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ENGINE CYCLE ANALYSIS USING AN IDEAL GAS WITH CONSTANT SPECIFIC HEATS AS A WORKING FLUID

Edwin N. Quiros, Ph.D.
Assistant Professor
Department of Mechanical Engineering
College of Engineering
University of the Philippines
Diliman, Quezon City, Philippines

ABSTRACT

This work presents the results of computer modelling of the basic and idealized engine cycles assuming the working fluid to be an ideal gas with constant specific heats. The various processes that constitute the engine cycles are assumed to be ideal. A thermodynamic analysis of the engine cycles is performed to calculate performance parameters. Parametric studies using the computer model was conducted to determine desired engine design and operating conditions.

INTRODUCTION

The usefulness of internal combustion engines can be maximized if they are designed and operated under optimum conditions. Consequently, the question "What engine design and operating conditions will yield optimum or desired engine performance:" has to be answered. This work presents some general answers based on basic and idealized theoretical considerations. Because of the assumptions used in the analysis, the usefulness of the results lies more in the qualitative indications or trends they provide rather than the quantitative values generated.

The parameters which will be used to quantify engine performance in this study are limited to the following:

1. **n** fuel conversion (or indicated thermal) efficiency
2. **imep/p_1** indicated mean effective pressure (normalized by the pressure at the start of compression)
3. **imep/p_3** the ratio of the indicated mean effective pressure to the maximum engine pressure.

The first parameter is a measure of how much of the chemical energy from the fuel is converted into work. The higher it is, the better. The indicated mean effective pressure is an equivalent "average pressure" (of the working fluid) acting on the pistons throughout the engine displacement to produce the indicated work or power. It may be thought of as a measure of how weil the engine displacement is used to produce power. A high mean effective pressure is desired. The third parameter is an indication of engine weight. A high value of this parameter means lower engine weight and thus is desirable.

The engine cycle models used together with the corresponding thermodynamic analysis will examine how these parameters vary with other engine design and operating variables to identify optimum conditions.

ENGINE CYCLE MODEL

Figure 1 is a pressure-volume diagram of the various idealized engine cycles considered in this study. These engine cycles are also known as **ideal gas standard cycles**. In the figure, engine operation is unthrottled for cycles (a) constant-volume combustion, (b) constant-pressure combustion, and (c) limited-pressure combustion. That for (d) is the throttled constant-volume cycle while (e) is the supercharged constant-volume cycle.

The following are the descriptions and corresponding assumptions regarding the processes that make up the cycles shown.

Process	**Assumptions**
Compression (1 - 2)	Adiabatic and reversible (hence isentropic)
Combustion (2 - 3)	1. Adiabatic 2. Combustion occurs at (a) constant volume (b) constant pressure (c) part at constant volume and part at constant pressure (called limited-pressure) 3. Combustion is complete, i.e. combustion efficiency is 100%
Expansion (3 - 4)	Adiabatic and reversible (thus isentropic)
Exhaust (4 - 5 - 6) and Intake (6- 7 - 1)	1. Adiabatic 2. Valve events occur at top- and bottom-center 3. No change in cylinder volume as pressure differences across open valves drop to zero 4. Inlet and exhaust pressures constant 5. Velocity effects negligible

Throttled engine operation means that the inlet pressure **pi** is less than the exit pressure **pe**. For supercharged (or turbocharged) engine operation, **pi > pe.**

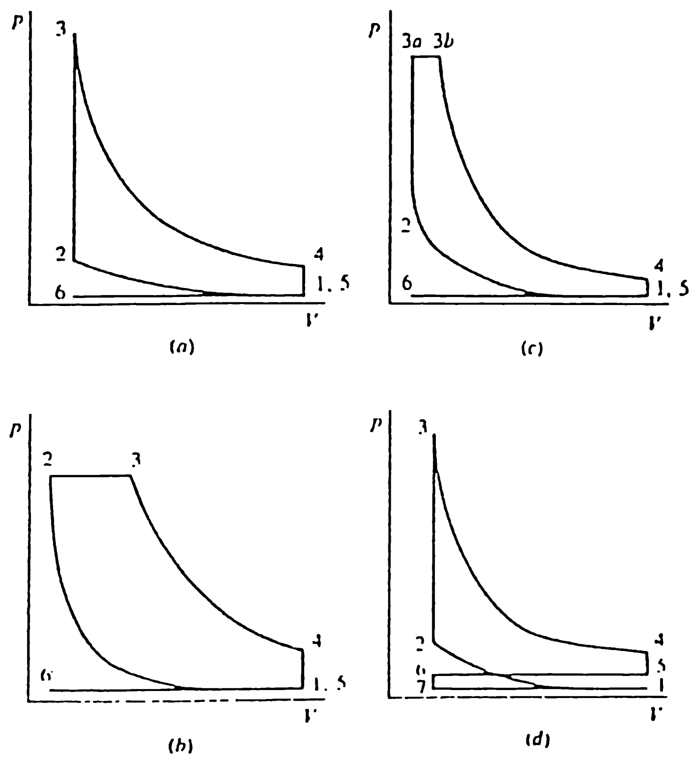

Figure 1

Pressure-volume diagrams of ideal cycles. Unthrottled operation: (a) constant-volume combustion; (b) constant-pressure combustion; (c) limited-pressure combustion; (d) Throttled constant-volume cycle; (e) supercharged constant-volume cycle.

