THE COMPUTER MODELING OF A MORE-COMPLETE-EXPANSION CYCLE WITH ITS APPLICATION TO AN IRRIGATION ENGINE

by

GEORGE ROSS EAKIN

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Approved by:

Donald L. Fenton

Major Professor:

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		Units
A/F ratio	: Air/fuel Ratio	
ABDC	: After Bottom Dead Center	deg
BBDC	: Before Bottom Dead Center	deg
BSFC	: Brake Spec. Fuel Consump.	g/kW-hr
BTDC	: Before Top Dead Center	deg
CD	: Drag Coefficient	
CP	: Constant Pressure Specific Heat	kJ/kmol-K
CR	: Compression Ratio	
D	: Valve diameter	cm
EFF	: Brake Thermal Efficiency	%
FMEP	: Frictional Mean Effective Pressure	kPa
FF	: Fuel Flow	kg/hr
H	: Instantaneous Heat Transfer Coefficient	kW/m^2-K
kW	: Power Output	kW
L	: Valve Lift	cm
MEP	: Mean Effective Pressure	kPa
MPS	: Mean Piston Speed	m/s
PH	: Peak Cylinder Pressure	mPa
P	: Instantaneous Pressure	kPa
RPM	: Engine Speed	rev/min
R	: Universal Gas Constant	kJ/kmol-K
SA	: Spark Advance	deg
S	: Stroke	m
TM	: Intake Manifold Temperature	K
T	: Instantaneous Temperature	K
VE	: Volumetric Efficiency	%
V	: Instantaneous Volume	m^3
VIC	: Volume at Intake Valve Closing	m^3

ABSTRACT

The more-complete-expansion cycle can be applied to a stationary irrigation engine commonly operated at constant partial-load burning liquid fuels. The analysis involved the development of a digital computer model to simulate a real engine.

The Chevrolet 5.74 liter (350 cu. in.) engine was simulated to determine the performance changes and possible advantages of applying the more-complete-expansion cycle features. The fuel used in this model was methane gas. The generated performance data is presented in figures. The modified engine specifications are results of these figures and are used to determine the cost effectiveness of the concept.

The performance results in conjunction with a cost analysis suggest that a fuel saving greater than 11% can be achieved by utilizing the more-complete-expansion cycle. If the engine is operated frequently, the amount of money saved in fuel is significant and could pay off the cost of the engine modification in less than a year. Therefore, this idea is recommended for further development and testing.

1.0 INTRODUCTION

The problem exists that some agricultural locations along the "Ogallala Aquifer" use ordinary automobile or tractor spark ignition engines to run irrigation pumps. These engines generally burn natural gas (about 95% methane) and operate at part-throttle, using only a fraction of the useful displacement volume while consuming large amounts of fuel. It is proposed that the engine can be modified in such a way to incorporate a more-complete-expansion process. This process involves the use of an expansion ratio greater than the effective compression ratio. The addition of the more-complete-expansion process tends to use the entire displacement volume more efficiently, thus decreasing the fuel consumption.

The purpose of this research is to determine the merit of implementing such a process in a stationary internal combustion engine. The engine will be a Chevrolet 5.74 liter V-8 (350 cu. in.). A computer program will be used to simulate the part-throttle reference case and the modified cases to determine the advantages, if any.

2.0 LITERATURE REVIEW

The more-complete-expansion cycle known as the Atkinson Cycle was named after Atkinson who first proposed the idea in 1885. This cycle has not been researched or used significantly in the past. Only three references found mention any work or use of such a cycle. The reason for this lack of interest is due to the fact that the more-complete-expansion cycle is most effective when the engine is operated under a constant heavy load. Another disadvantage of utilizing this cycle is the size of the engine required. For this cycle to be successful, a large displacement volume is needed to compensate for the lower mean effective pressure which results.

The first reference is a book entitled, "The Internal Combustion Engine in Theory and Practice", Vol. II (p.402), by Charles Fayette Taylor. This reference includes an ideal theoretical example of designing a large gas engine to operate a natural gas compressor unit. The purpose of using the more-complete-expansion cycle is three-fold. First, the compressor station requires great reliability, second, long life and third, good fuel economy. All these characteristics can be achieved by utilizing the more-complete-expansion cycle.

The second reference is a technical paper entitled, "Improved Engine Efficiency with Emphasis on Expansion Ratios", by Rodney Herbert and Brian Lanoway. The paper discusses ways in which the expansion ratio effects the efficiency of an engine. It mentions the use of a modified Atkinson Cycle where the intake valve is closed late. This allows some of the freshly inducted charge to be forced back into the intake manifold which effectively reduces the compression ratio.

The third is a book entitled, "Combustion Engine Processes", by Lester C. Lichty. The information included in this reference deals with the ideal advantages of applying the more-complete-expansion cycle.

Late valve timing has been examined by Miller with the Cadillac and Nordberg engines and lately by SAAB and Edward LaForce. (Herbert, Lanoway, 1979, p15) LaForce claimed that his engine operating on the modified Atkinson Cycle demonstrated a 25% increase in fuel economy, however the American EPA tested the engine and found it to give better fuel economy but only at the expense of reduced power output and increased emissions. The paper also suggests that SAAB found a slight increase in fuel economy but feels the cost of production is unwarranted.

Not having found any useful references through published literature or the library computerized search in regards to the more-complete-expansion cycle, this study will develop a computer simulation of a more-complete-expansion cycle.

3.0 PROGRAM DEVELOPMENT

3.1 Description:

The computer program listed in Appendix III used in this research was developed largely from the discussion in Ashley S. Campbell's book entitled, "Thermodynamic Analysis of Combustion Engines". The program simulates the processes of an actual engine and generates performance information. Some of the information produced by the program includes the spark advance, mean piston speed, mean effective pressure, brake power output, brake specific fuel consumption, volumetric efficiency, pressure before the compression stroke, peak cylinder pressure and brake thermal efficiency.

The equations used to calculate the performance information are shown below.

$$MEP = \frac{WNET}{VDISP}$$

$$EFF = \frac{-100*WNET*NMO}{NM*URP}$$

$$POWER = WNET*RPM*\left[\frac{1min}{60sec}\right]*\left[\frac{1\ cycle}{2\ rev}\right]$$

$$SFC = \frac{FF}{POWER}$$

where,

- WNET= net work per cycle, kJ
- VDISP= displacement volume, m^3
- NMO= stoichiometric moles, kmol
- NM= number of kmol of fuel mixture, kmol
- URP= constant volume heat of reaction, kJ/kmol
- RPM= engine speed, rev/min
- FF= fuel flow, g/hr
- POWER= engine power per cycle, kW
- MEP= mean effective pressure, kPa
- EFF= brake thermal efficiency, %
- SFC= brake specific fuel consumption, g/kW-hr

The model takes into account progressive combustion, realistic valving and heat transfer during the compression, combustion and expansion processes. Friction and gas properties are determined from correlated functions. One of the largest uncertainties in the program is the heat transfer to or from the cylinder walls. Specifically, there exists no precise approximation to the heat transfer coefficient inside the cylinder during the cycle. This is an area of present research. The program does include a correlation for the heat transfer coefficient. This correlation appears to produce satisfactory coefficients for use in this model.

As noted before, the computer model was developed primarily from Campbell's book. However, there are three important features included in the program which are not discussed completely in this text. The features are heat transfer, mechanical friction and mass flow coefficients. Each of these areas were added through research of published literature.

The instantaneous heat transfer coefficient correlation was found in a technical paper entitled, "Convective Heat Transfer in Reciprocating Engines", by Bardon and Rao. The paper uses flow visualization studies and probe explorations to determine the complex nature of the turbulent flow inside a cylinder of an internal combustion engine. These results were combined with convection heat transfer theory to produce the following correlation.

$$H=.0006*\frac{S*CP*P}{T*R}*\left[\frac{VIC}{V}\right]^{\frac{1}{3}}$$

where,

H= inst. heat transfer coefficient, kW/m^2-K
S= stroke, m
CP= constant pressure specific heat, kJ/kmol-K
P= inst. pressure, kPa
VIC= cylinder volume at intake closing, m^3
V= inst. volume, m^3
T= inst. temperature, K
R= universal gas constant, kJ/kmol-K

This correlation produced instantaneous heat transfer coefficients of 200-3000 W/m^2-K. The occurrence of these values during the cycle have been examined and proven satisfactory for use in the model.

Another area of improvement in the model is the discharge coefficients during the valving events. The correlations used to determine the flow coefficients was taken from Taylor's text (Vol. I, p.173). It is listed below.

$$CD= \frac{-20}{3}*\left[\frac{L}{D} - .3\right]^2 + \frac{1}{3}*\left[\frac{L}{D} - .3\right] + .7$$

where,

CD=drag coefficient
L = valve lift, m
D = valve diameter, m

The drag coefficient is a function of the lift/valve diameter ratio which is found instantaneously during the valving subroutines.

