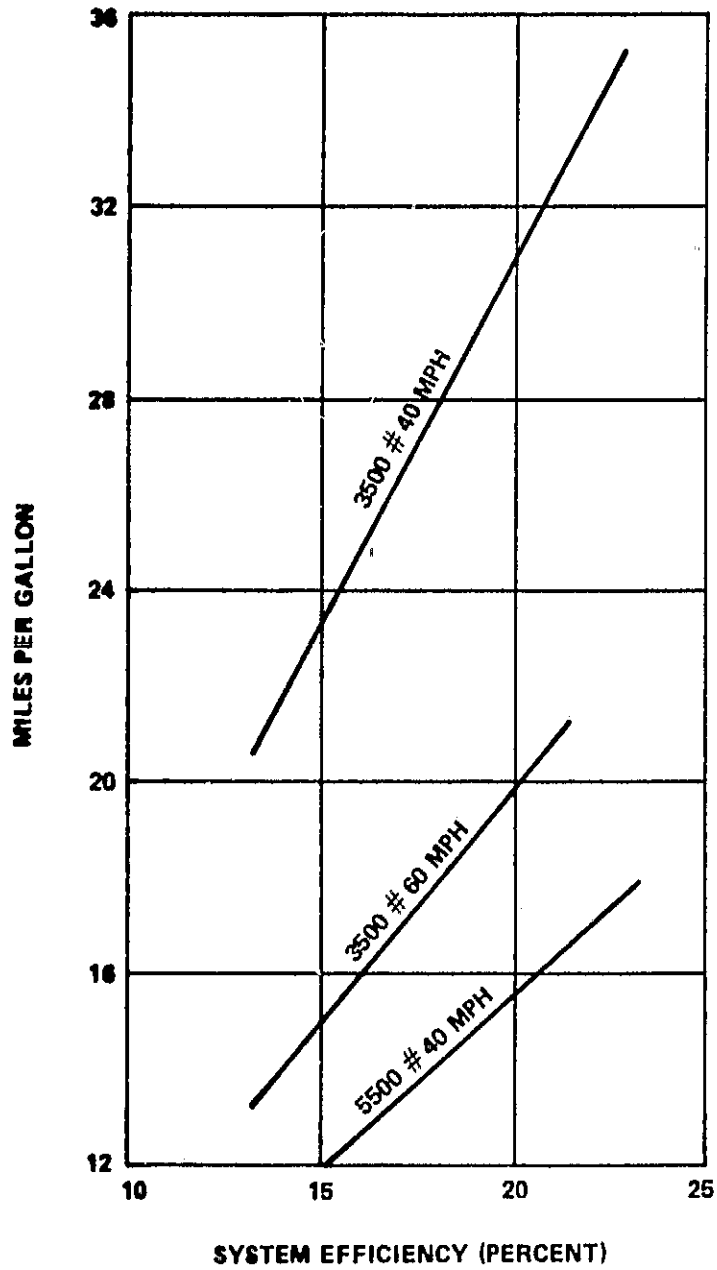
Figure 16. Relationship between laboratory measured efficiency
and mpg of the Guffin Engine (the one-to-one relationship between
efficiency and mpg is invalid as a generalized conclusion).

VIII. TINKER ENGINE CYCLE

A novel regenerative steam cycle has been proposed by Tinker* which duplicates the processes of the Guffin Engine and also has unique design factors which reduce system components. In view of the lessons learned by studying the Guffin engines, this particular cycle deserves introduction to the scientific steam community.

In order to understand Tinker's approach, consider Figures 17, 18, and 19. Figures 18 and 19 are vertical adaptations of the same principles as the horizontal adaptations introduced in the basic Tinker Patent. First, think of the volume suggested by the area a-b-h-i (Fig. 17) as a clearance volume when the piston is at TDC. This clearance volume involves the usual clearance between piston and cylinder head, plus admission porting and the volumes V-V of Figures 18 and 19. In the chambers V-V, the clearance volumes are caused by the fine mesh screen separating them, the screen being marked S.

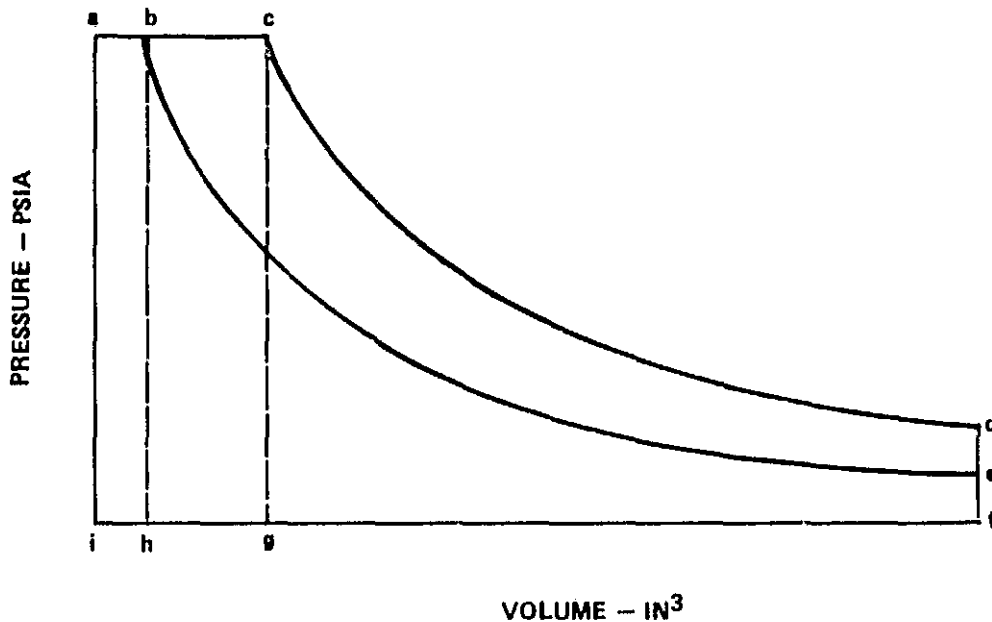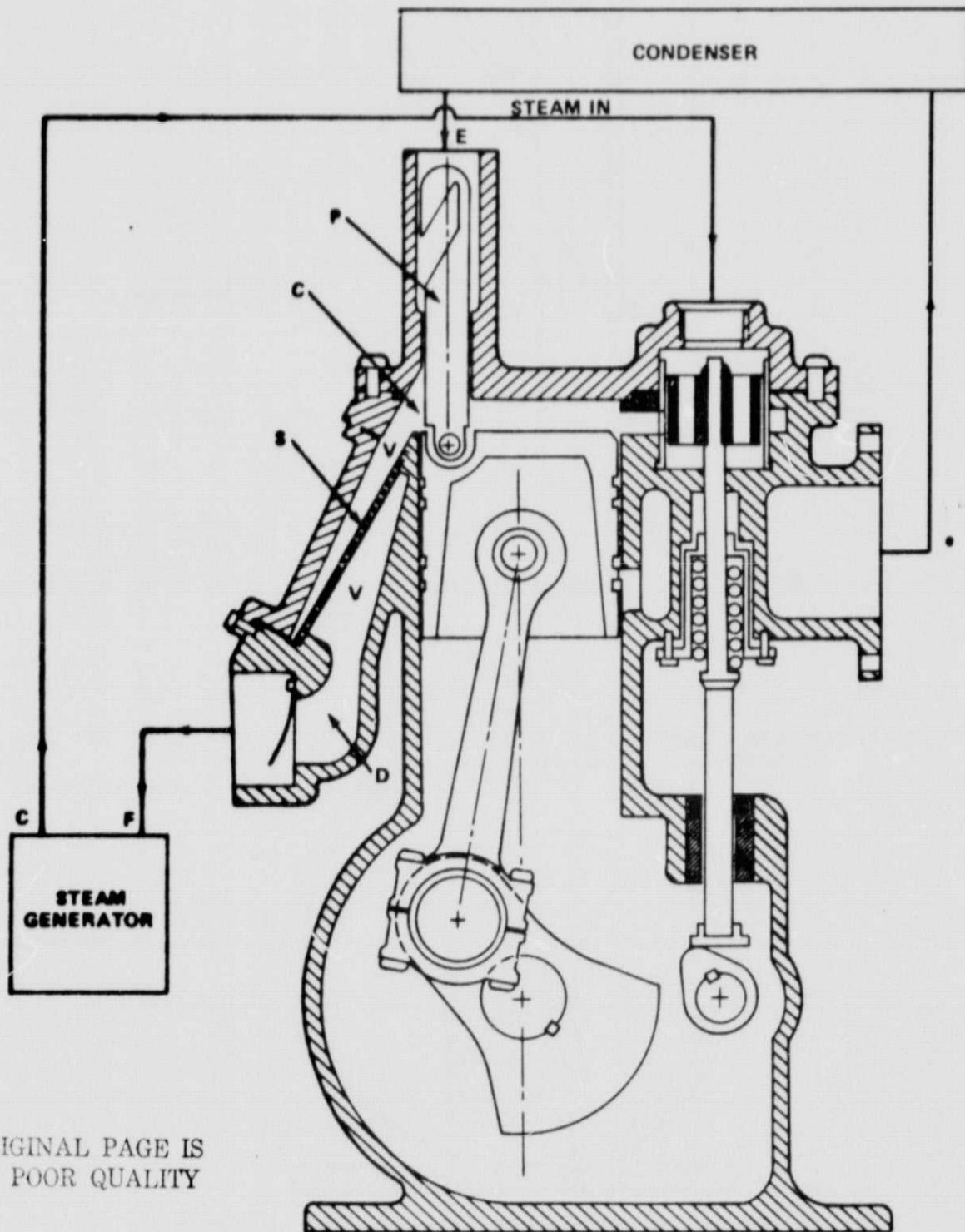VOLUME — IN^3