The assumptions made on the combustion process determines to a great extent the usefulness of these ideal cycles as indicators of engine performance. In the real engine, combustion occurs over a period between 20 and 70 crank angle degrees. Thus, the constant-volume cycle is the limiting case of infinitely fast combustion at top dead center (TDC); the constant-pressure cycle corrsponds to slow and late combustion; the limited-pressure cycle is somewhere between.

Thermodynamic Analysis

A thermodynamic analysis of the various processes is next presented. The working fluid is assumed to be an ideal gas with constant specific heats. In the analysis, the properties at the different end states of the cycles are determined. The work and heat transfer for each of the processes are likewise calculated as necessary. The performance parameters !!SYMBOL 104 \f "Symbol"§, imep/p1, and imep/p3 are then determined.

Intake Stroke

The intake stroke begins with state 7 and proceeds at constant pressure until state 1 is reached. In the case of unthrottled (i.e., wide-open throttle) and without supercharging operations (see Figure 5-2 a, b, & c), states 6 and 7 coincide. The properties at state 7 correspond to the inlet state denoted by the subscript i. Thus

p7 = pi	pi = inlet (or intake) pressure
T7 = Ti	Ti = inlet (or intake) temperature
v7 = vi	vi = inlet specific volume
V7 = VC	VC = clearance volume

For throttled and wide-open throttle without supercharging operations, residual gas will be present in the cylinder. The amount of this residual gas that mixes with the intake charge affects the properties at state 1 (the end of the intake stroke and the start of the compression stroke). There is no residual gas with supercharging.

The amount of residual gas in the cylinder is expressed by the **residual gas mass fraction xr** defined as

$$xr \equiv mr / m \qquad (1)$$

For the constant-volume cycle, this becomes [1]

$$xr = (1/rc).(pe/pi)^{1/\Phi}/[1+Q^*/(CvT1.rc^{\Phi-1})]^{1/\Phi}\} \qquad (2)$$

The temperature at the end of the intake stroke (also the beginning of the compression stroke) is then

$$T1 = (1-xr)Ti/\{1-[pe/pi+(\gamma-1)]/(\gamma xrc)\} \tag{3}$$

and the residual gas temperature is

$$Tr = (pe/pi)^{1/\gamma)/\gamma}.[1+Q^*/(CvT1.rc^{1-\gamma})]^{1/\gamma}.T1 \tag{4}$$

In the above expression, Q* is defined as

$$Q^* = mf\ QLHV/m \tag{5}$$

and may be thought of as the heat release (from fuel combustion) per unit mass of the working fluid. Alternative expressions for Q* can be developed as follows.

$$Q^* = mfQLHV/m = (mf/mi)\ (mi/m)\ QLHV \tag{6}$$

But

$$m = mi + mr \tag{7}$$

Combining eq. (1) and eq. (7) gives

$$mi\ /\ m = (1 - xr) \tag{8}$$

Assuming there is no exhaust gas recirculation (EGR), then

$$mi = mf + ma \tag{9}$$

so that

$$mf/mi = mf/(mf+ma) = 1/[1+(A/F)] = 1/[1+(A/F)s/\phi]$$

or

$$mf/mi=\phi/[\phi+(A/F)s] \tag{10}$$

Combining eqs. (6), (8), & (10),

$$Q^* = [\phi/[\phi+(A/F)s].(1 - xr).QLHV \tag{11}$$

Another way to calculate Q^* is to express it as

$$Q^* = mf.QLHV/m = (mf/ma)(ma/m)QLHV \tag{12}$$

A common approximation to ma/m is to assume that fresh air fills the displaced volume and the residual gas fills the clearance volume at the same density [2]. Then

$$ma/m \approx (rc-1)/rc$$

so that eq. (12) becomes

$$Q^* = [\phi/(A/F)s] [(rc-1)/rc] QLHV \tag{13}$$

Equations (11) or (13) may be more convenient to use since Q^* is expressed in terms of the more common and easily known parameters. Note however that $\phi \leq 1.0$ in either equation since only for such can the assumption that combustion efficiency (theoretically) is 100% be consistent. The upper limiting value for Q^* in terms of ϕ is at $\phi = 1.0$.

To start the cycle calculations, the parameters Ti, Pi, Pe, ϕ, QLHV, rc, Φ, (A/F)s, and Cv are specified. An assumed value of xr is then made (xr = 0.03 may be a good start). T1 is then calculated from eq. (3) and Q* from either eq. (11) or (13). Eq. (12) is next used to calculate a new value of xr which is compared with the assumed value. When the assumed and the calculated values of xr are not yet close enough, the calculated value of xr becomes the new assumed value of xr and new values for T1 and Q* are computed using eqs. (3) and (11) or (13) respectively. The iteration stops when the assumed and calculated values of xr are close enough. Then, xr from eq. (2) and T1 from eq. (3) are taken as the final values. Tr is then calculated from eq. (4).