The final feature added is a simple correlation for engine friction in a standard 4-stroke spark ignition engine. This correlation shown below was also found in Taylor's text (Vol. II, p.382).

$$FMEP= \frac{101}{14.7}*\left[.076*[MPS]^2 + .92*[MPS] + 7.23\right]$$

where,

FMEP=frictional mean effective pressure, kPa
MPS = mean piston speed, m/s

Figures 1a & 1b show the effect of adding the three non-ideal conditions to the model derived from Campbell's book. The simulations were conducted on a Chevy 5.74 liter engine burning gasoline. Curve one is the model developed from the text. The second curve is the same model with the addition of flow

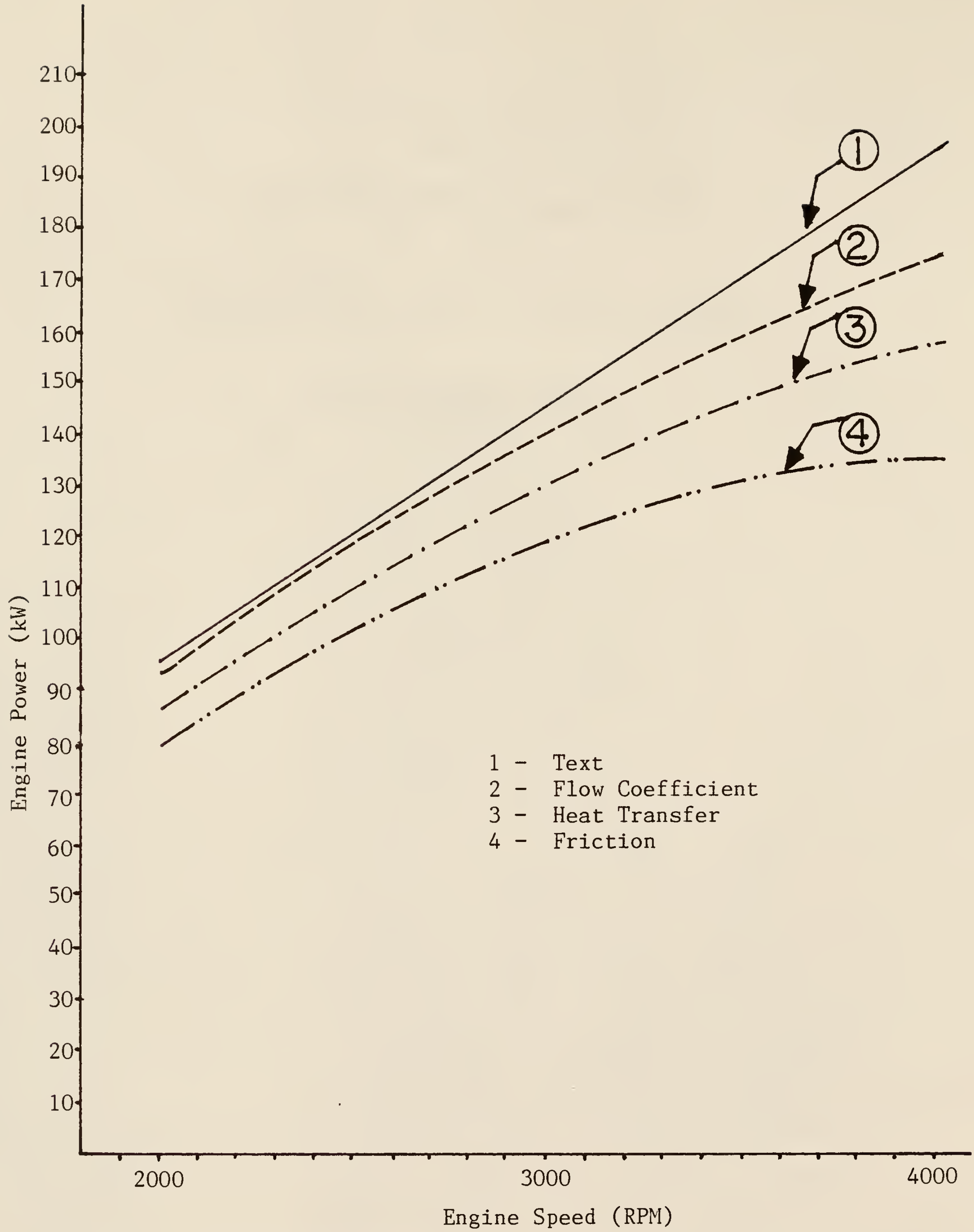


Figure 1a - Engine power versus engine speed for a Chevy 5.74 liter engine with various non-ideal processes.

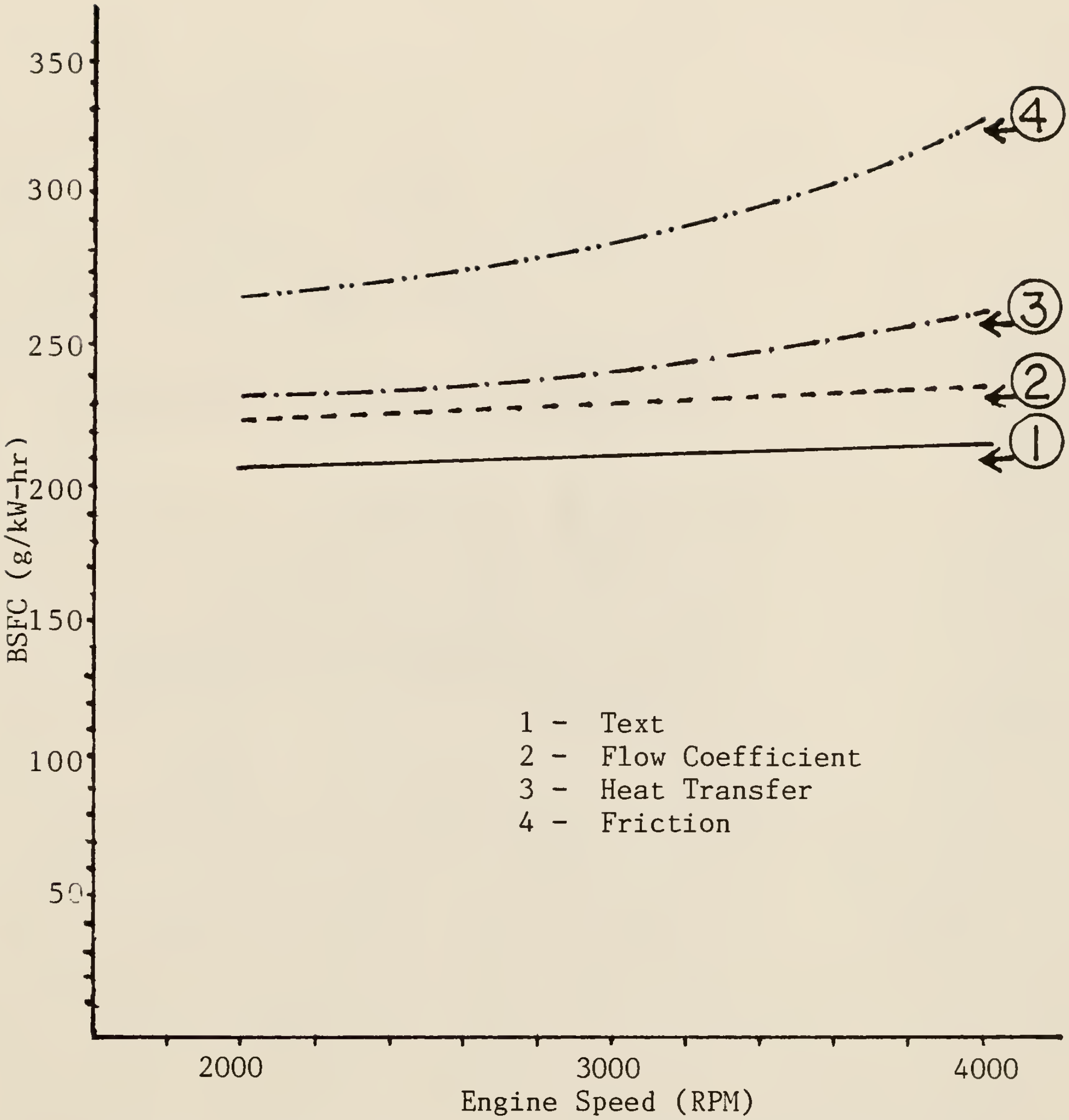


Figure 1b- Brake specific fuel consumption versus engine speed for a 5.74 liter engine with various non-ideal processes.

coefficients during valving. The third curve shows the characteristics of the model including flow coefficients and heat transfer. The final curve, number 4, which best approximates the actual engine, is the model used in this work including flow coefficients, heat transfer and friction.

Other irreversible processes such as inertial effects, chemical equilibrium, heat transfer during valving, pumping losses and parasitic losses are not included in this model. These losses combine for at least a 10% uncertainty within the program.

The input parameters for the model are the compression ratio, type of fuel, air/fuel ratio, bore (cm), stroke (cm), connecting rod length (cm), valve lift (cm), valve dimensions (cm), valve timing, exhaust pressure (kPa), engine speed (RPM), intake manifold pressures (kPa) and temperatures (K). All of these parameters are important characteristics necessary for an approximate simulation of an actual engine.

3.2 Validity:

After developing the model it was necessary to determine the validity of its predictions. Since all engines can be stocked or modified in several different configurations, it was difficult to receive performance information from a professional testing agency for the particular engine in question. The predicted performance information for a Chevy 5.74 liter V-8 engine burning gasoline was compared with some good experimental data which Dr. Don Fenton, professor at Kansas State University, had taken while employed at New Mexico State University. The results of the comparison can be seen in Figures 2 & 3. From Figure 2 note that the computer simulation predicts the experimental data to within about 10 percent. Agreement is better at high engine speeds -- above 3800 RPM. Note also that the uncertainty is consistently positive (experimental data less than predicted power) and so the model is expected to yield accurate trends associated with comparisons. Figure 3 suggests an uncertainty of less than 8% between model prediction and experimental data for the BSFC. Therefore, it is assumed that the model exhibits enough validity to continue with the modeling of a more-complete-expansion cycle.