Figure 17. P-V diagram for Tinker engine.

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*Patent No. 3,877,231.

CONDENSER

STEAM IN

E

F

C

S

V

V

D

C

F

STEAM GENERATOR

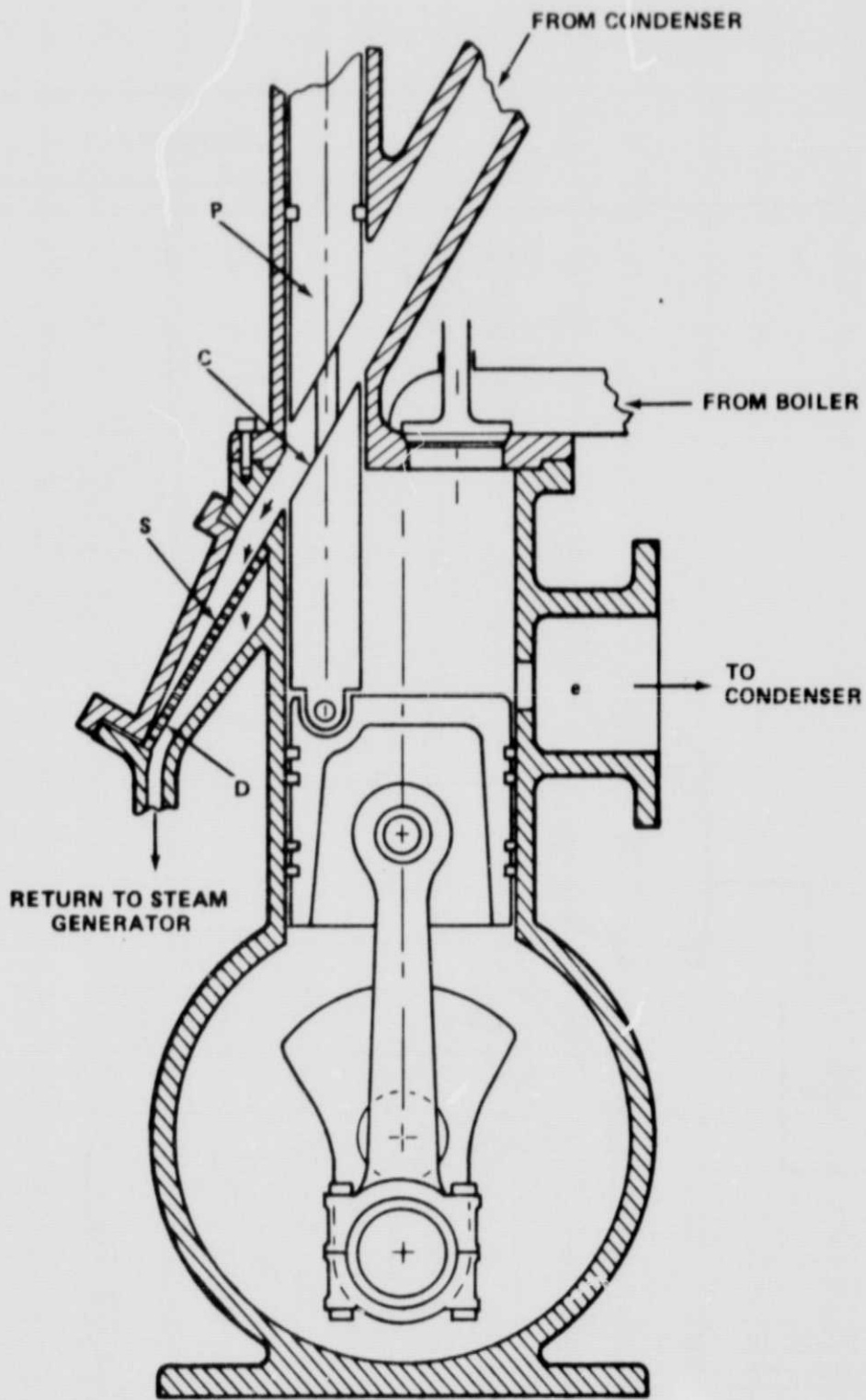Figure 18.   Tinker engine cross section at TDC.

Figure 19. Tinker engine cross section at BDC.

Admission begins along b-c of the P-V diagram and, as indicated in Figure 17, cutoff is at c, followed by adiabatic expansion down to d. At this juncture, any type of valve gearing may be used, and either cutoff control, throttle control or other controls, depending upon overall system design objectives and plans.

Between d and e is what is usually considered to be a constant volume expansion or free expansion when the piston uncovers the exhaust porting, with a portion of the steam going to the condenser, as indicated in Figure 19, and a portion remaining in the cylinder.