The following variables for state 1 are computed as follows after the final values for xr, Tr, and T1 are determined:

$$p1 = pi$$

$$v1 = R \cdot T1/p1$$

$$V1 = rc.Vd/(rc-1)$$

$$m = p1.\ V1/(R.T1)$$

where

$$R = (\gamma - 1)\ Cv$$

Vd = any arbitrary engine displacement; used here to facilitate some calculations

Compression Stroke

The properties at state 2 are calculated as follows.

$$p2 = rc^{\gamma} \cdot p1$$

$$T2 = rc^{\gamma-1} \cdot T1$$

$$v2 = R \cdot T2/p2$$

$$V2 = VC$$

where

$$VC = \text{clearance volume} = Vd/\ (rc - 1)$$

Also,

$$Wc = W12 = Cv\ (T1 - T2) \qquad (14)$$

Constant-Volume Combustion Process

The computation for the properties at state 3 are shown below. The notations used are for the constant-volume cycle.

$$V3 = V2$$

$$v3 = v2$$

$$T3 = Q^{*}/Cv + T2$$

$$p3 = R \cdot T3/v3$$

Expansion Stroke

For the expansion process,

$$V4 = V1$$

$$p4 = p3 / rc^{\gamma}$$

$$T4 = T3 / rc^{\gamma-1}$$

$$v4 = R . T4/p4$$

and

$$WE = W34 = Cv\ (T3 - T4) \tag{15}$$

Blowdown Process

The working fluid remaining in the cylinder is assumed to expand isentropically from state 4 to p5. Thus,

$$p5 = pe$$

$$V5 = V4$$

$$v5 = v4\ (p4 / p5)^{1/\gamma}$$

$$T5 = p5 . v5 / R$$

Exhaust Process

At the end of the exhaust stroke,

$$p6 = pe$$

$$V6 = V2$$

$$v6 = v5$$

$$T6 = T5$$

The differences among the constant-volume, constant-pressure, and limited-pressure cycles lie only in the combustion process. These result into the working fluid properties at the end of their respective combustion processes being calculated in differing manner. The following paragraphs show how these properties are determined.

Limited-Pressure Cycle Combustion Process

The combustion process for this cycle is partly at constant-volume and partly at constant-pressure, refer to Fig. 1(c) for notation. To define the combustion process in this case, a maximum value of p3 is specified and expressed as

$$Pmax = p3 = P3a = C3a.p1$$

where

$$C3a = \text{some multiple of } p1$$

This definition of the combustion process is justified by the fact that the maximum cylinder pressure is a limiting factor in the design of the engine structure and components, among other things.

For the **constant-volume combustion part**, the properties at state 3a are then

$$V3a = V2$$

$$v3a = v2$$

$$fcv = (P3a \,.\, v3a/R - T2)\ Cv\ /\ Q^*$$

$$T3a = fcv \,.\, Q^*/Cv + T2$$

where

fcv = fraction of total fuel burned at constant-volume = mfcv / mf

mf = mfcv + mfcp

mfcv = mass of fuel burned at constant-volume

mfcp = mass of fuel burned at constant-pressure

For the **constant-pressure combustion part**, the properties at state 3b are

$$P3b = P3a$$

$$V3b = (T3b / T3a) V3a$$

$$Cp = R + Cv$$

$$T3b = (1 - fcv) Q^{*}/Cp + T3a$$

$$v3b = R . T3b / P3b$$

and

$$W3ab = (1-fcv) Q^{*} - Cv (T3b-T3a) \tag{16}$$

The other properties for the other state points in the limited-pressure cycle are calculated similarly to those which have been previously derived in the case of the constant-volume cycle.

Constant-Pressure Cycle Combustion Process

At the end of the combustion process for this cycle,

$$p3 = p2$$

$$V3 = (T3 / T2) V2$$

$$T3 = Q* / Cp + T2$$

$$v3 = R . T3 / p3$$

and

$$W23 = Q^{*} - Cv (T3 - T2) \tag{17}$$

Performance Parameters

The performance parameters of interest, namely **n**, **imep/p1**, and **imep/p3** were calculated in the following manner.

Fuel Conversion Efficiency n

The gross indicated work for the various cycles are

$$Wi,g = Wc + WE \qquad (\text{constant-volume cycle})$$

$$Wi,g = Wc + W23 + WE \quad (\text{ constant-pressure cycle }) \quad (18)$$

$$Wi,g = Wc + W3ab + WE \quad (\text{ limited-pressure cycle })$$

The fuel conversion efficiency is given by

$$n = Wi,g \, / \, Q^{*} \quad (19)$$

In the case of the constant-volume cycle, it can be shown that the fuel conversion efficiency is a function only of rc and Φ as

$$n = 1 - 1 \, / \, rc^{\gamma - 1} \quad (20)$$

The fuel conversion efficiency of the limited-pressure cyle can also be shown to be

$$n = 1-(1/rc^{\gamma-1}).\{(\alpha\beta \;\; - 1)/[\alpha\gamma(\beta-1)+\alpha \; - 1]\} \quad (21)$$

where

$$\alpha = p3/p2 \qquad\qquad \beta = V3b/V3a$$

Note that when $\beta=1$, n is for the constant-volume cycle; when $\alpha=1$, n is for the constant-pressure cycle.

Mean Effective Pressure

The mean effective pressure is normalized both by the pressure at the end of the intake stroke pi and the maximum cylinder pressure p3. This performance parameter is therefore expressed as

$$imep/p1 = m \, . \, Wi,g \, / \, (\, Vd \, . \, p1) \quad (22)$$

and

$$imep/p3 = m \, . \, Wi,g \, / \, (Vd \, . \, p3) \quad (23)$$

A computer program for each of the engine cycles above was written to determine the variation of engine performance parameters as a function of some engine design and operating variables.