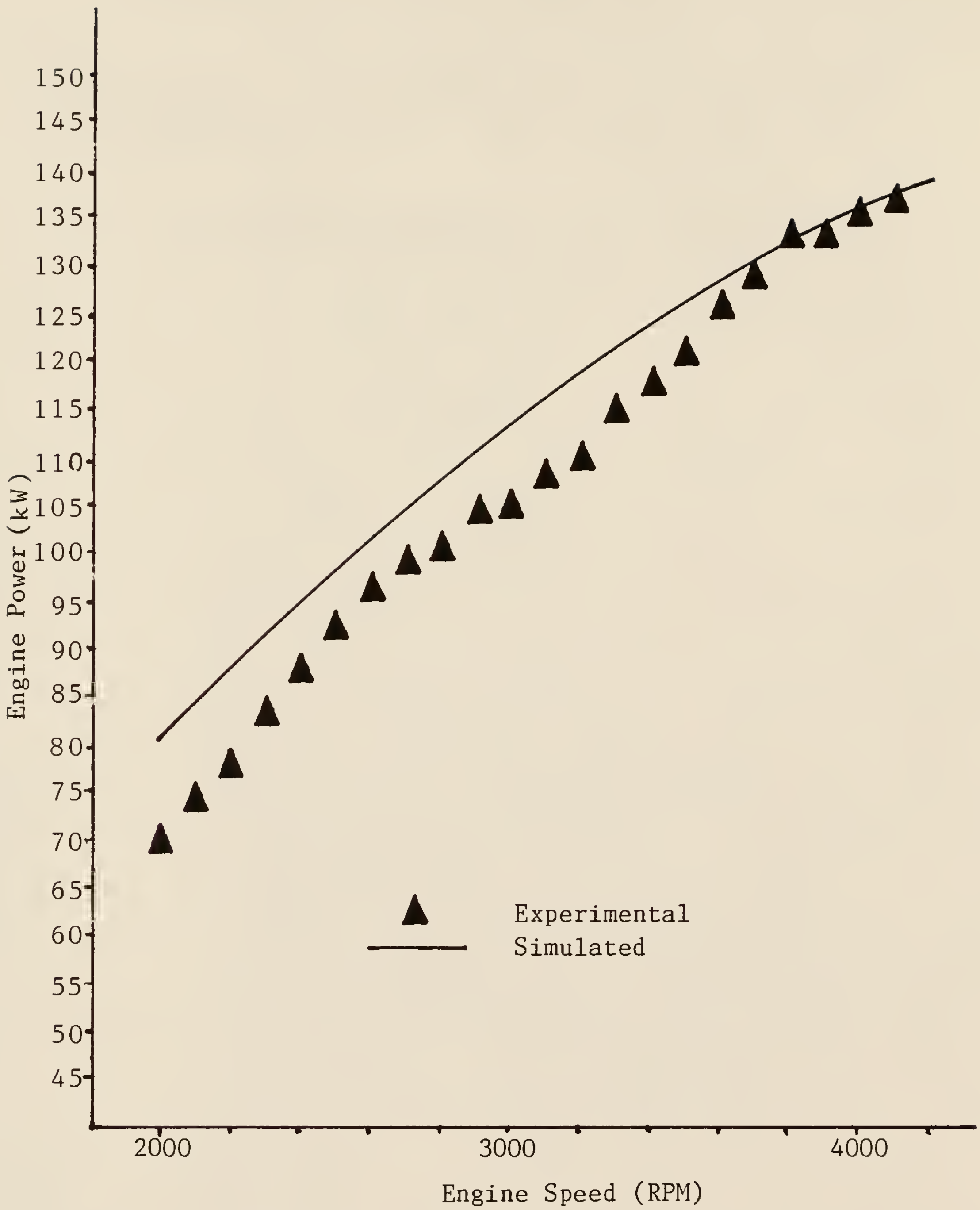


Figure 2 - Engine power versus engine speed for an experimental and computer simulated Chevy 5.74 liter with CR=8.5.

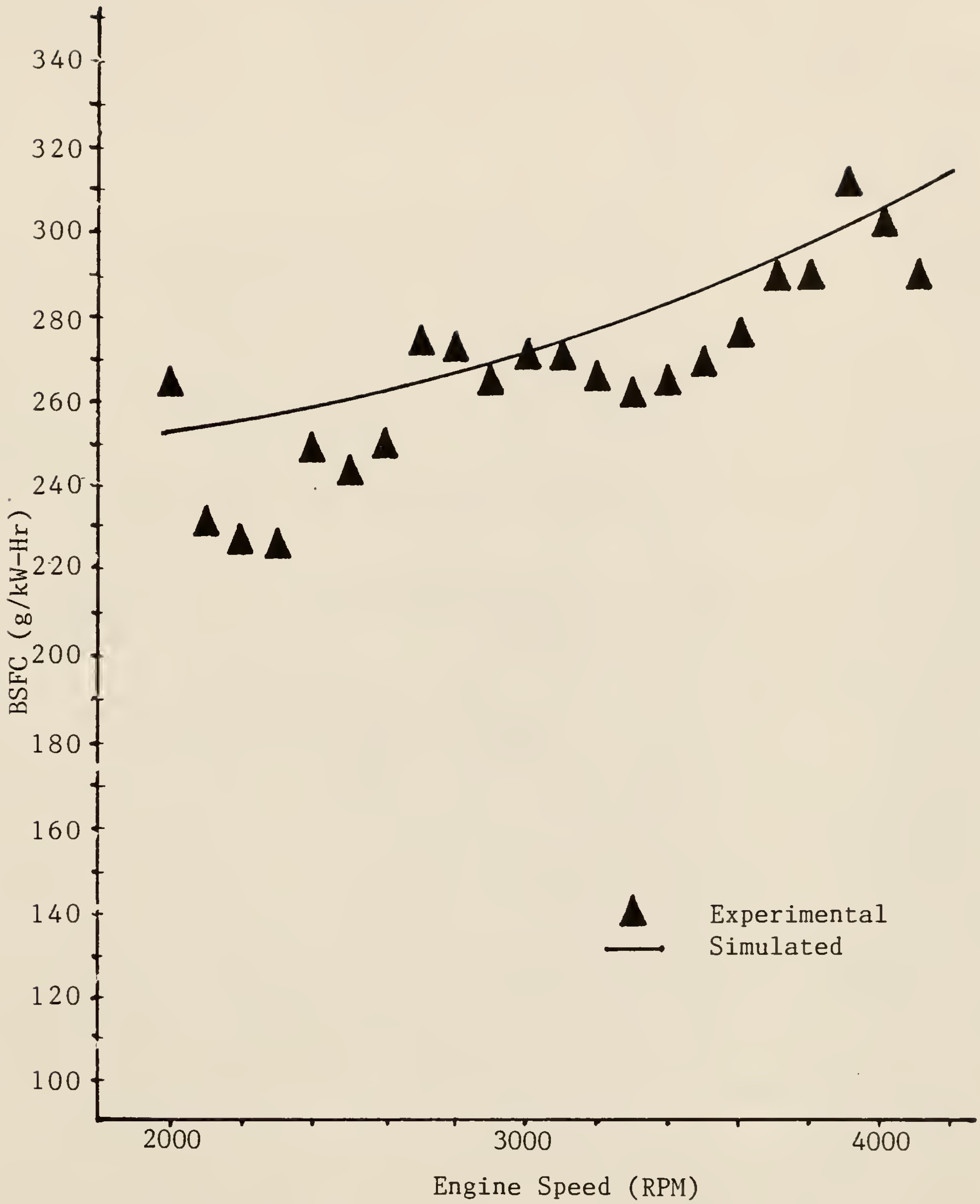


Figure 3 - Brake specific fuel consumption versus engine speed for an experimental and computer simulated Chevy 5.74 liter with CR=8.5.

4.0 DISCUSSION OF REFERENCE CASE

The reference engine will be a Chevy 5.74 liter V-8 operated at part-throttle, approximately 178 mm Hg vacuum, and standard atmospheric conditions. The reason for selecting this type of internal combustion engine is due to its popularity and accessibility of modified parts. The input parameters for the reference case are listed in Appendix I. These values are true measurements for a stock Chevy 5.74 liter V-8 engine.

Simulations were conducted on the reference engine to determine the correct air/fuel ratio for maximum performance. The stoichiometric air/fuel ratio for methane gas is 2. This means that for every 1 mole of fuel 2 moles of air are required for complete combustion. Complete combustion refers to the condition where the products contain neither carbon monoxide nor excess air. This condition also results in the greatest adiabatic flame temperature which is a very important factor in an internal combustion engine.

Experimental tests show that maximum power output and fuel efficiency are achieved at the stoichiometric condition or when slightly lean. (Ferguson, 1986). Figure 4 shows the results from the simulation. It shows that the model predicts maximum performance occurring at a lean air/fuel ratio of 2.2. This value will be used in the remaining simulations.

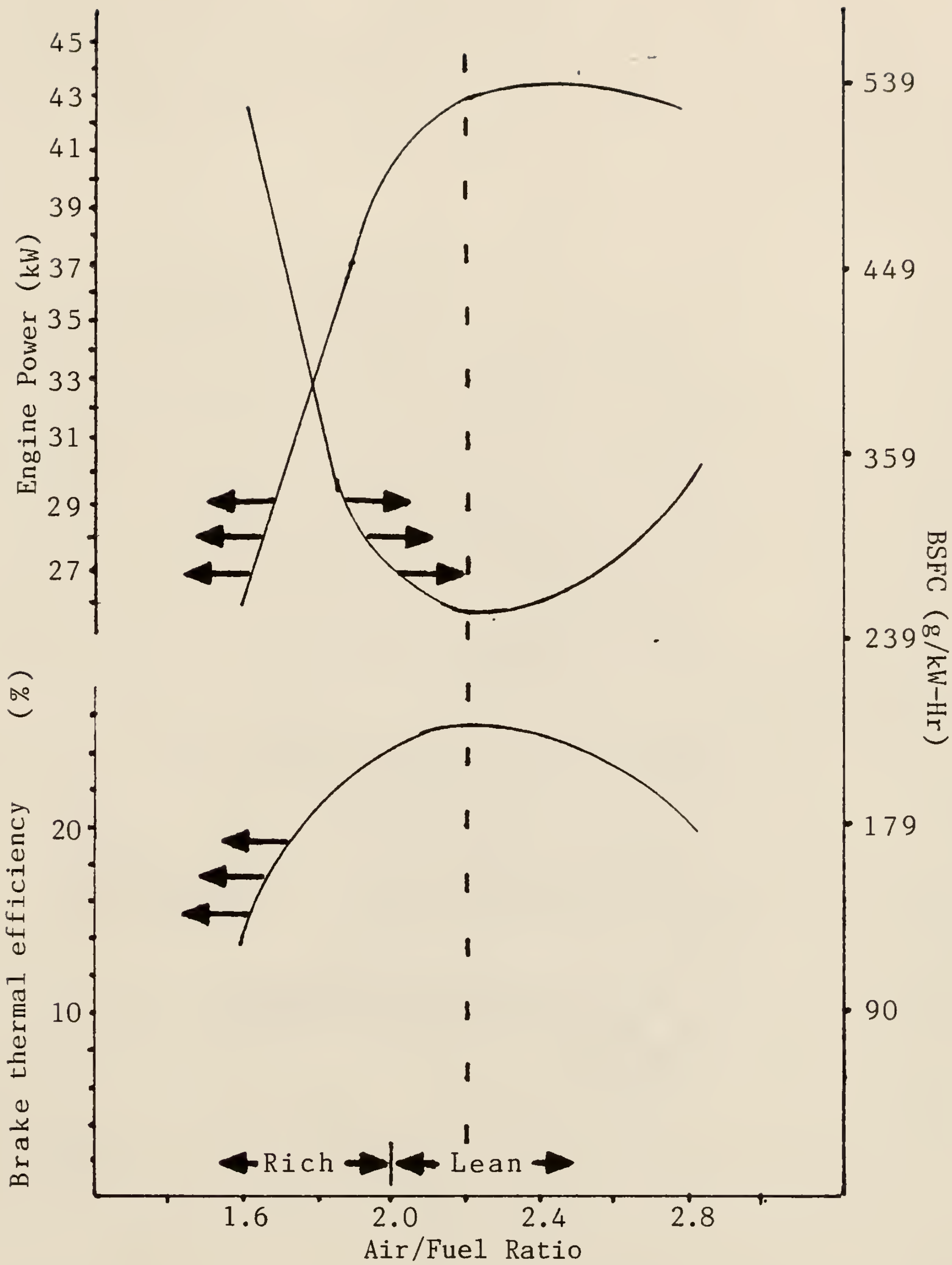


Figure 4 - Air/Fuel ratio versus efficiency, engine power, and brake specific fuel consumption for a Chevy 5.74 liter with CR=8.5. RPM= 2000.