In the position shown in Figure 19, the plunger pocket throws liquid from the condenser onto the screen. This throwing event takes place because maximum acceleration occurs at the end of the stroke, just prior to bottom dead center, when the piston and attached plunger approach zero velocity. The resultant throw vector occurs from a component of plunger velocity and velocity from any pressure differential affecting the liquid in the pocket.

At the start of compression, at e, the entropy is represented by the combination of steam that remains in the cylinder and the liquid that returns from the condenser. The effective wet compression negative work between e and b may be considered adiabatic and reversible if proper account is made for the mixing of the characteristic mass with the injected liquid mass. At TDC, a terminal condition F results where both steam at high pressure, along with dispersed liquid and pure liquid exists. These ideas are illustrated on the T-S diagram of Figure 20. This figure shows the mixing of the normalized characteristic mass, y^*, with the injected pocket mass of state point E. The injected mass has an increase in entropy and the characteristic mass has a decrease in entropy. The results of the mixing is represented at state point, G. The mixing which results in point G can be described by $(h_{e_1} - h_G)$ $y = (h_G - h_E)(1 - y)$. The effective compression work is $(h_F - h_G)$.

It is noted that the effective mixing process does not take place completely within the cylinder. A large part of the entropy loss from point e_1 to G results from energy removed by the condenser. The T-S diagram does not represent simultaneity. Events occur at different times. Some occur over a time period, as in the example of the condenser. Figure 18 relates the state points to the actual hardware.

Consider the process that occurs to the normalized characteristics mass, y, and that portion which is returning to the condenser. The portion returning to the condenser is $(1 - y)$. Ultimately this amount must undergo a

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*Characteristics mass divided by admission mass.
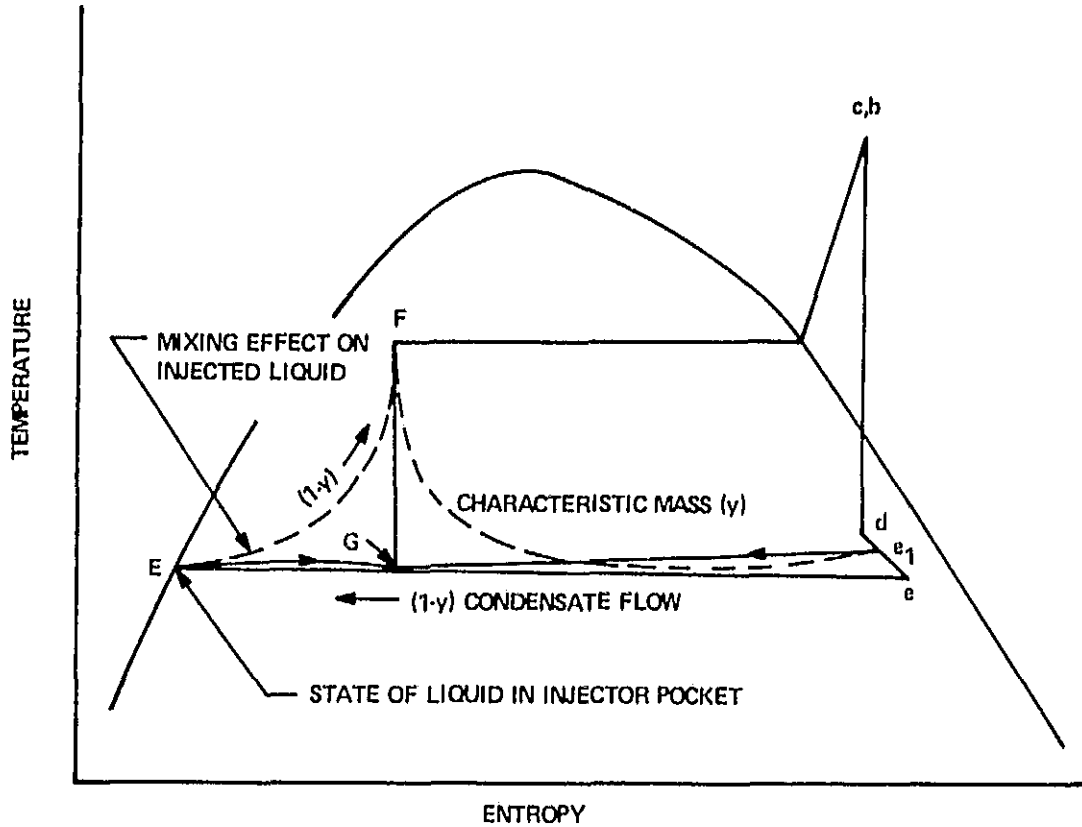
Figure 20. A chart of entropy versus temperature for the Tinker engine.

condensation/pressurization and heat addition to return to system operating conditions. The normalized characteristic mass in its compression mixing state, as the piston returns to TDC, must have some arbitrary state as indicated at F. Point F will always be at system pressure, but not necessarily at system temperature. The exact location of point F can probably be determined only by tests. Of course, the actual mixing of the characteristic mass reduced the total heat load which must be applied to the steam generator. This fact can be expressed in terms of the cycle thermal efficiency, which would be:

$$\eta = \frac{(h_C - h_d) - (h_F - h_G)}{(h_C - h_F)} \quad , \tag{12}$$

The efficiency, as described in equation (12), will always be greater than that of the classical Rankine cycle. This fact results from the reduced head load $(h_C - h_F)$, even though the compression work $(h_F - h_G)$ is greater. The steam generator feed pump work must be relatively small because the mixture is at or near boiler pressure and only a transfer loss is involved. If the admission port should be arranged so that the main piston closes it off just before dead center, the main piston performs the function of the feed pump. It is commonly known that often the uniflow engine must have relief valving to relieve the compression after the main cylinder ports are closed by the piston; but, in this cycle, this condition is automatically handled. The Tinker engine handles this automatically and with simplicity.

The ultimate thermal gain involves the difference in enthalpy of the admitted steam from b to c on the diagram and the heat in the liquid returned to the boiler at the end of wet compression, at b or at any designed terminal compression pressure, as with the generalized condition defining the Guffin Engine.

There seems to be three vital areas of design which are likely to arise in the mind of the reader as objectives or points involving excessive loss or even impossibilities. For example, compression phase of any vapor cycle is usually frowned upon. These critical considerations are:

(1) The plunger pocket must receive fluid from the condenser much like the circumstances of cavitation (NPSH) with a centrifugal pump. Therefore, pocket shape is determined by limiting rpm. Sealing of the plunger will not be easy, but is well within sealing technology required to solve this problem.

(2) Since the port marked e (Figs. 18 and 19) is open all the time, it can be argued that the new steam entering will be considerably condensed in the cavities V-V surrounding the screen (or screens, as the design may be). It is imperative that during the compression stroke the liquid will go to the bottom of the cavity at the point marked D. The main point is that the liquid at the bottom will expose very little of its surface for condensation. No doubt some condensation will occur as in any clearance space. The reed valve shown will deliver fluid back to the generator. At high rpm, it is suspected that the flow will condensate into the plunger pocket and the flow of hot liquid, via the final discharge valve, may be rather continuous or almost approach a steady flow.

(3) What about the matter of efficiency of regenerating and time period during a compression stroke required for it to happen? According to the inventor, steam water injectors at about any combination of pressures and temperatures can accomplish high rates of mixing and heat transfer in small volume, even though stratification may occur. Intimate mixing with high interface is enhanced by the screen (or screens).