MODELLING RESULTS

Figure 2 shows the variation of the fuel conversion (or thermal) efficiency η with compression ratio r_c and γ of the working fluid for the constant-volume cycle. The graph shows that efficiency increases with increasing compression ratio. The improvement in efficiency decreases however for rc beyond about 10.

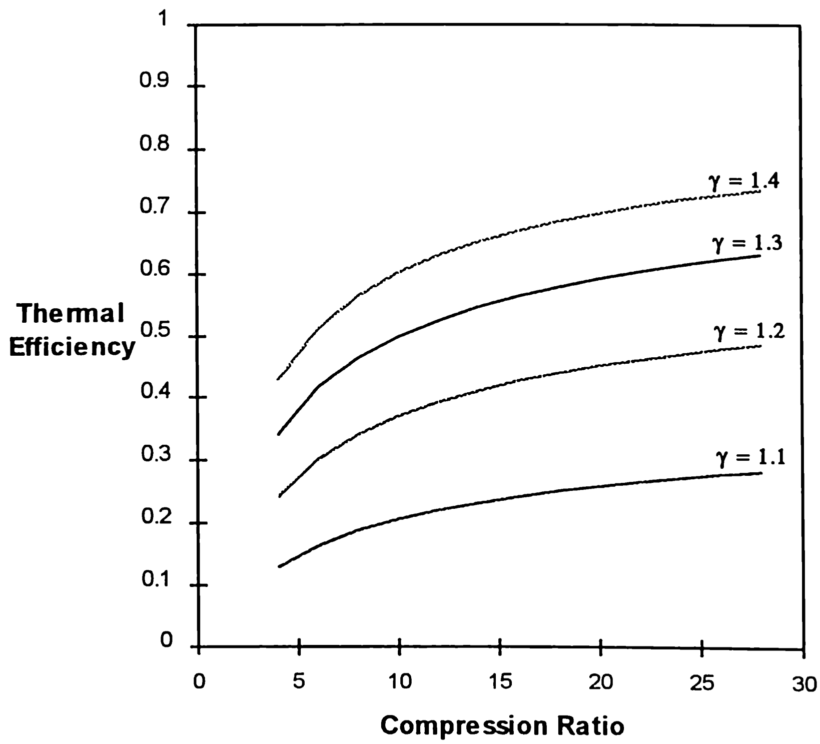

Figure 2
Variation of thermal efficiency with compression ratio for the ideal gas constant-volume combustion cycle; $\gamma = C_p/C_v$, $p_e/p_i = 1.0$

Also, the efficiency increases with increasing γ for a given compression ratio. Note that $\gamma = 1.4$ is the value for air alone while $\gamma = 1.3$ for real combustion gases. This indicates that efficiency is improved for a given engine the leaner it is operated. There is however a lean limit for which the fuel-air mixture will ignite and this dictates how lean an engine can operate.

Figure 3 shows fuel conversion (or thermal) efficiency as a function of compression ratio for the constant-volume, limited-pressure, and constant-pressure engine cycles. In this figure, ϕ = 1.0. pe/pi = 1.0, and = 1.3 for all the cycles. The curves between the constant-volume and constant-pressure curves are for limited-pressure cycles with various ratios of p3/p1 (33, 67, & 100) as indicated.

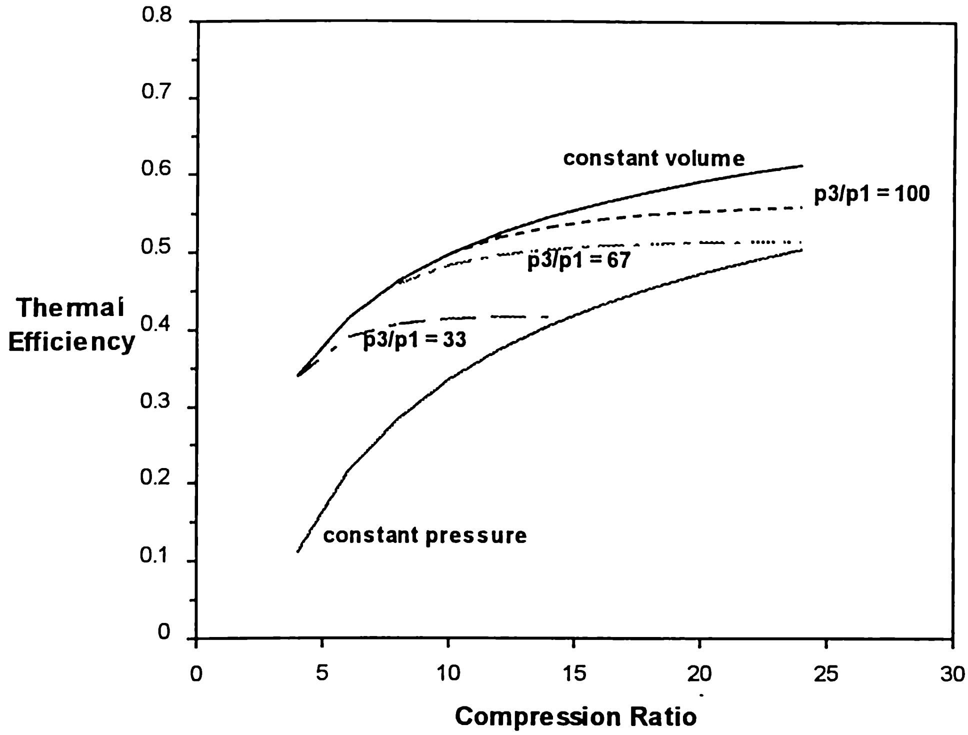

Figure 3
Variation of thermal efficiency with compression ratio for the constant-volume, constant-pressure, and limited-pressure ideal gas cycles; γ = 1.3, ϕ = 1.0, p_e/p_i = 1.0

The graph shows that for a given engine compression ratio, constant-volume cycle operation will yield the highest efficiency. In practice, this means that efficiency is highest if combustion can be completed while the piston remains very near top dead center.