Table 1 is a list of performance data generated by the computer for the reference case varying the RPM. The information shown will be useful for comparison purposes as the research progresses.

TABLE 1

Computer generated performance data for a Chevy 5.74 liter engine operating at part-throttle, approx. 178 mm Hg vacuum, and standard atmospheric conditions. CR=8.5, A/F=2.2, TM=300 K, and Methane Fuel.

RPM	SA DEG BTDC	MEAN PISTON SPEED (m/s)	VALVE TIMING EXH OPEN DEG BBDC	INT CLOSE DEG ABDC	MEP (kPa)	kW	BSFC (g/kW-Hr)	FF (kg/Hr)	VOL EFF (%)	PEAK PRESSURE (MPa)	EFF (%)
1600	27.2	4.71	77.0	76.0	472	36.1	274	79	78	3.4	26.3
1700	27.6	5.01	77.0	76.0	469	38.2	277	84	78	3.4	26.1
1800	28.0	5.30	77.0	76.0	467	40.3	279	89	78	3.4	25.8
1900	28.4	5.60	77.0	76.0	464	42.2	281	95	78	3.4	25.6
2000	28.9	5.89	77.0	76.0	461	43.9	284	100	79	3.4	25.3
2100	29.3	6.19	77.0	76.0	457	46.0	287	105	79	3.4	25.1
2200	29.7	6.48	77.0	76.0	453	47.7	290	111	79	3.4	24.9
2300	30.1	6.78	77.0	76.0	450	49.4	293	116	79	3.4	24.5
2400	30.5	7.07	77.0	76.0	444	50.9	297	121	79	3.4	24.2
2500	30.9	7.37	77.0	76.0	440	52.4	301	127	79	3.4	23.9

5.0 MORE-COMPLETE-EXPANSION CYCLE

The more-complete-expansion cycle is analogous to a standard Otto Cycle except for the increased expansion ratio and decreased compression ratio. Both cycles involve compression, combustion, expansion, exhaust, and intake of a certain fuel which is burnt to produce useful power output. In the more-complete-expansion cycle, the ignition pressure is controlled by early or late closing of the intake valve. Typically, the cycle in which the intake valve is closed early is known as an Atkinson Cycle and the other is called a Modified Atkinson Cycle. Whichever cycle is chosen the results are usually very similar.

The expansion process in an ideal more-complete-expansion cycle is lengthened by late opening of the exhaust port, thus allowing expansion of the gas to continue until it reaches atmospheric pressure. Upon reaching atmospheric pressure, the piston is usually at bottom dead center at which time the exhaust valve opens and the piston pushes the gases out of the cylinder. However, this type of cycle is not viable for engines that operate at partial-load because in light load situations the mean cylinder pressure at the end of expansion tends to be near or below the pressure required to overcome friction. Under such circumstances the application of a more-complete-expansion process leads to a net loss rather than a gain in efficiency.

The reason for this arrangement is that maximum cylinder pressure is usually limited by detonation or knock. In the more-complete-expansion cycle, controlling the ignition pressure results in a more controllable maximum cylinder pressure. In the past, spark ignition engines used increased compression ratios to increase cycle efficiency. This was severely restricted because of detonation and requirements for higher octane fuels. Therefore, the only other feasible way to significantly increase cycle efficiency is to increase the expansion ratio.

In the simulations to follow, this investigation will attempt to determine the effects of adding such a process to an irrigation engine that operated at part-throttle. The results will be compared to the reference case to determine the advantages of the modification.

5.1 Matched ignition pressure with late intake closing:

The technique used in the simulations was first to open the throttle plate wide open allowing the intake manifold pressure to be atmospheric. Secondly, while increasing the compression ratio the intake valve closing was delayed until the pressure prior to ignition was the same as the ignition pressure in the reference engine. The reason for matching the ignition pressures is so all the performance information generated by the model will have one point in common. Comparing the two results provide a useful way to determine how the more-complete-expansion process effects the remaining cycle.

The increase in compression ratio not only improves efficiency but also helps offset the decrease in peak cylinder pressure caused by the late closing of the intake valve.

The results of the first simulation varying the compression ratio from 8.5:1 to 13.5:1 while matching ignition pressures can be seen in Figures 5a-5g. Figure 5a illustrates how the compression ratio affects efficiency. It shows that as the compression ratio increases the efficiency increases, but only to a point where it levels off, namely at a compression ratio of about 12.5:1. This could have been

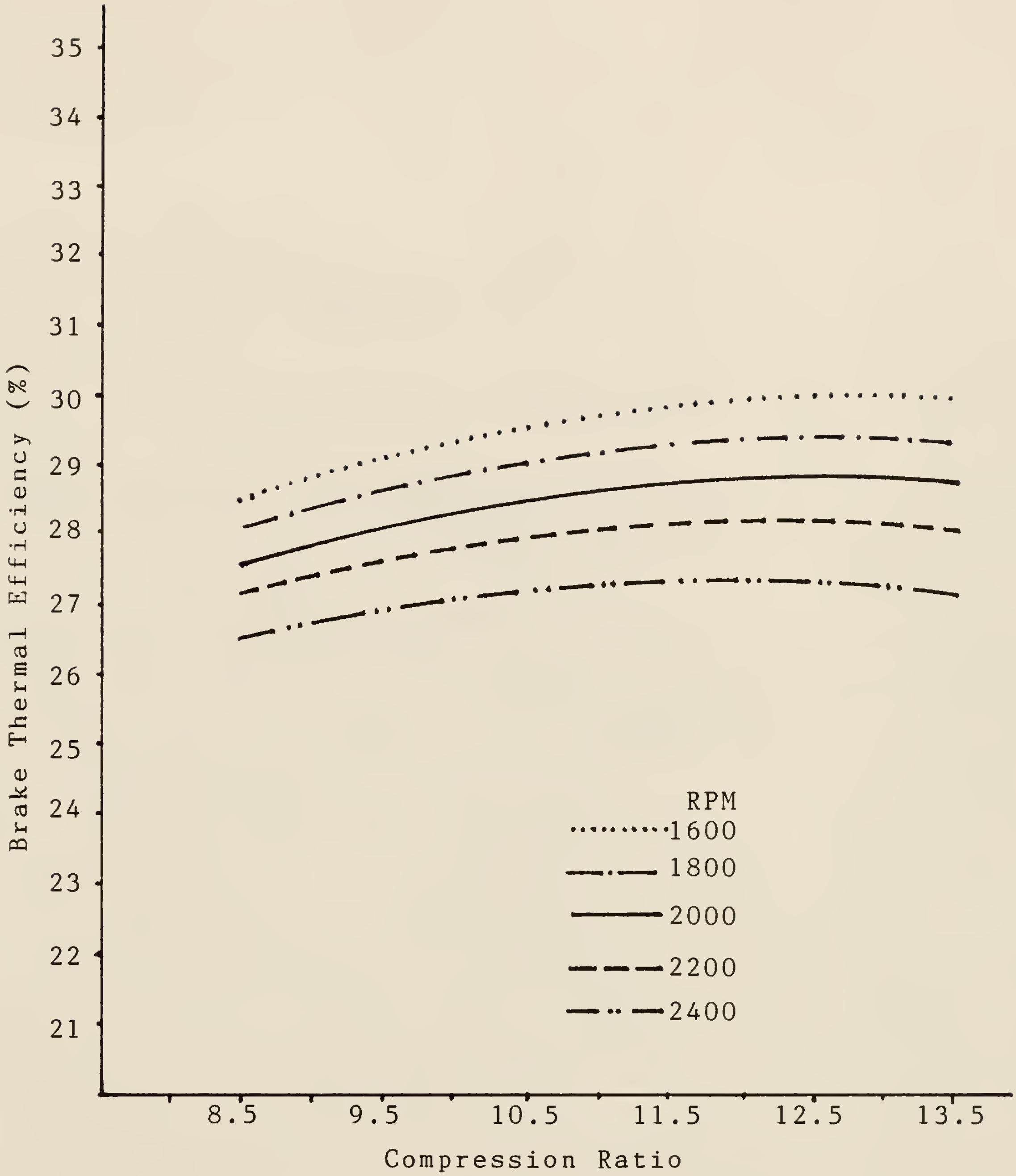


Figure 5a - Efficiency versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (late intake valve closing)