Items 1, 2, and 3 above relate to the dumping, spraying, or throwing of the condensate from the condenser by a low pressure device consuming relatively little energy, as compared to high pressure pumps and their problems. It also must be emphasized that the contact between liquid condensate from the condenser and the vapor trapped in the main cylinder at the start of wet compression, at e, does not occur in the cylinder, but in the separate compartment, known or identified for clarity as the screen cavity. The screen cavity design enhances the effectiveness of the cycle.

IX. CONCLUSIONS

Typical uniflow type steam engines provide maximum miles per gallon at about 7 percent cutoff. High pressure will degrade miles per gallon rather than improve it. Increased temperature will improve miles per gallon, but at a decreasing rate.

Theoretical thermal efficiency of the classical Rankine Cycle cannot be generalized into a simple criteria for determining miles per gallon. System efficiency, as measured in the laboratory, can be converted into miles per gallon by the appropriate relationships given.

APPENDIX A

MILES PER GALLON RELATIONSHIP FOR THE GUFFIN ENGINE

This derivation is developed for a vehicle being driven in a straight path on a level road. It is general and can be applied to any vehicle size and weight. Engine friction and drive train losses are not included. However, anyone interested in performing a point design can include friction characteristics after studying and understanding this derivation. The results reported herein are therefore optimistic for vehicles employing the Guffin type engine.

The author does not intend to imply that there is anything new about the Guffin engine. The term Guffin has been selected as an identity separate from the ideal P-V diagram. Actually, the Guffin P-V diagram has been tailored to represent the type of diagram usually associated with uniflow design. However, since "uniflow" is a broad term which can be associated with pressure release values, it would be possible to have a uniflow engine performance that would not be compatible with the derivation given in this report. Therefore, in considering engines having clearance volume, compression work, and release volume, strict definitions and assumptions are required. The Guffin engine P-V diagram is illustrated in Figure A-1. It is identical to the Ideal P-V diagram, except provisions have been made for clearance, compression work, and release volume. Parameters used for this analysis were:
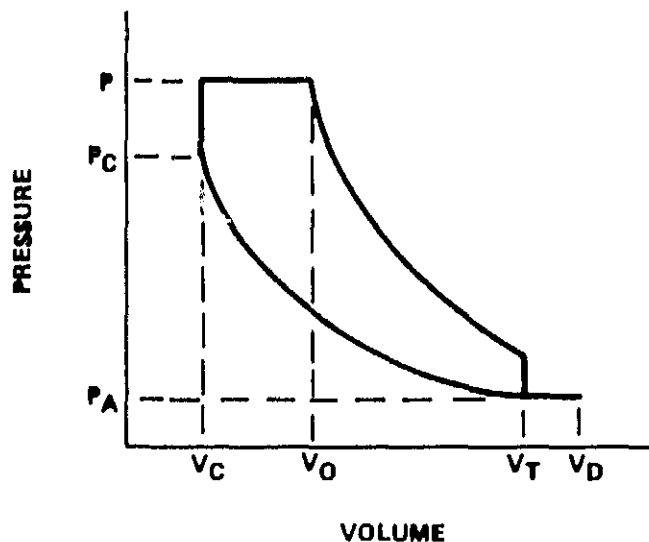

Figure A-1. P-V diagram for the Guffin engine.

P = throttle pressure (psia)

P_C = compression pressure at TDC

P_A = condenser suction pressure (psia)

V_C = clearance volume (in.^{3})

V_0 = volume at cutoff (in.^{3})

V_T = volume at release (in.^{3})

V_D = volume at BDC

For further clarification, the release volume would be $(V_D - V_T)$ and the engine displacement would be $(V_D - V_C)$.

Compression begins at V_T and continues until some compression pressure, P_C, is reached. At TDC, the admission valve opens and mixes steam with the compressed residual steam (characteristic mass), to pressure, P. It is important to recognize the need of accounting for the effect of mixing the residual steam with the admission steam. First, the characteristics steam mass will affect the additional admission mass required to displace the piston to its cutoff volume, V_0. Second, the final mixing temperature at cutoff will be less than the temperature of the admission steam.

As for as the applicability of this derivation, there are two constraints

$$\frac{V_0}{V_T} \ge \left(\frac{P_A}{P}\right)^{\frac{1}{\gamma}} \qquad \text{(Expansion Constraint)} \qquad\qquad \text{(A-1)}$$

$$\frac{V_C}{V_T} \ge \left(\frac{P_A}{P}\right)^{\frac{1}{k}} \qquad \text{(Compression Constraint)} \qquad\qquad \text{(A-2)}$$

The first constraint is required to prevent "looping" and the second limits the compression pressure to less than or equal to the admission pressure, p.

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There are simplifying assumptions made in the miles per gallon deriva-
tion. These will be introduced and justified as they arise. Now that definition
has been given as to what is being studied, consider these arguments. There
are two steam mass flow rates that are important and necessary in analysis
of a steam Rankine system. First is the steam mass flow rate, M_G, capable

of being generated by the steam generator. Second, is the steam mass flow rate,
M_R, required to sustain a set of operating conditions for a given engine speed.
The first flow rate is readily obtainable [1] and obvious from water heating
requirements,

$$ M_G = \eta_B \frac{EF_R \rho}{60 \, \Delta H} \quad . \tag{A-3}$$

Obtaining an expression for the required mass flow rate is very difficult
because of the mixing phenomenon already mentioned. To begin with, we con-
sider the effect of the compressed residual steam upon the final mixing tempera-
ture at cutoff. This situation is illustrated in Figure A-2. The characteristic
mass, m, at its compression temperature, T_C, is mixed with the required

admission mass, M_R, at temperature, T_0. After mixing a total mass $(M_R +$
m), this results in a final mixing temperature, T_F. The characteristic mass
is replaced every revolution and therefore represents a flow having units of
lb/min.
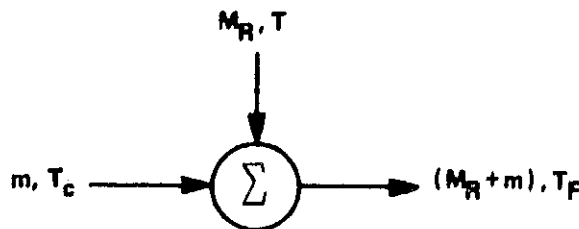

Figure A-2. Mixing of the characterics mass with
the admission mass.

An energy balance for the heat gained by the characteristics mass must
be equal to the heat loss by the admission mass,

$$ M_R \int_{T_F}^{T} C_{P1} \, dT = m \int_{T_C}^{T_F} C_{P2} \, dT \quad . \tag{A-4}$$

The heat capacity C_{P1} is associated with the required admission mass, and C_{P2} is associated with the characteristic mass. Equations for heat capacity, as a function of pressure and temperature, do not allow an explicit solution for T_F.

There is a nature about the variation of heat capacity which lessens the agony. First, high pressure, high temperature steam would in general be mixed with relatively low pressure, low temperature steam. These differences are compensating and brings the heat capacity of each mass nearly equal. For example, at 1000° F and 1200 psia, the heat capacity is about 0.59 whereby, at 700° F and 600 psia, the heat capacity is about 0.58. The worse case occurs at high pressure and low temperature. For example, at 1600 psia and 700° F, the heat capacity is about 0.82.