On the other hand, for the same maximum cylinder pressure, constant-pressure cycle operation will yield the highest efficiency but will require the highest engine compression ratio.

Note also that in the case of the limited-pressure cycle, at a given p_3/p_1, efficiency improves very little as rc is increased to above a value of around 8 to 10.

Figure 4 shows the variation of **mep/pi** with compression ratios for the different engine cycles. The shape of the curves essentially follows that for **n** vs. **rc**. For a given compression ratio, the constant-volume cycle gives the highest mean effective pressure while the constant-pressure cycle the lowest.

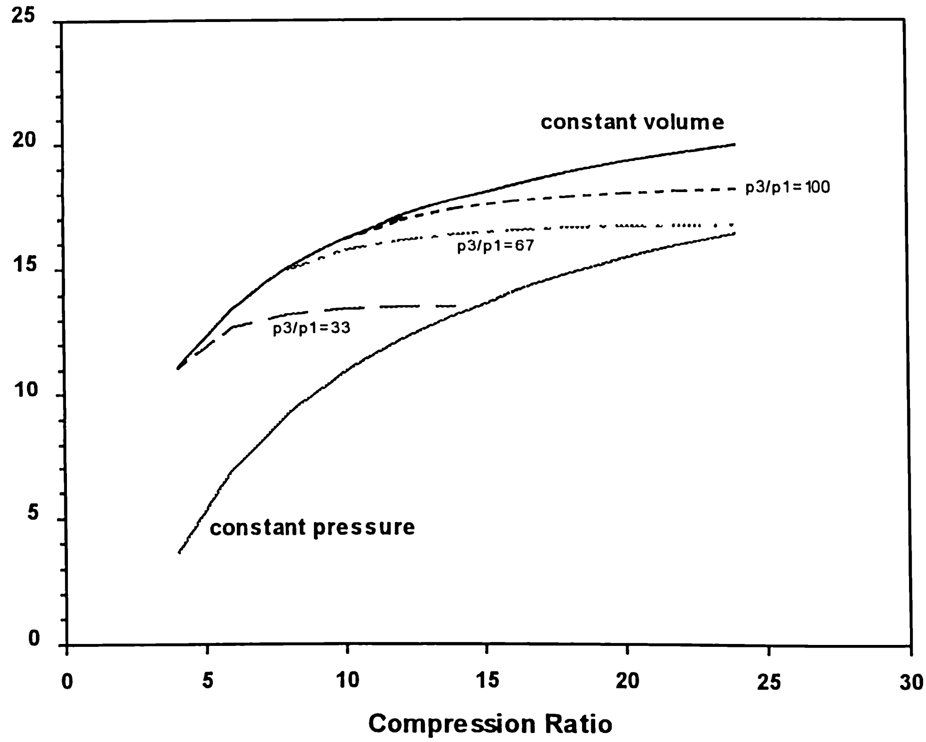

Figure 4
Variation of mep/p_i with compression ratio for the constant-volume, constant-pressure, and limited-pressure ideal gas cycles; γ = 1.3, ϕ = 1.0, p_e/p_i = 1.0

Figure 5 shows that for a given compression ratio, the constant-volume cycle has the lowest **mep/p3** while the constant-pressure cycle the highest. This is due to the fact that under this condition, maximum pressure will be highest under constant-volume combustion

Engine operation is also affected by throttling and supercharging. Throttling and supercharging are quantified in this analysis by the ratio of the exhaust and inlet pressures, **pe/pi**. For throttling, **pe/pi > 1**; for supercharging, **pe/pi < 1**, and **pe/pi = 1.0** for wide-open throttle (wot) operation. Figure 6(a)-6(d) shows the effect of pe/pi on various engine operating and performance parameters.