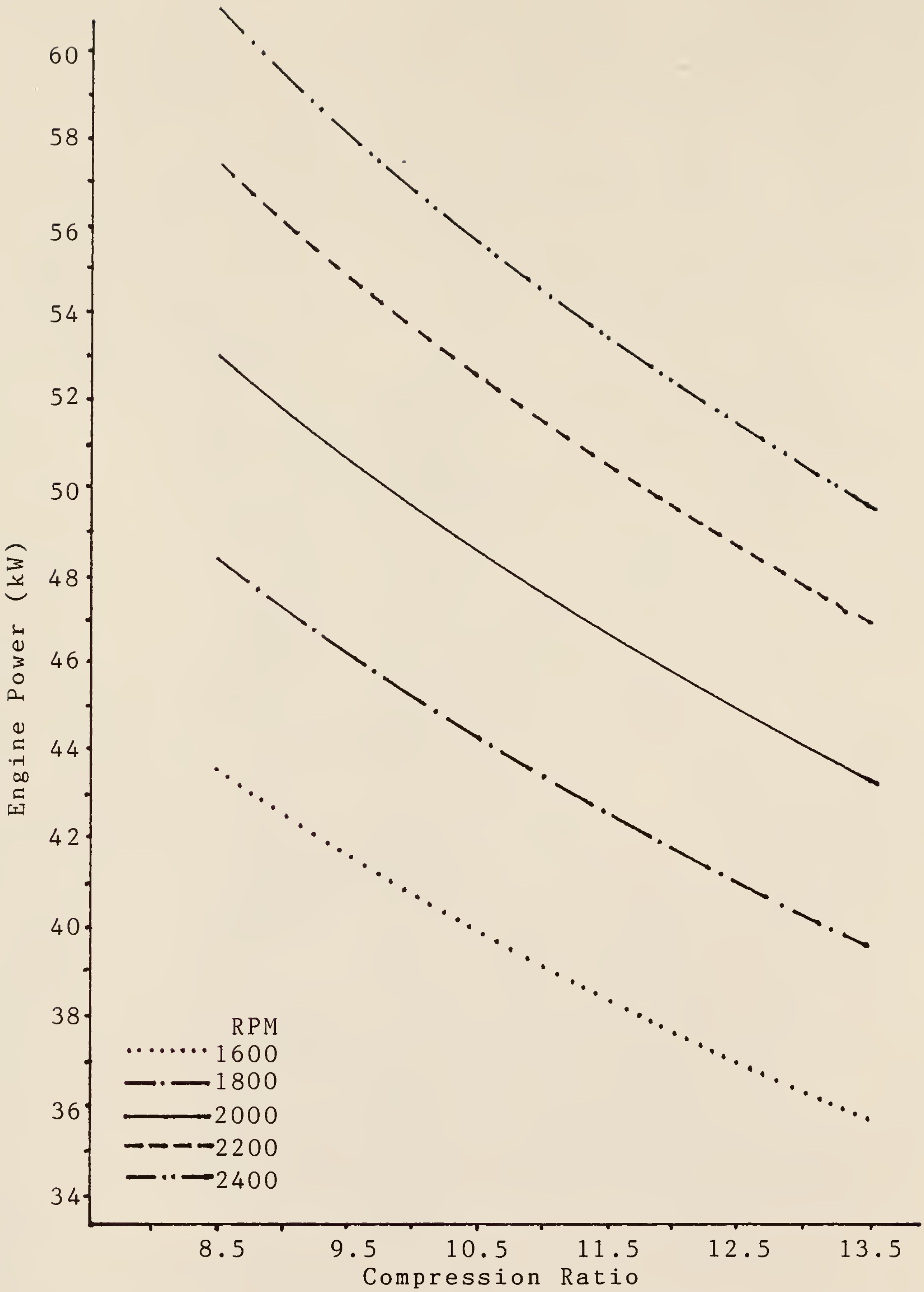


Figure 5b - Engine power versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (late intake valve closing)

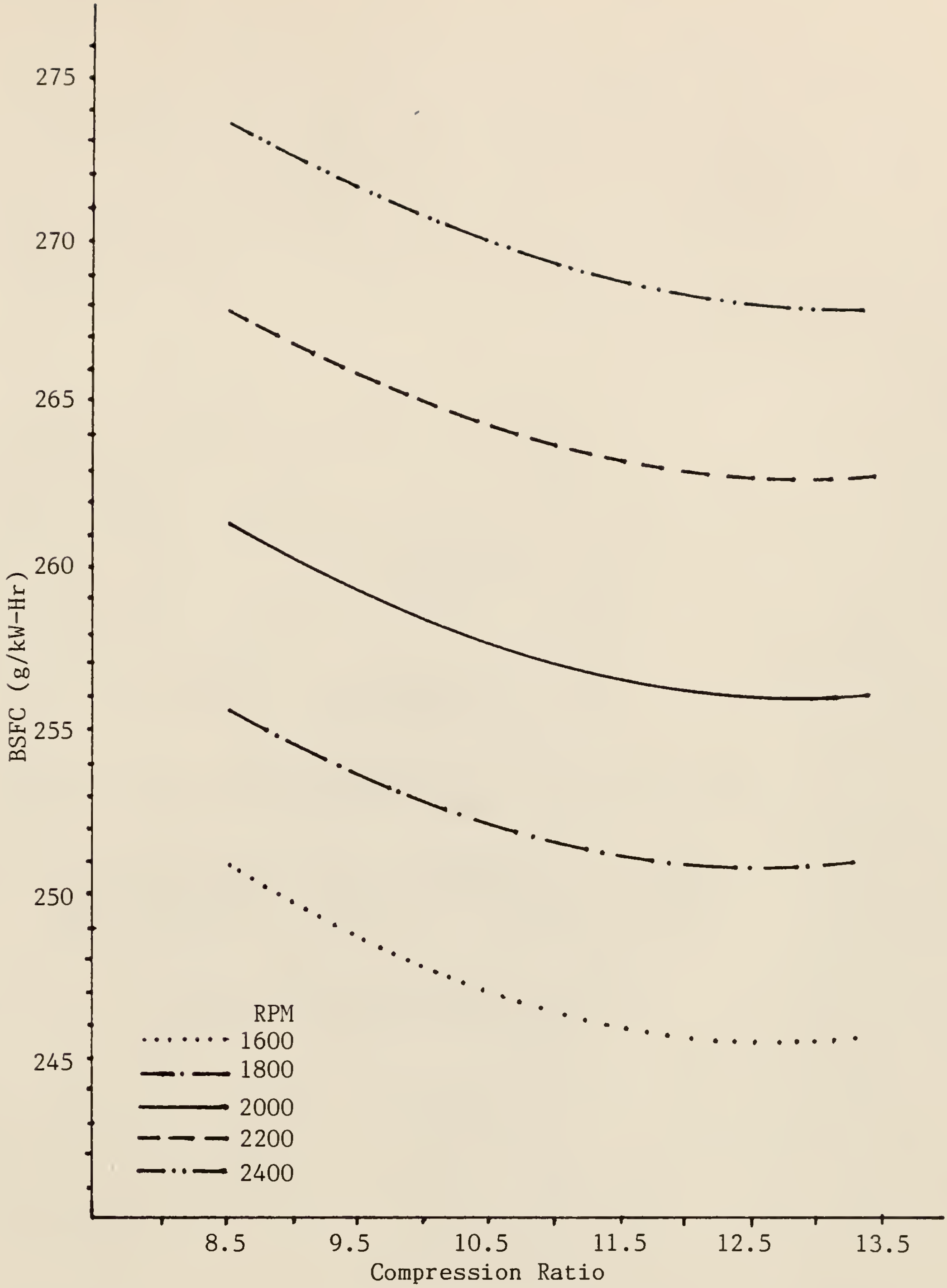


Figure 5c - Brake specific fuel consumption versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (late intake valve closing)

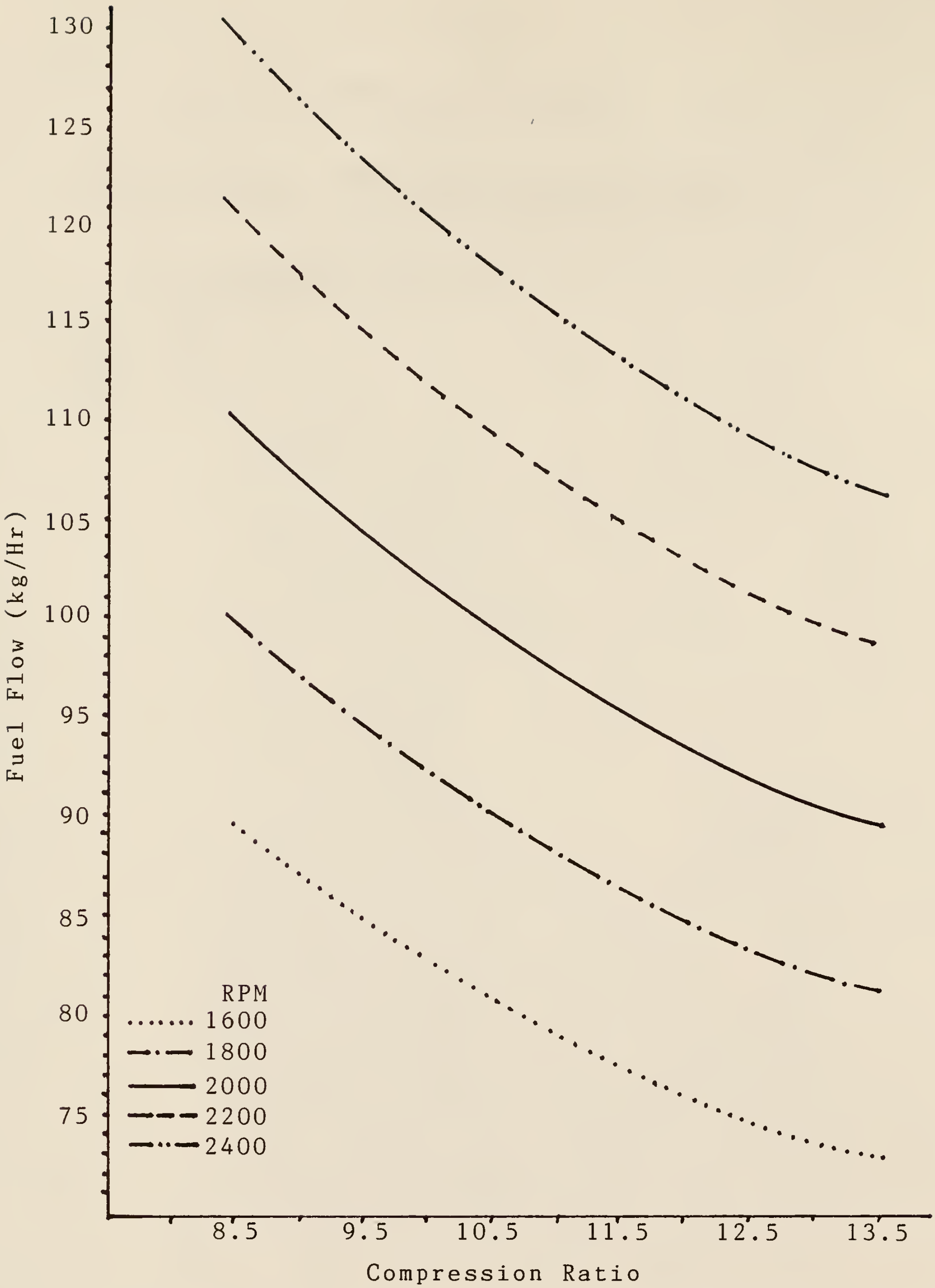


Figure 5d - Fuel flow versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (late intake valve closing)