In order to reduce equation (A-4) into a usable form, it will be assumed that an average C_{P1} and C_{P2} exists, such that:

$$M_R C_{P1} \int_{T_F}^{T} dT = m C_{P2} \int_{T_C}^{T_F} dT \quad . \tag{A-5}$$

Solving this expression, we find:

$$T_F = \frac{M_R T + m \dfrac{C_{P2}}{C_{P1}}}{M_R + m \dfrac{C_{P2}}{C_{P1}}} \tag{A-6}$$

Now we notice that no matter how far apart the two heat capacities may be, there is a compensating trend, since C_{P2}/C_{P1} occurs both in the numerator and denominator.

For those cases where high admission pressure and temperatures are involved, $C_{P2}/C_{P1} = 1.0$ is a good assumption. If the admission pressure and temperature is very close to the pressure temperature of the characteristic

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mass, $C_{P2}/C_{P1} = 1.0$ is an excellent assumption. In those cases involving high admission pressure and low temperature, C_{P2}/C_{P1} results in an optimistic T_F. Even so, the error is considered small since the ratio $0.82/0.58$ is proportionately small to both the numerator and denominator, since $M_R T_0 > m\, C_{P2}/C_{P1}\, T_C$ and $M_R > C_{P2}/C_{P1}\, m$.

Thus, the author feels that the following equation is a good representation of the final mixing temperature for the wide temperature and pressure ranges considered. In the worse case, the final mixing temperature will result in optimistic performance. Thus, in this respect we know the limits achievable,

$$T_F = \frac{M_R T + mT_C}{M_R + m} \quad . \qquad\qquad (A\text{-}7)$$

In the order to solve for the required admission mass, M_R, it is necessary to write another equation involving T_F and M_R. Thus, T_F can be eliminated and the equations can be solved for M_R.

This additional equation can be simply formulated by writing the equation of state at the cylinder volume, V_0 (referring to Figure A-1),

$$\frac{N}{12} PV_0 = [M_R + m]\, Z_F R T_F \quad , \qquad\qquad (A\text{-}8)$$

where Z_F is the compressible factor at pressure P and temperature T_F.

Combining equation (A-8) with (A-7), gives

$$\frac{N}{12} PV_0 = Z_F R\, [M_R T + mT_C] \quad . \qquad\qquad (A\text{-}9)$$

The characteristic mass, m, is

$$m = \frac{P_A V_T N}{12 \, Z_A RT_A} \quad ,$$

(A-10)

where Z_A is the compressible factor for the exhaust conditions.

The relationship for the compression temperature is

$$\frac{T_C}{T_A} = \left(\frac{V_T}{V_C}\right)^{k-1} \quad .$$

(A-11)

Substituting these two expressions into equation (A-9) and solving for the required mass flow rate, M_R, is

$$M_R = \frac{PV_T}{12 \, Z_F RT} \left[\frac{V_0}{V_T} - \frac{Z_F R}{Z_A R} \left(\frac{P_A}{P}\right) \left(\frac{V_C}{V_T}\right)^{1-k} \right] N \quad .$$

(A-12)

Now, we are faced with another dilemma. Expressions must be developed for the compressible factors. The characteristic mass is easy to deal with, since, at release the mass is at low pressure and low temperature (near saturation). For this reason, $Z_A R$ will be taken as a constant 85.8 ft-lb/lbm - °R, which is the steam gas constant for saturated vapor at atmospheric pressure. In most cases, we would expect a slight difference in the supply temperature and the final mixing temperature. It will be assumed that $Z_F = Z_0$, where Z_0 is the compressible factor at the supply temperature, T , and pressure, P. The change in ZR with pressure and temperature is illustrated in Figure A-3. Although a wide variation is illustrated, errors are introduced only by the difference in value read at two temperatures. For example, assume that the supply temperature is 900° F at 1400 psia. Reading from the chart, $Z_0 R$ equals about 78 ft-lb/lbm-°R. Now, if the final mixing temperature is
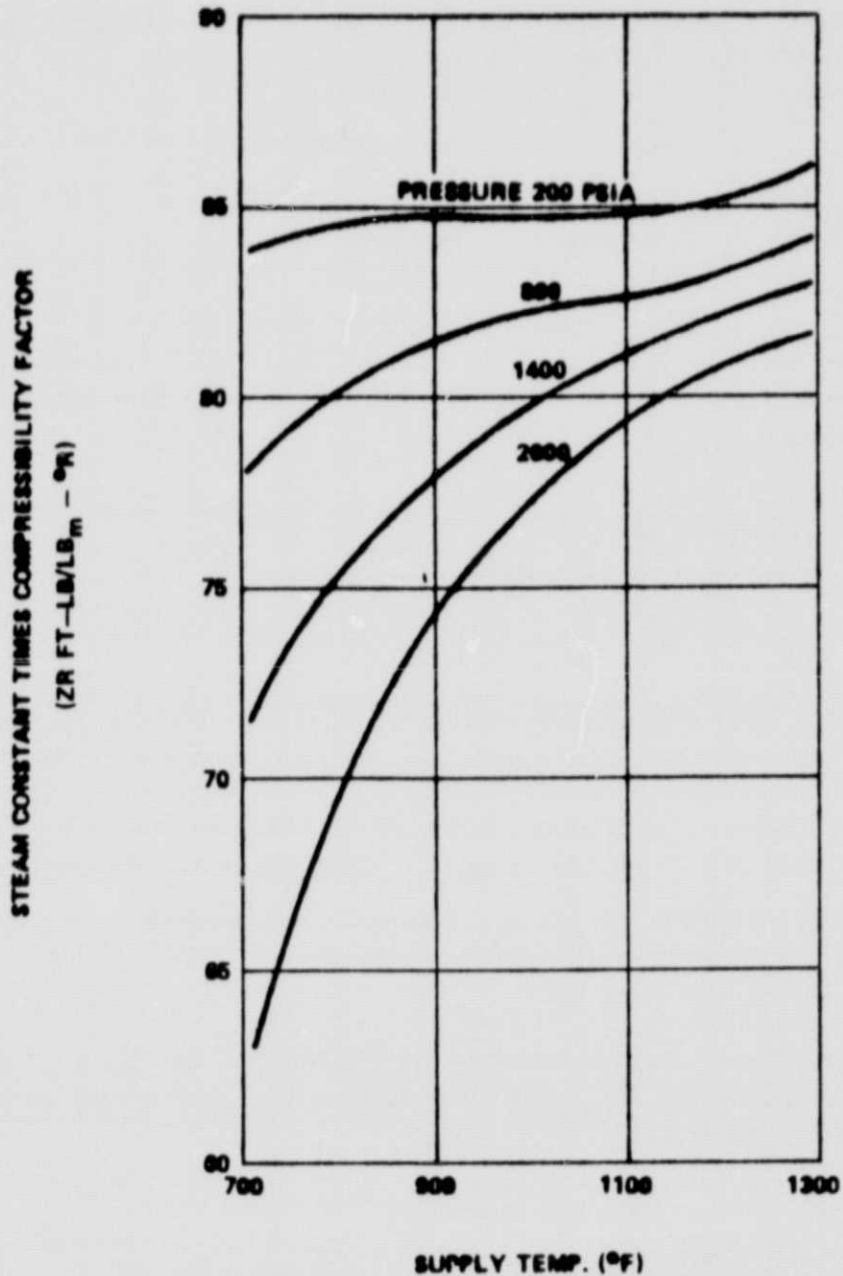
Figure A-3. Effects of the compressible factor.