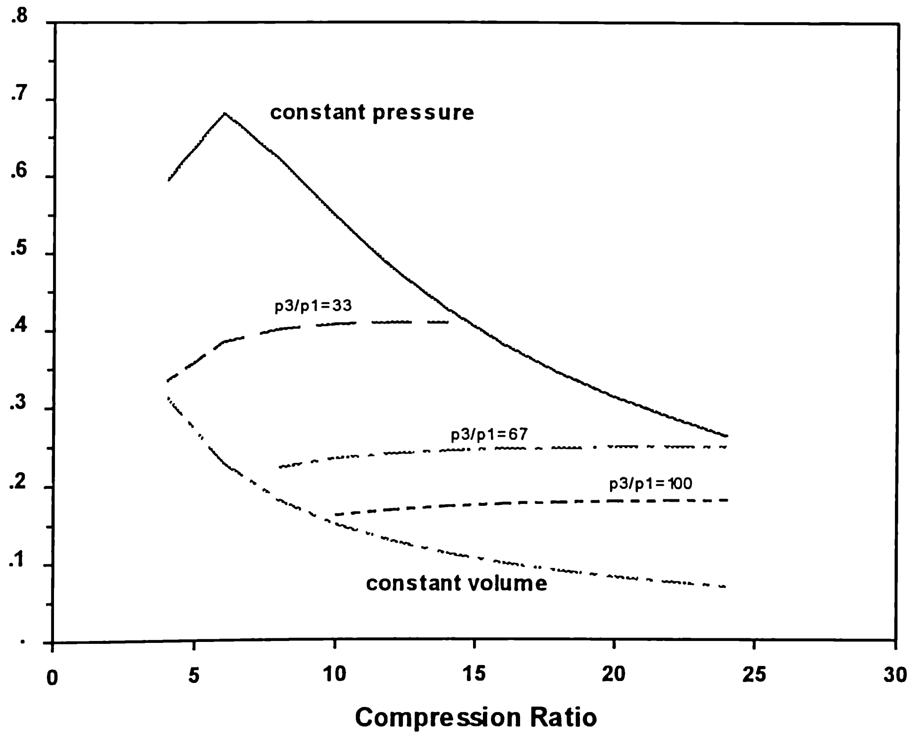

Figure 5

At a given compression ratio, the residual gas mass fraction **xr** increases as throttling is increased, see Figure 6(a). The residual gas decreases the more supercharged the engine is. The difference in the residual gas mass between supercharged and throttled engine operations decreases at higher compression ratios. This is due to the decrease in the clearance volume **VC** percentage relative to the displaced volume **Vd**.

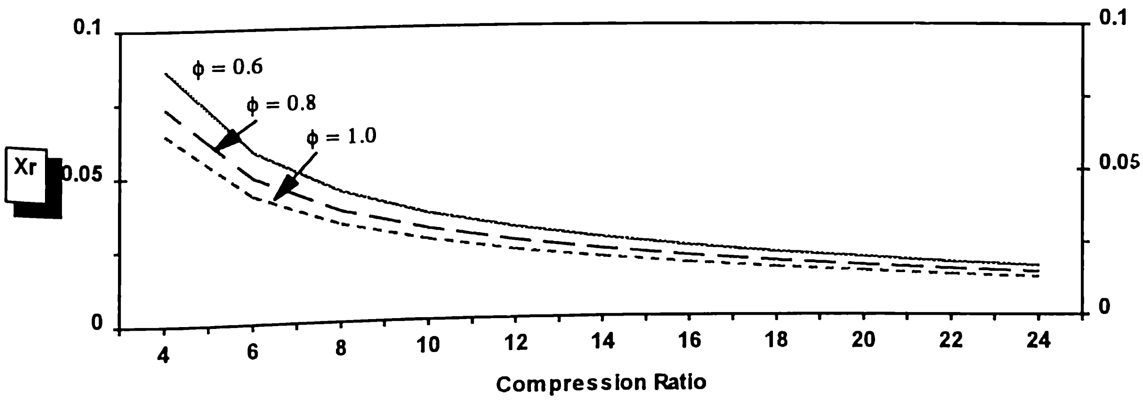

Figure 6(a)

Supercharging increases while throttling decreases the **mep/p1** for all **rc** as mep/p1 increases with rc, see Figure 6(b). On the other hand, mep/p3 decreases with increasing rc. The difference in mep/p3 between supercharging and throttling is not significant.

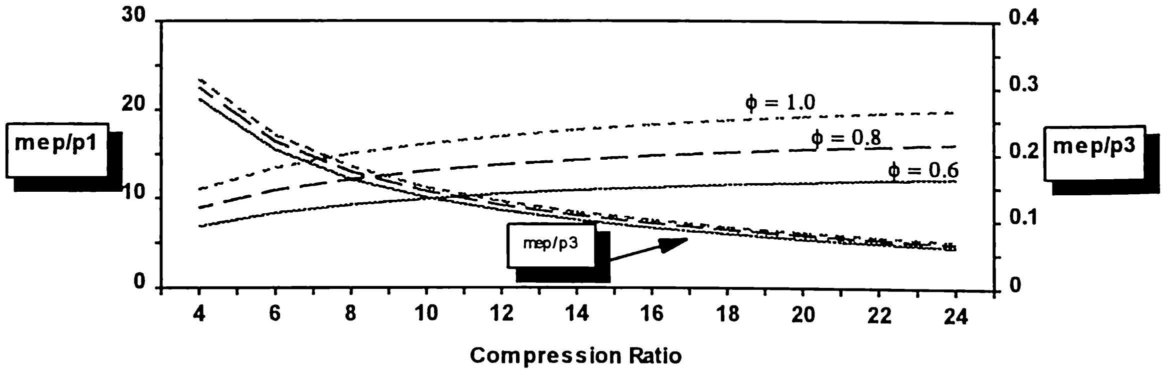

Figure 6(b)

The residual gas temperature **Tr** also increases with increased throttling and decreas with supercharging, see Figure 6(c). Higher compression ratios tend to lower Tr for the san reason as with **xr**.

The above-mentioned behavior of **xr** and **Tr** at a given compression ratio result into a similar behavior for **T1**. However, throttling and supercharging have a significant effect on **T1** only at low compression ratios.