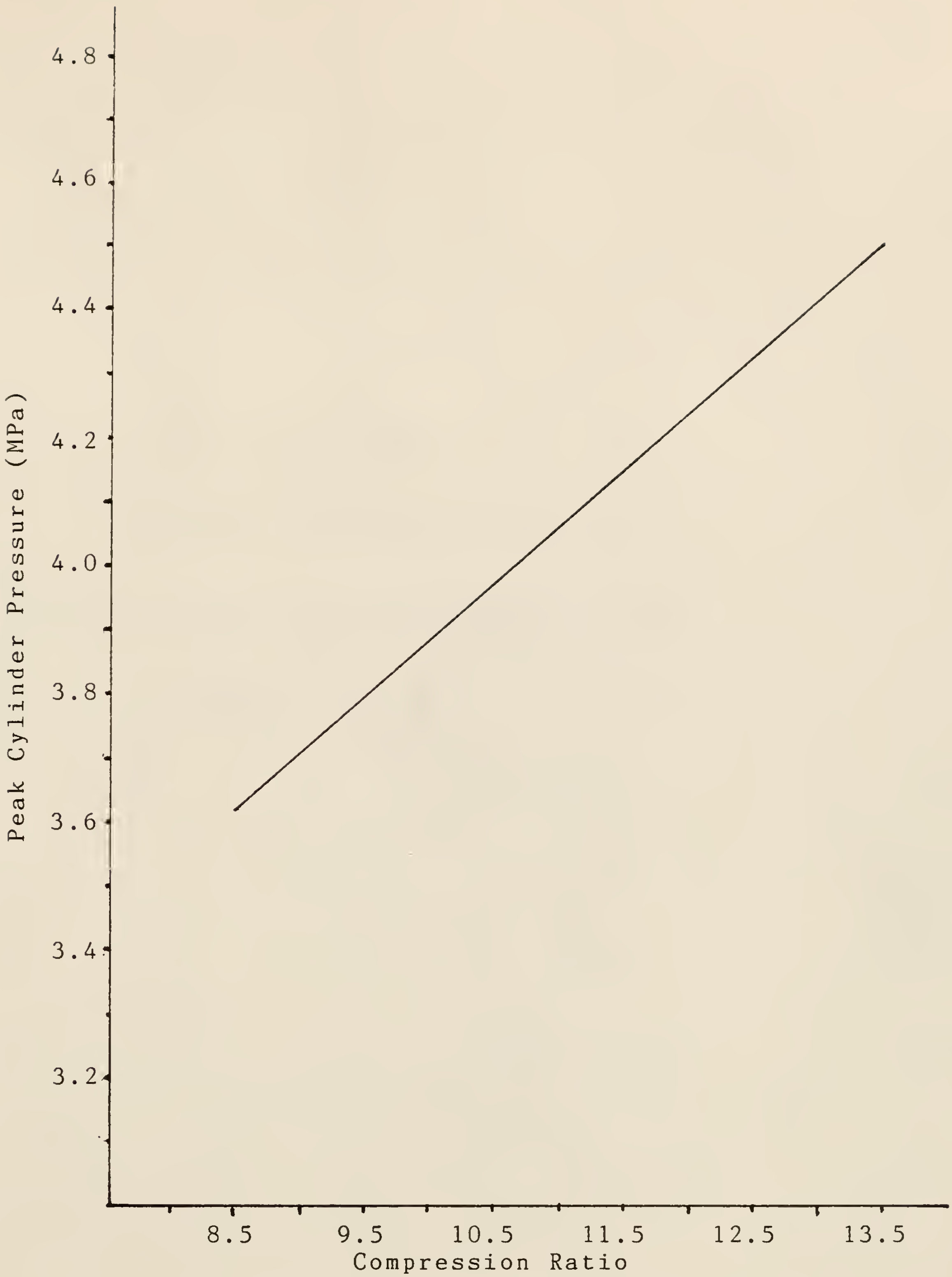


Figure 5e -Peak cylinder pressure versus compression ratio for Chevy 5.74 liter with matched ignition pressures. (late intake valve closing)

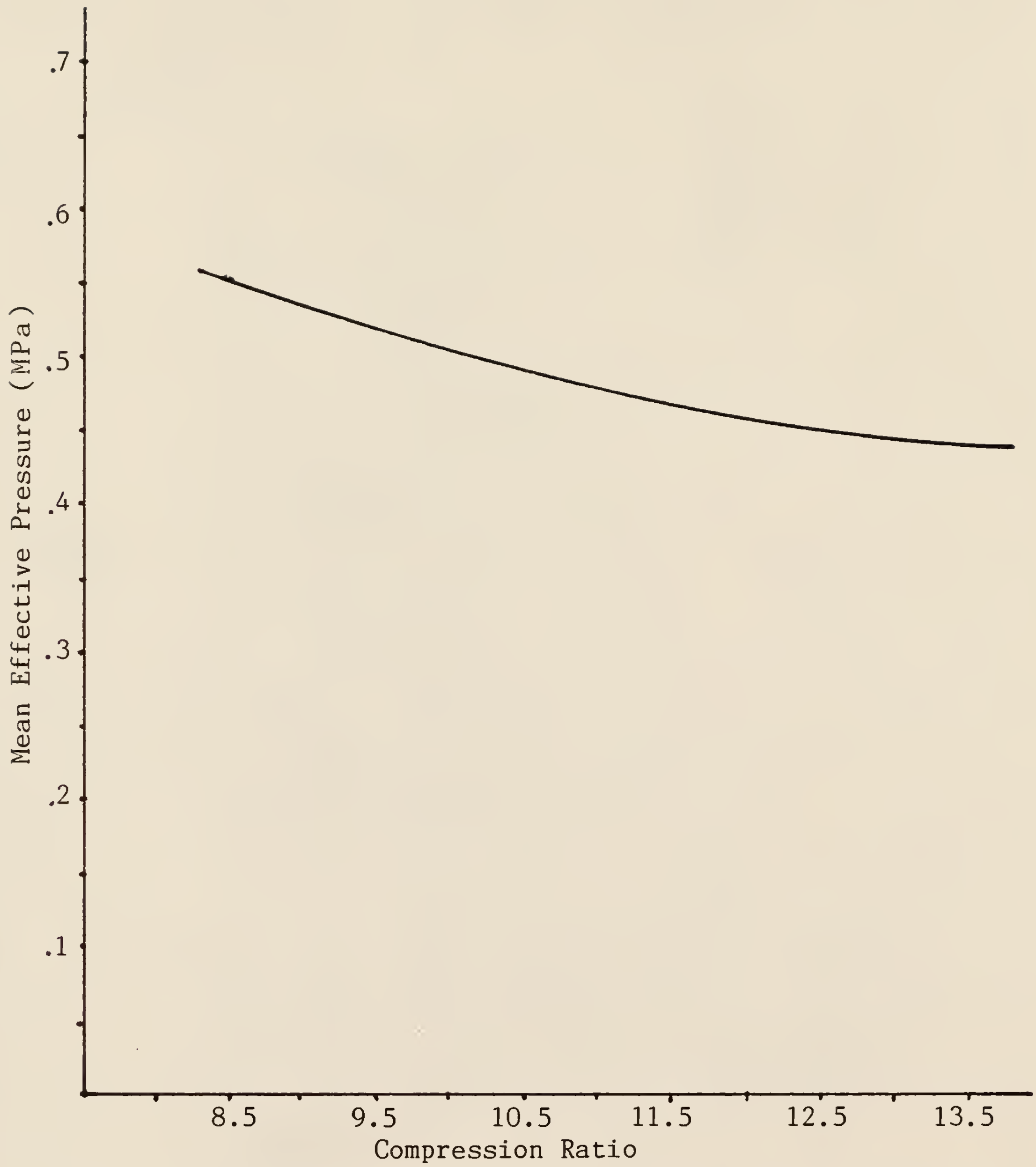


Figure 5.f - Mean effective pressure versus compression ratio for Chevy 5.74 liter with matched ignition pressures. (late intake valve closing)

Figure 5g -Volumetric efficiency versus compression ratio for Chevy 5.74 liter with matched ignition pressures. (late intake valve closing) predicted. Similar results occur for an air-standard cycle, except for the rapidly leveling off of the curve.

Figure 5b shows the effect of the compression ratio on the power output. This curve continues to decrease as the compression ratio increases. The decrease in power output is due to the fact that the intake valve closing is continually retarded in order to match the ignition pressures. This retarding causes more of the freshly ingested charge to be expelled back into the intake manifold, thus decreasing the amount of fuel available for conversion to work. The exact effect of pushing some of the heated charge into the intake manifold is not known, however, increasing the intake manifold temperature by 8% tends to decrease the thermal efficiency by 4% and the power output by 12.3%. This situation is beyond the scope of this research because of the complexity involved in determining the correct intake manifold temperature. Should this configuration be chosen for experimentation, this effect must be examined since it tends to deteriorate engine performance.

Figure 5c is a plot of brake specific fuel consumption versus compression ratio. This curve is inversely similar to the efficiency curve since they are essentially dependent upon one another. The minimum BSFC point occurs at a compression ratio of 12.5:1.

As can be seen in Figure 5d, the fuel flow behaves like the power curve. Since the compression ratio is increased it is necessary to retard the intake valve closing so the pressure at ignition is matched. In the process of closing the intake valve later, more of the fresh charge is pushed out of the cylinder causing a reduction in fuel flow.

An unexpected situation arises when the peak pressure is plotted versus compression ratio. One would expect the peak pressure to be lower since a smaller amount of fuel is being burned as the compression ratio increases. This is not the case. (See Figure 5e) In fact, the peak pressure increases linearly as the compression ratio increases.

The underlying causes of this behavior are not completely understood. The analysis of this condition is complex and only a speculation can be made to determine the reasons for the linear behavior. Since the only two parameters which changed from cycle to cycle were the compression ratio and valve timing, it is understood that the peak cylinder pressure is a function of both parameters. The shape of the actual function may or may not be linear. The reason for this uncertainty is because of the simple model within the program used for the simulation of the combustion process. If a more accurate function is sought, a more detailed and complex model will be necessary.

It has been determined that the two factors involved in this behavior are the change in valve timing and an increased compression ratio. If each parameter is examined individually a better understanding of the speculation will be achieved.