$800°$ F at 1400 psia, the chart shows $Z_F R$ equal to about 76 ft-lb/lbm-$°$R. Thus, the error caused by $Z_F = Z_0$ is represented by the difference between 78 and 76. Note that this assumption always makes M_R greater than actual. Thus

the assumption is conservative. The worst errors occur at high pressures and low temperatures, where the curves have a steep slope. Using the assumption $Z_F = Z_0$,

$$Z_F R \frac{P_A}{P} \approx Z_0 R \frac{P_A}{P} \frac{T}{T} = 144 \frac{P_A v_0}{T} \quad . \tag{A-13}$$

In this expression, v_0 is the specific volume (ft^3/lb) at T and supply pressure, P. Substitution in $(A-12)$ is the expression for the mass flow rate, which is obtained thusly,

$$M_R = \frac{PV_T N}{12\ Z_0 RT} \left[\frac{v_0}{V_T} - \frac{144}{85.8} \frac{P_A v_0}{T} \left(\frac{V_C}{V_T} \right)^{1-k} \right] \quad . \tag{A-14}$$

Finally after much manipulation, we have an expression for the admission mass flow rate. The expression accounts for the final mixing temperature and compression pressure. It is worth noting that the assumption concerning the final mixing temperature produced optimistic results, whereas, the assumption $Z_F = Z_0$ results in a conservative value. Thus, the two assumption again are compensating. Equation $(A-14)$ is a very good compromise in lieu of great complexities, if precise relationships were used. A test for the adequacy of equation $(A-8)$ will be given later.

From hereon, the derivation is straightforward and exact. Under steady state driving conditions, $M_G = M_R$, setting equation $(A-14)$ equal to $(A-3)$, and solving for N, gives

$$N = \frac{\eta_B \, EF_R \, \rho}{5 \, \Delta HPV_T} \frac{1}{\left[\dfrac{v_0}{V_T} - \dfrac{144}{85.8} \dfrac{P_A v_0}{T} \left(\dfrac{V_C}{V_T} \right)^{1-k} \right]} \quad . \tag{A-15}$$

The velocity, V_M (mph), of a vehicle is

$$V_M = 0.07136 \, NG \quad . \tag{A-16}$$

Substituting the value of N from (A-15) into (A-16) and solving for the ratio, V_M/F_R, gives the desired expression for miles per gallon:

$$mpg = 0.0142 \, \eta_B E \rho \, \frac{Z_0 RT}{\Delta H} \, \frac{1}{\left(\dfrac{PV_T}{G}\right)} \, \frac{1}{\left[\dfrac{V_0}{V_T} - \dfrac{144}{85.8} \dfrac{P_A v_0}{T} \left(\dfrac{V_C}{V_T}\right)^{1-k} \right]} \quad . \tag{A-17}$$

It should be noted that equation (A-17) has been based upon a single cylinder with a release volume of V_T. If multiple cylinders are involved, V_T must be multiplied by the number of cylinders. For a multicylinder engine, V_T must represent the sum of all release volumes.

The adequacy of equation (A-17) depends upon the level of representation of equation (A-14) for the required admission mass. It's total validity cannot be determined without more rigorous analysis. However, there exists one situation where the accuracy of equation (A-14) may be tested. This occurs when the conditions are such that M_R is zero. The requirement is met when

$$\frac{V_0}{V_T} = \left(\frac{P_A}{P}\right)^{\frac{1}{\gamma}} \quad , \quad \frac{V_C}{V_T} = \left(\frac{P_A}{P}\right)^{\frac{1}{k}} \quad , \tag{A-18}$$

where $\gamma = k$. The conditions are as represented by the dashed line in Figure A-4. The expansion work is exactly equal to the compression work, resulting in zero net work. No steam is exhausted since the expansion ends at P_A.

There is zero admission steam, since the compression pressure is equal to supply pressure. In every revolution, the energy of the original steam charge is transferred from expansion work to compression work, and so on. There is zero admission work.
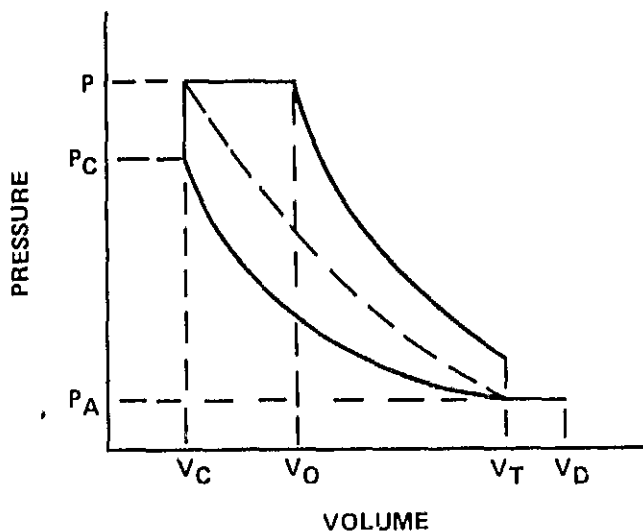
Figure A-4. Guffin engine pressure — volume diagram.

In order for the admission steam to be zero, the following portion of equation (A-14) must be zero:

$$\left[\frac{V_0}{V_T} - \frac{144}{85.8} \frac{P_A V_0}{T} \left(\frac{V_C}{V_T} \right)^{1-k} \right] = 0 \quad . \tag{A-19}$$

If the conditions for $M_R = 0$ are substituted, the above expression reduces to:

$$\left(\frac{P_A}{P} \right)^{\frac{1}{\gamma}} \left[1 - \frac{144}{85.8} \frac{v_0 P}{T} \right] = 0 \quad . \tag{A-20}$$

In order to test equation (A-20), three cases will be worked out numerically. In each case, 8 psia will be taken as a representative value for P_A. Gamma will be taken as 1.3.

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(1) <u>High Pressure, High Temperature.</u> This is the case where best agreements are to be expected, as argued previously. Let $P = 1400$ psia, $T = 1300°$ F, and from the steam tables, V_0 is 0.725 ft^3/lb. Substituting, we find that

$$\left\{ \left(\frac{8}{1400}\right)^{\frac{1}{1.3}} \left[1 - \frac{(144)(.725)(1400)}{(85.8)(1760)} \right] \right\} = +0.00045 \approx 0 \quad . \quad (A-21)$$

(2) <u>High Pressure, Low Temperature.</u> In this case, the poorest argument is expected, as stated previously. Let $P = 1400$ psia, $T = 700°$ F, and the steam tables, $v_0 = 0.406$ ft^3/lb. Substituting, we find that

$$\left\{ \left(\frac{8}{1400}\right)^{\frac{1}{1.3}} \left[1 - \frac{(144)(.406)(1400)}{(85.8)(1160)} \right] \right\} = +0.0032 \approx 0 \quad . \quad (A-22)$$

(3) <u>Intermediate Pressure and Temperature.</u> In this case, we expect an agreement between cases 1 and 2. Let $P = 800$ psia, $T = 900°$ F, and from the steam tables, $v_0 = 0.963$ ft^3/lb. Substituting, we find that

$$\left\{ \left(\frac{8}{800}\right)^{\frac{1}{1.3}} \left[1 - \frac{(144)(.963)(800)}{(85.8)(1360)} \right] \right\} = +0.0011 \approx 0 \quad . \quad (A-23)$$

Several factors may be noted:

(1) The value calculated is not M_R, but proportional to M_R.