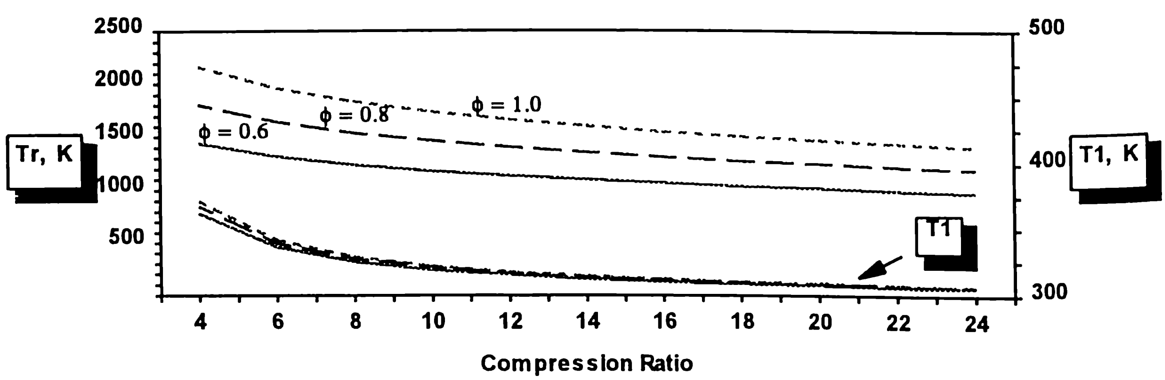

Figure 6(c)

The mass **m** of the working fluid inside the cylinder increases with supercharging and decreases with throttling, see Figure 6(d). The variation for **m** with **pe/pi** becomes less at higher compression ratios. This is due to the decrease in **VC** relative to **Vd**.

The effect of lesser T1 with supercharging is more dominant in making the heat release per unit mass of the working fluid (normalized by CvT1) **Q*/CvT1** higher with supercharging.

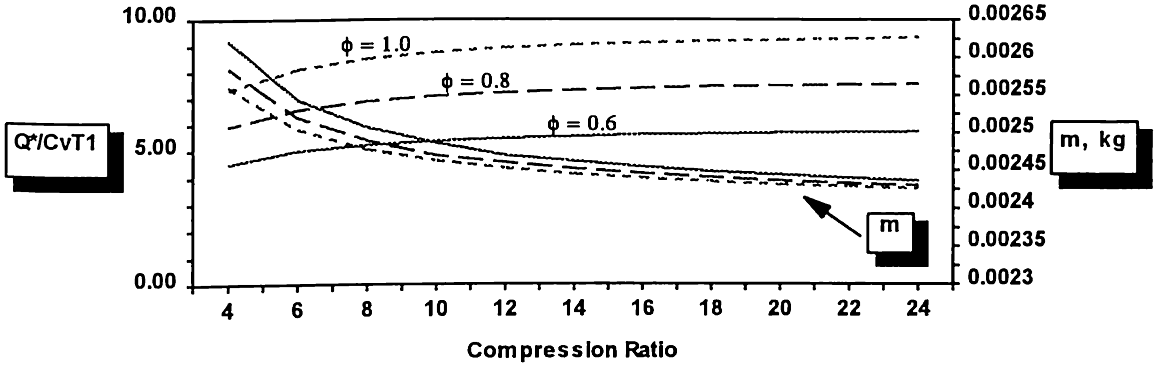

Figure 6(d)

Figure 7 shows the effect of the equivalence ratio ϕ on various engine operating and performance parameters. Higher values of ϕ mean increased fuel supplied and burned (or higher engine load). At a given compression ratio therefore, this translates to a higher Q*/Cv T1. Both Tr and T1 increase with ϕ; xr and m decrease with increasing ϕ. This is due to higher burned gas temperatures at higher ϕ.

For a given rc, the mep/p_1 increases with ϕ (more fuel burned results to more work done) while mep/p_3 decreases with increasing ϕ (more fuel burned means higher maximum pressure).

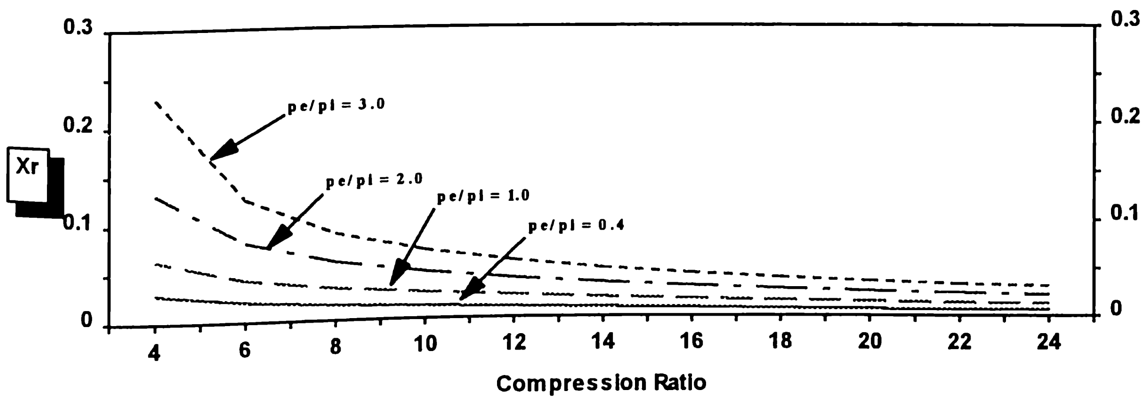

Figure 7(a)