The result of retarding the valve timing while holding the compression ratio constant is a reduced number of moles of fuel mixture ingested into the cylinder to be burned. The peak cylinder pressure and the mean effective pressure are lowered due to a smaller amount of fuel available for work.

The result of increasing the compression ratio while holding the number of moles of fuel constant is an increased peak cylinder pressure. This is caused by a change in the cylinder pressure during combustion which is broken into two parts - compression and combustion. Since the amount of fuel is constant, the pressure change associated with combustion remains the same. Therefore, the increased compression ratio causes the compressive pressure change to be larger. This results in a greater peak cylinder pressure.

When the two conditions occur simultaneously, each of the individual behaviors combine and offset one another to produce the linear function. Remember, each of these functions are unknown and it is only speculated that they combine to produce the straight line. As stated earlier, a detailed examination of each individual function is needed in order for a complete understanding of this behavior.

The following two figures (Figures 5f & 5g) are curves demonstrating how the more-complete-expansion cycle effects the mean effective pressure and the volumetric efficiency. The mean effective pressure remains somewhat constant while the volumetric efficiency deteriorates due to the retarding of the intake valve.

Another concern regarding application of the more-complete-expansion cycle is which exhaust valve opening position will maximize the cycle efficiency. From the previous analysis, it appears that increasing the compression ratio above 12.5:1 is unwarranted since cycle efficiency tends to level out. It is unwarranted because of the additional peak stresses placed on the cylinder to produce only a small change in efficiency.

For the purpose of determining the optimum exhaust valve timing a compression ratio of 12.5:1 was chosen. The results from the model are shown in Figures 6a-6c. All of these figures show that the optimum exhaust valve opening is 20 degrees BBDC. The reason for opening the exhaust valve before bottom dead center is because the remaining pressure in the cylinder is needed to overcome the drag forces on the exhaust gases caused by the valve dynamics.

On the basis of the previous simulations, the model predicts that the greatest advantage in performance can be achieved by modifying the reference engine in the following ways. Increase the compression ratio to 12.5:1, intake valve closing as shown in Table 2 and the exhaust valve opening at 20 degrees BBDC. The model predictions for this configuration are shown in Table 2 and the P-V diagrams of this cycle and the reference cycle are shown in Figure 7.

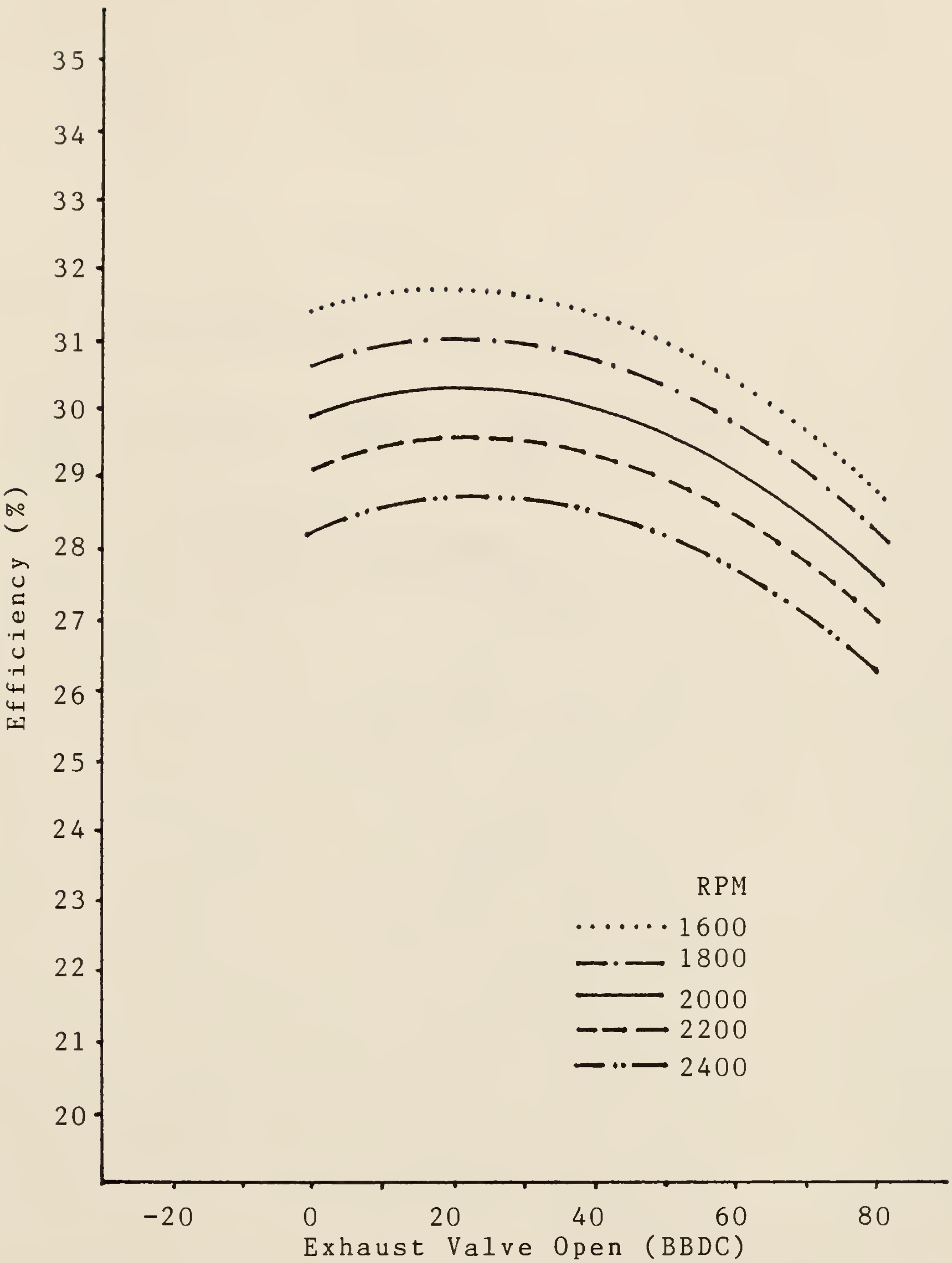


Figure 6a - Efficiency versus exhaust valve opening for a Chevy 5.74 liter with CR=12.5 and matched ignition pressures. (late intake valve closing)

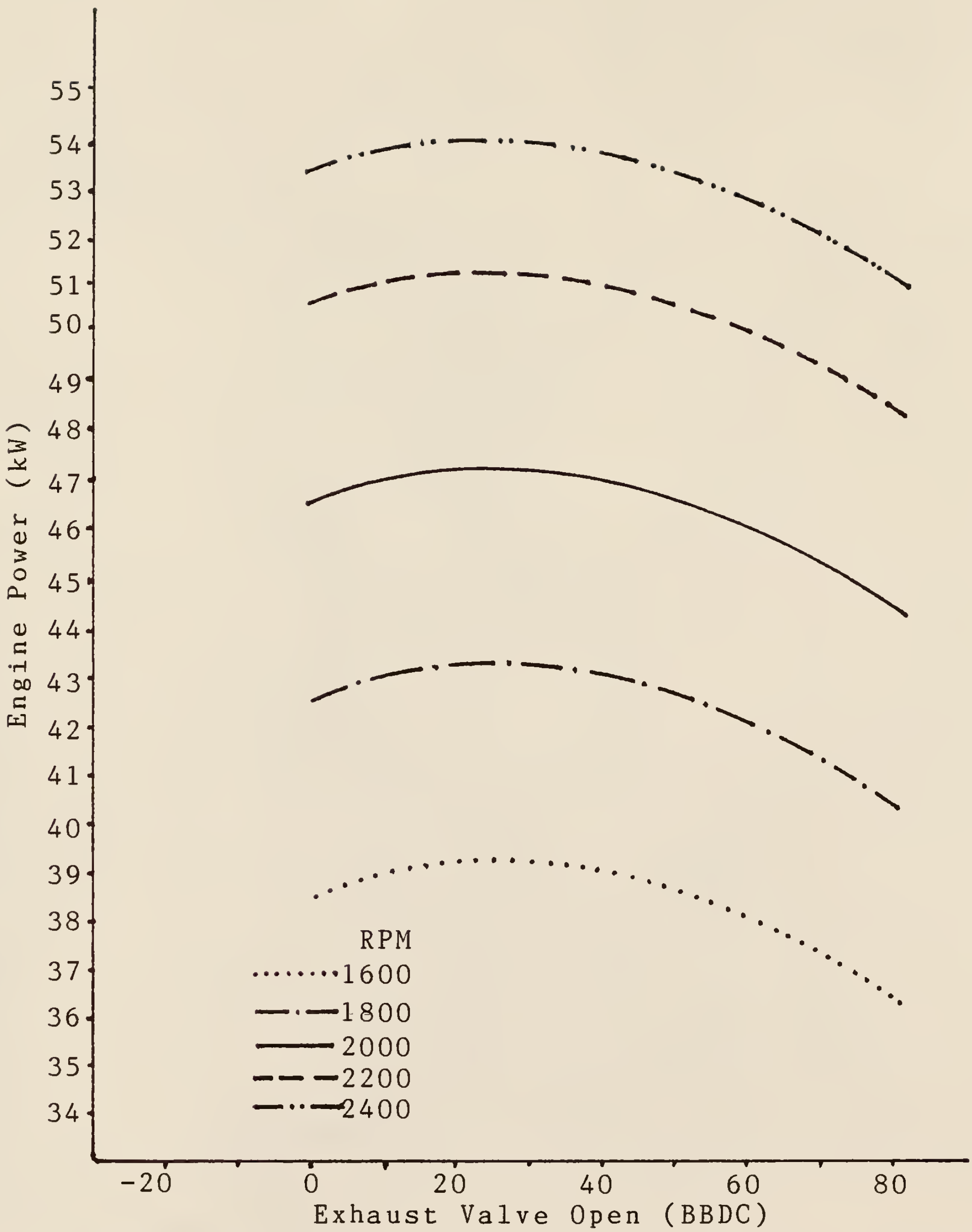


Figure 6b - Engine power versus exhaust opening for a Chevy 5.74 liter with CR=12.5 and matched ignition pressures. (late intake valve closing)