(2) The order of results were as predicted.

(3) All computed values were positive. This means that equation (A-14) yields overall pesimistic results (more steam required than necessary). The performance reported represents the worst that can be expected at the specified pressure and temperature levels.

(4) On the basis of the values obtained for the three special cases, equations (A-14) is considered to be a very good representation of the effects of mixing the characteristic mass with the admission mass. For purposes of discussion the quantity,

$$\left[\frac{V_0}{V_T} - \frac{144}{85.8} \frac{P_A v_0}{T} \left(\frac{V_C}{V_T} \right)^{1-k} \right] = X_M \quad , \tag{A-24}$$

of equation (A-14) has been given a special name, mass ratio cutoff, and designation as X_M. The mass ratio cutoff is the ratio between the required admission mass rate and that mass required if the entire release volume was filled at the supply pressure and temperature. The usefulness of this term is indicated in the body of this report. It is noted that X_M serves the Guffin engine, as X served ideal engine described by equation (A-5) of Reference 1, page 112.

APPENDIX B

VEHICLE LIMITING SPEED CAPABILITY

For a given engine size and operating conditions, the vehicle is capable of a predetermined limiting speed. For values of P, V_T, and G used to compute mpg, a vehicle speed exists which corresponds to that computed mpg. The purpose here is to develop the expression for the vehicle limiting speed. To begin with, it is necessary to compute the engine average torque, T_E. To obtain a value for T_E, it is first necessary to compute the net work, W_N, per revolution. This will be accomplished by computing admission work, expansion work, and compression work. These individual expressions add together to give the net work.

Admission Work, W$_A$, (Shaded Area)

$$W_A = \frac{P}{12} [V_0 - V_C]$$

$$W_A = \frac{PV_T}{12} \left[\frac{V_0}{V_T} - \frac{V_C}{V_T} \right]$$
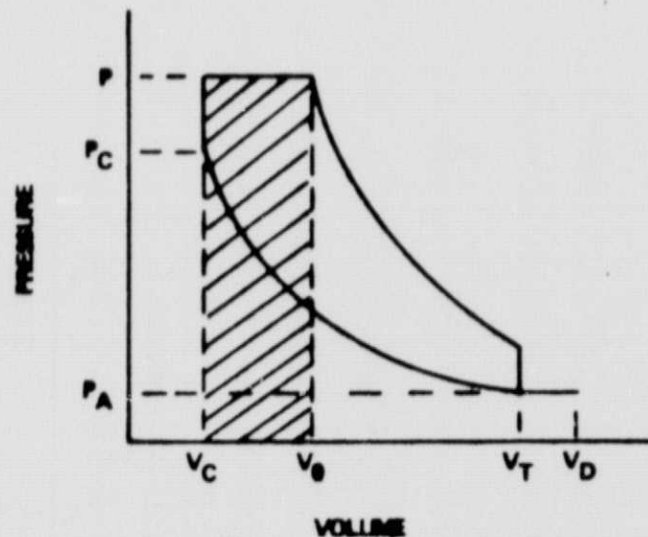
Expansion Work, W_E, (Shaded Area)

$$W_E = \frac{1}{12} \int_{V_0}^{V_T} PdV = \frac{P_0 V_0{}^\gamma}{12} \int_{V_0}^{V_T} \frac{dV}{V^\gamma}$$

$$W_E = \frac{PV_T}{12} \left[\frac{\dfrac{V_0}{V_T} - \left(\dfrac{V_0}{V_T}\right)^\gamma}{\gamma - 1} \right]$$
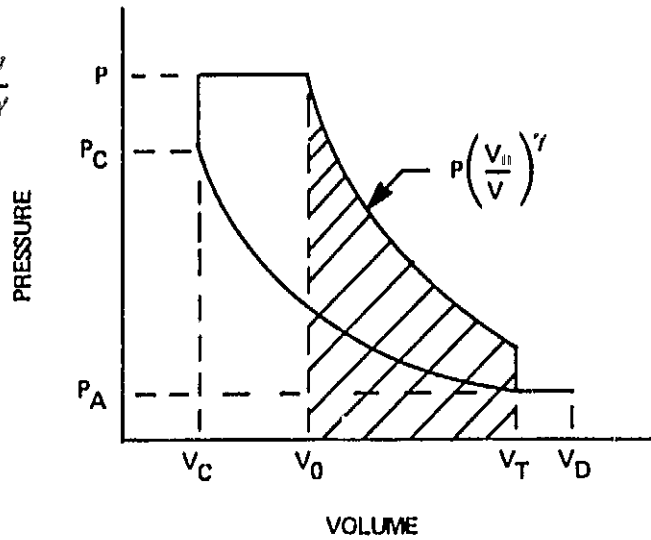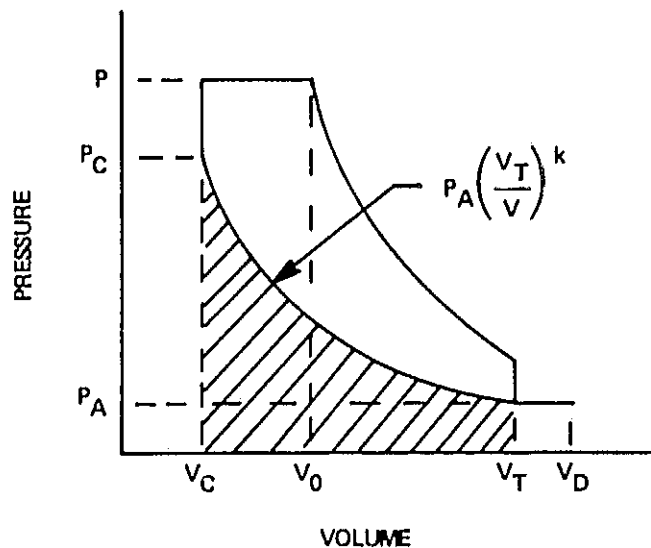

Compression Work, W_C, (Shaded Area)

$$W_C = \frac{1}{12} \int_{V_T}^{V_C} PdV$$

$$W_C = \frac{P_A V_T{}^k}{12} \int_{V_T}^{V_C} \frac{dV}{V^k}$$

$$W_C = \frac{P_A V_T{}^k}{12} \left(\frac{V_C{}^{1-k}}{1-k} - \frac{V_T{}^{1-k}}{1-k} \right)$$

Rearranging and changing to make, W_C, positive,

$$W_C = \frac{P_A V_T}{12(k-1)} \left[\left(\frac{V_C}{V_T} \right)^{1-k} - 1 \right] \quad .$$

The net work is therefore:

$$W_N = W_A + W_E - W_C \quad .$$

Making these substitutions:

$$W_N = \frac{P V_T}{12} \left\{ \frac{\frac{V_0}{V_T} - \left(\frac{V_0}{V_T} \right)^{\gamma}}{\gamma - 1} + \frac{V_0}{V_T} - \frac{V_C}{V_T} - \frac{1}{k-1} \left(\frac{P_A}{P} \right) \left[\left(\frac{V_C}{V_T} \right)^{1-k} - 1 \right] \right\} \quad .$$

(B-1)

From a historical point of view, it is desirable to relate to the classical geometric cutoff so that the net work can be related to other data by the cutoff. The geometric cutoff, X, is just the percent of piston displacement by which admission is allowed:

$$X = \frac{V_D - V_C}{V_D - V_C} = \frac{\frac{V_0}{V_T} - \frac{V_C}{V_T}}{\frac{V_D}{V_T} - \frac{V_C}{V_T}} \quad .$$

Dividing the numerator and denominator by, V_T, and solving for V_0/V_T gives:

$$\frac{V_0}{V_T} = \frac{V_C}{V_T} + X \left(\frac{V_D}{V_T} - \frac{V_C}{V_T} \right) \quad .$$

(B-2)

Equation (B-2) will not be substituted into (B-1) because of the length and resulting complexity.