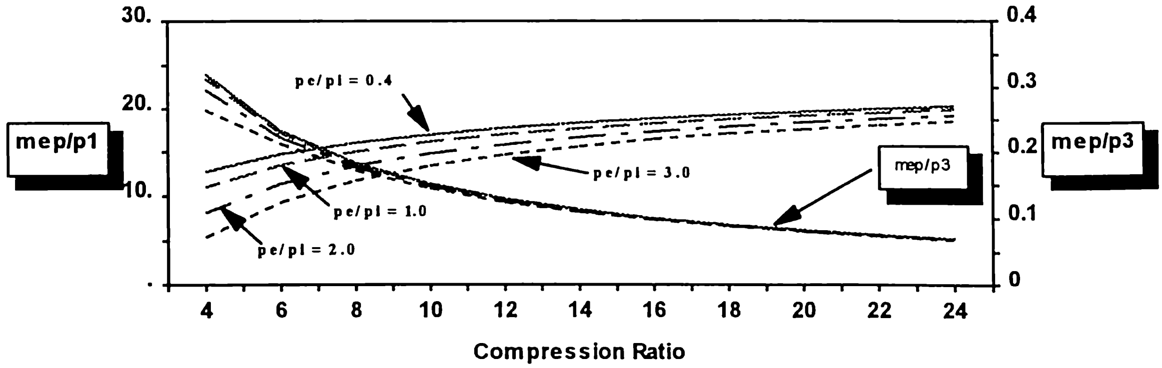

Figure 7(b)

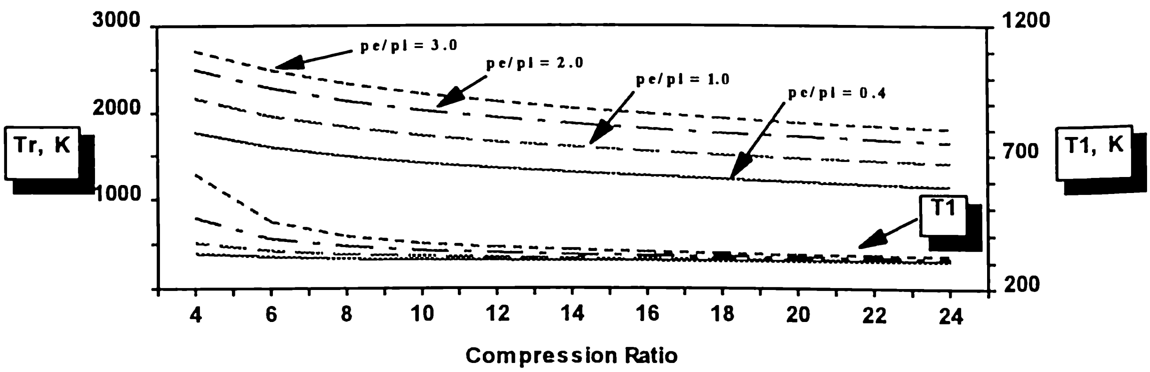

Figure 7(c)

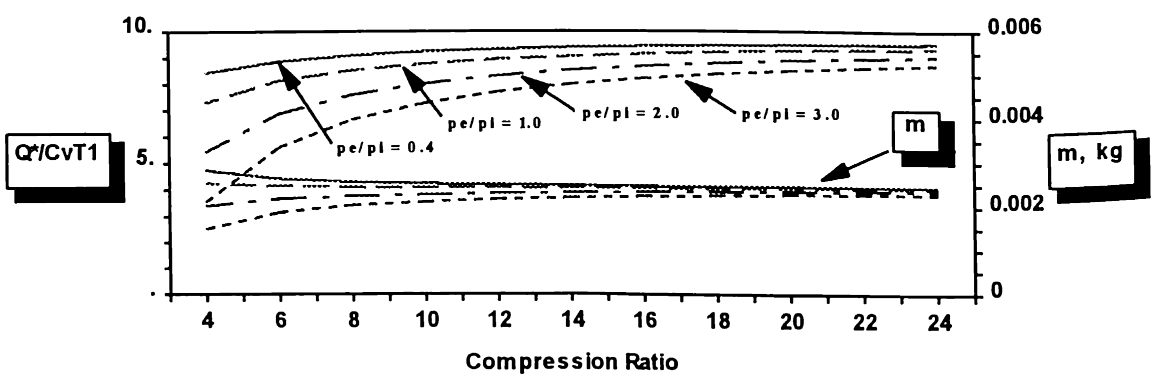

Figure 7(d)

CONCLUSION

A computer model of the basic and idealized engine cycles was created assuming an ideal gas with constant specific heats as the working fluid. Thermodynamic analysis of the various processes in the cycles was done to calculate different engine operating and performance parameters. Parametric studies were done to determine desired engine design and operating variables that will produce desired performance.

Results of the studies indicate that:

1. higher fuel conversion efficiencies can be obtained with higher compression ratios and/or lean mixtures

2. highest fuel conversion efficiency, mean effective pressure, and maximum pressure can be achieved with constant-volume combustion for a given engine compression ratio

3. for the same maximum cylinder pressure, constant-pressure combustion will yield the highest efficiency but will require the highest engine compression ratio

4. supercharging increases mean effective pressure but also raises the maximum cylinder pressure (and therefore temperature) for a given compression ratio

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