However, referring to equation (1) of Reference 1* (page 9), the bracket term serves the same function as the bracket term in equation (B-1). This equation is:

$$W = \frac{PV}{12} \left(\frac{Xk - X^k}{k - 1} - \frac{P_A}{P} \right)$$

For purposes of simplification, this entire bracketed term will be referred to as the Guffin Dour Number, D_G. The net work per revolution is therefore,

$$W_N = \frac{PV_T}{12} D_G$$ (B-3)

It is noted that the Guffin Dour Number is an exact representation and does not depend upon the assumption that $P_A/P \ll 1.0$ is associated with the Dour Number for the ideal P-V diagram.

From Reference 2, the average engine torque is

$$T_E = \frac{PV_T D_G}{2\pi \, 12}$$, (B-4)

therefore, the wheel torque is

$$T_W = \frac{R_G}{r} \frac{PV_T}{24\pi} D_G$$. (B-5)

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*The symbol k used in Reference 1 has been substituted for the value of γ (polytropic expansion) in this report.

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This wheel torque must be balanced by the total retarding forces. From Kent's Mechanical Engineer's Handbook [3], the retarding Force F can be described as,

$$F(\text{lb}) = K_1 + K_2 AV^2_M \quad , \tag{B-6}$$

where:

V_M = maximum vehicle speed = mph,

K_1 = 20 pounds per 1000 pounds of vehicle weight,

K_2 = varies between 0.001 and 0.002, and

A = frontal area, ft^2.

Multiplying the retarding force by wheel radius, R_W, gives the resistance torque on the rear axle, thus

$$\left(K_1 + K_2 AV^2_M \right) R_W = \frac{R_G}{r} \frac{PV_T}{24\pi} D_G \quad . \tag{B-6}$$

Solving for V_M gives

$$V_M = \sqrt{\frac{D_G}{24\pi} \frac{PV_T}{G} \frac{1}{K_2 A} - \frac{K_1}{K_2 A}} \quad . \tag{B-7}$$

APPENDIX C
A TABULATION OF TYPICAL DATA

TABLE C-1. TYPICAL VALUES FOR GUFFIN DOUR NUMBERS AND MASS RATIO CUTOFF

D_G \ X_M	Ratio of Clearance to Release Volume (V_C/V_T)			
Geometric Cutoff (Percent)	0.020	0.040	0.080	0.160
0		0.047 / 0.024	0.119 / 0.066	0.214 / 0.147
5	0.136 / 0.054	0.173 / 0.075	0.224 / 0.114	0.291 / 0.192
10	0.250 / 0.106	0.278 / 0.125	0.315 / 0.163	0.360 / 0.236
20	0.433 / 0.209	0.450 / 0.226	0.466 / 0.260	0.478 / 0.325
40	0.686 / 0.415	0.689 / 0.428	0.681 / 0.454	0.650 / 0.503

TABLE C-2. VALUES OF SPECIFIC VOLUME AND ENTHALPHY FOR WATER

Each cell lists specific volume v_0 (upper value) and enthalpy h (lower value).

Supply Temperature (°F)	\	Supply Pressure – psia						
	v_0 \ h	2000	1700	1400	1100	800	500	200
700	v_0	0.249	0.315	0.406	0.544	0.783	1.30	3.38
	h	1240	1269	1295	1318	1338	1357	1373
900	v_0	0.352	0.425	0.528	0.686	0.963	1.57	4.00
	h	1409	1421	1433	1444	1455	1466	1476
1100	v_0	0.431	0.5132	0.630	0.811	1.12	1.82	4.60
	h	1536	1542	1551	1558	1566	1573	1580
1300	v_0	0.501	0.594	0.725	0.930	1.29	2.07	5.27
	h	1655	1660	1666	1671	1677	1683	1688
1500	v_0	0.568	0.671	0.818	1.05	1.44	2.31	5.82
	h	1773	1777	1781	1786	1790	1795	1799

APPENDIX D

DEVELOPMENT OF THE THOMPSON EQUALITY

In consideration of the relationship between laboratory measured efficiency and miles per gallon, several questions arise concerning any additional information that may be available as a result of the test. The author knows of no technique where a single value of efficiency can be related to miles per gallon without having additional information. This fact will be obvious as the following relationship is developed.

Laboratory measured efficiency is taken to be the ratio of the shaft power output (hp) of the engine to the equivalent heat power input (Q),

$$\eta_S = \frac{hp}{Q} \quad . \tag{D-1}$$

The fidelity of equation (D-1) can be obtained by expanding the constituents of Q,

$$\eta_S = 2545 \; \frac{Btu/hr}{horsepower} \; \frac{hp}{EF_R \rho} \quad , \tag{D-2}$$

where:

E = heat of combustion = Btu/lb

F_R = fuel flow = gal/hr, and

ρ = fuel density = lb/gal.

Under the assumption of zero transmission losses, all of the developed horsepower is consumed by the work rate to overcome rolling road resistance and aerodynamics. The measured hp is capable of being converted to an equivalent force, F, at the wheels (Reference 1, Appendix C), which results in vehicle velocity, V,

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$$hp = FV = \frac{88 \frac{ft}{min} \frac{hr}{mile}}{33000 \frac{ft\text{-}lb/min}{horsepower}} (K_1 + K_2 AV^2) V \quad . \qquad (D\text{-}3)$$

Substituting (D-3) into (D-2) gives

$$\eta_S = 6.78 \frac{(K_1 + K_2 AV^2) V}{E\rho F_R} \qquad (D\text{-}4)$$

Noting that $V/F_R = mpg$, then

$$\eta_S = 6.78 \frac{(K_1 + K_2 AV^2)}{E\rho} mpg \quad . \qquad (D\text{-}5)$$

On the test bench, the velocity capability of the vehicle is unknown, but can be related to engine operating and design conditions through the Abatement Number and Guffin Dour Number. Carrying through with the derivation used in Appendix C of Reference 1 and substituting for the vehicle speed, V, in equation (D-5), gives,

$$mpg = 11.1 \frac{\eta_S E\rho}{\left(\frac{PV_T}{G}\right) D_G} \quad . \qquad (D\text{-}6)$$

Equation (D-6) is referred to as the Thompson Equality. It relates to a measured engine efficiency, as defined in equation (D-1), to miles per gallon. It is important to recognize that the release volume, V_T, Guffin Dour Number, and supply pressure are not independent variables. For a measured efficiency, the values of P, V_T and D_G sustained during the test must be the values used to compute the resulting miles per gallon. It is improper to indiscriminately substitute any selected value in the Thompson Equality.

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Gear ratio, G , is an independent variable, and increasing this parameter will increase mpg for a given measured efficiency. Increasing G is analogous to overdrive.

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