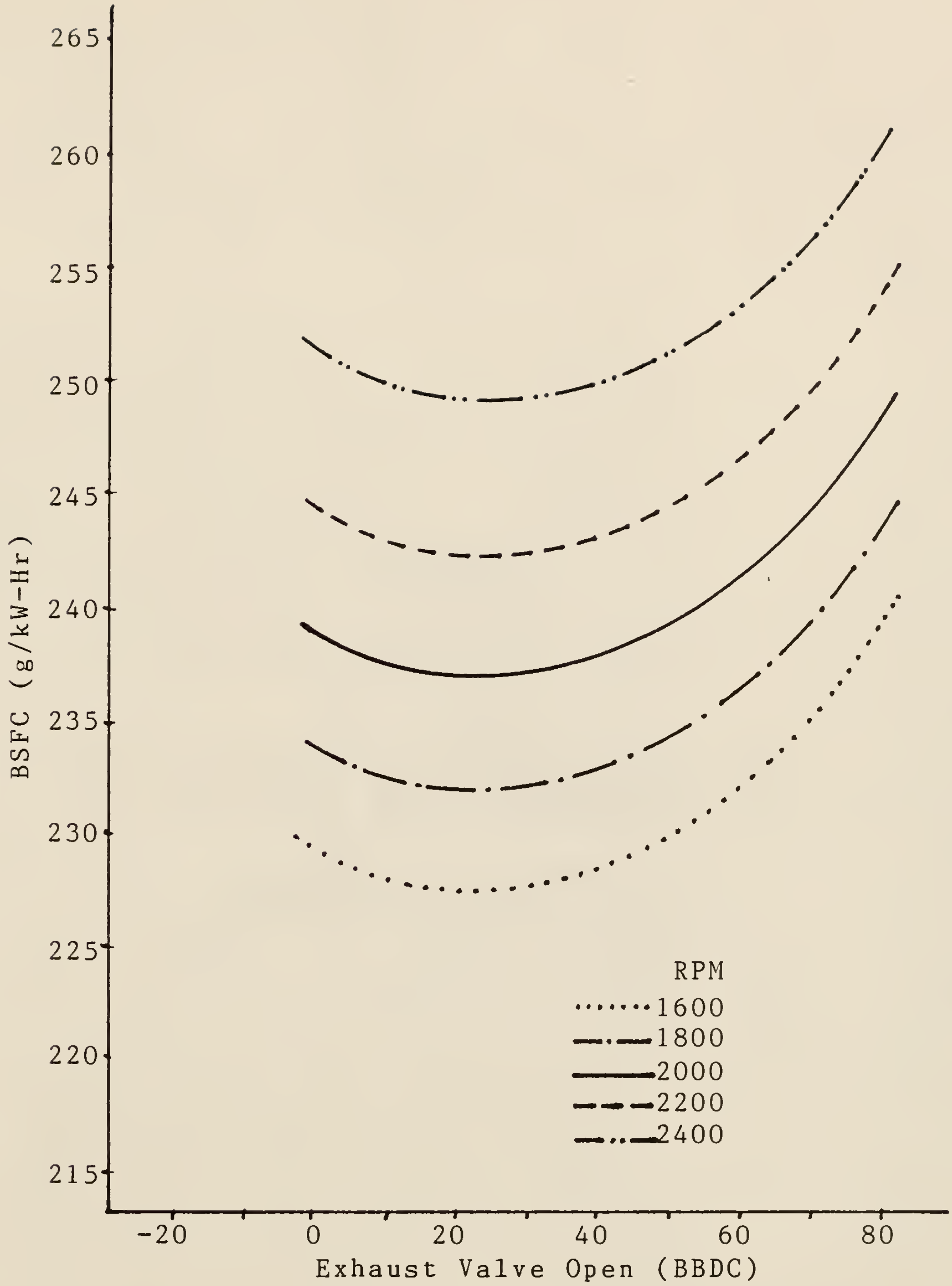


Figure 6c - Brake specific fuel consumption versus exhaust valve opening for a Chevy 5.74 liter with CR=12.5 and matched ignition pressures. (late intake valve closing)

TABLE 2

Computer generated performance data for a Chevy 5.74 liter engine operating at full-throttle, same ignition pressures as part-throttle case, and standard atmospheric conditions. CR=12.5, A/F=2.2, TM=300 K, and Methane Fuel.

RPM	SA	MEAN PISTON SPEED	VALVE TIMING		MEP	kW	BSFC	FF	VOL EFF	PEAK PRESSURE	EFF
			EXH OPEN	INT CLOSE							
	DEG BTDC	(m/s)	DEG BBDC	DEG ABDC	(kPa)		(g/kW-Hr)	(kg/Hr)	(%)	(MPa)	(%)
1600	27.2	4.71	20.0	112.8	505	38.6	229	71	52	4.3	31.6
1700	27.6	5.01	20.0	113.7	495	40.4	232	75	52	4.3	31.2
1800	28.0	5.30	20.0	113.8	495	42.7	234	80	52	4.3	30.8
1900	28.4	5.60	20.0	113.6	495	45.4	236	86	53	4.4	30.6
2000	28.9	5.89	20.0	114.4	495	47.0	239	90	53	4.3	30.2
2100	29.3	6.19	20.0	114.9	485	48.8	242	95	53	4.3	29.8
2200	29.7	6.48	20.0	114.9	485	51.0	245	100	53	4.3	29.5
2300	30.1	6.78	20.0	116.3	464	51.7	250	103	53	4.3	28.9
2400	30.5	7.07	20.0	116.3	464	53.9	253	109	53	4.3	28.6
2500	30.9	7.37	20.0	116.1	464	56.1	256	115	54	4.4	28.2

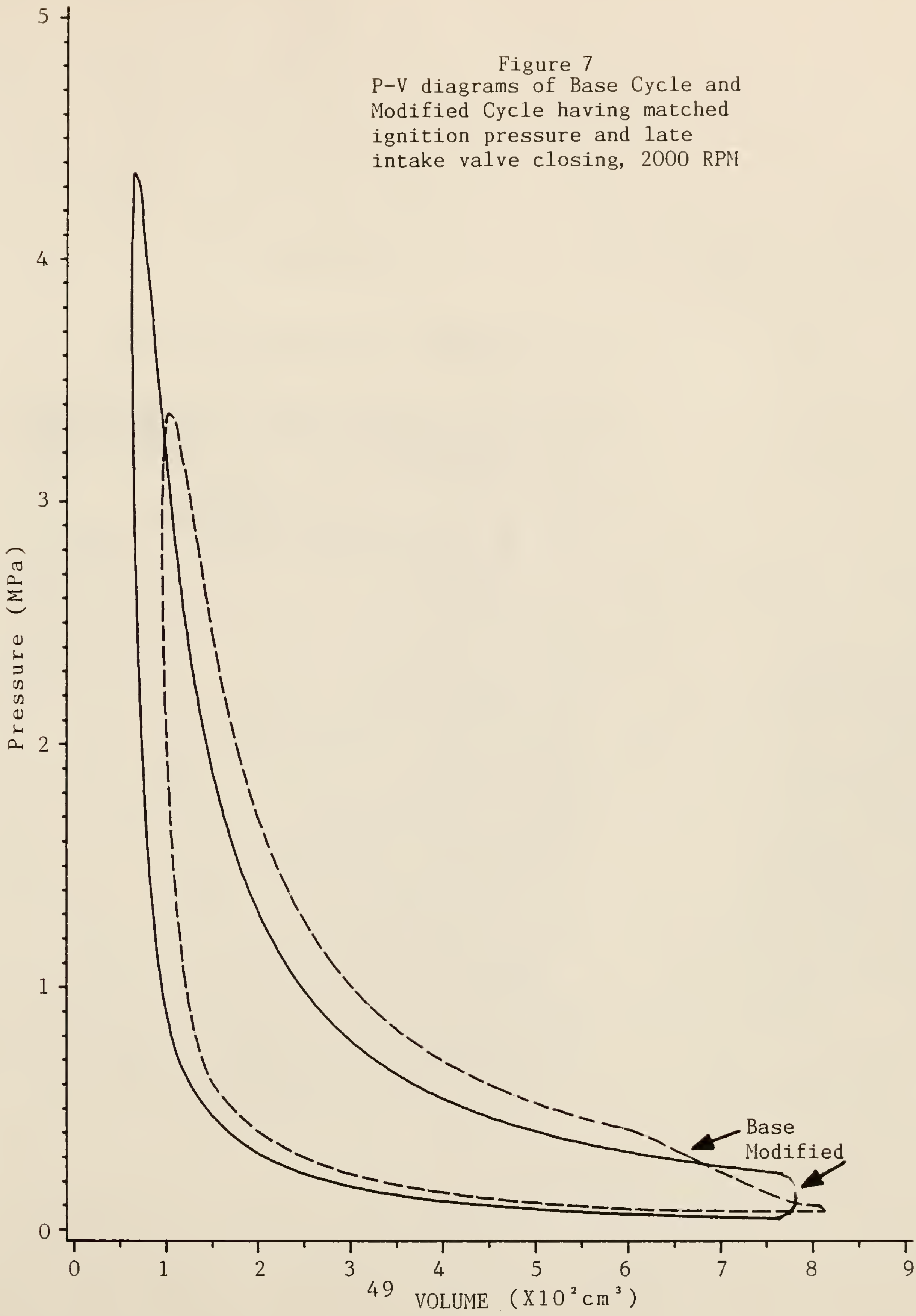


Figure 7
P-V diagrams of Base Cycle and Modified Cycle having matched ignition pressure and late intake valve closing, 2000 RPM

5.2 Matched ignition pressure early intake closing:

The technique used in the following simulation is similar to those used in the late closing case. In the early intake closing case the intake valve is closed before the intake process is complete. The gases in the cylinder are then pulled below atmospheric pressure until bottom dead center. After bottom dead center, the piston begins to compress the gases until the appropriate ignition pressure is reached. The advantage in using this type of arrangement is that none of the heated gases are forced back into the intake manifold causing a temperature increase. The purpose of closing the intake valve early and then pulling the gases in the cylinder toward a vacuum is to effectively reduce the compression ratio. The cycle analysis of this situation is much the same as before.

Once again the ignition pressures are matched, compression ratio varied and the intake valve is closed early. In Figures 8a-8g the effects of the more-complete-expansion cycle and varying the compression ratio can be seen. Each of the curves behave similarly to the case in which the intake valve closed late. Therefore, the best compression ratio to select without stress consideration is 12.5:1.

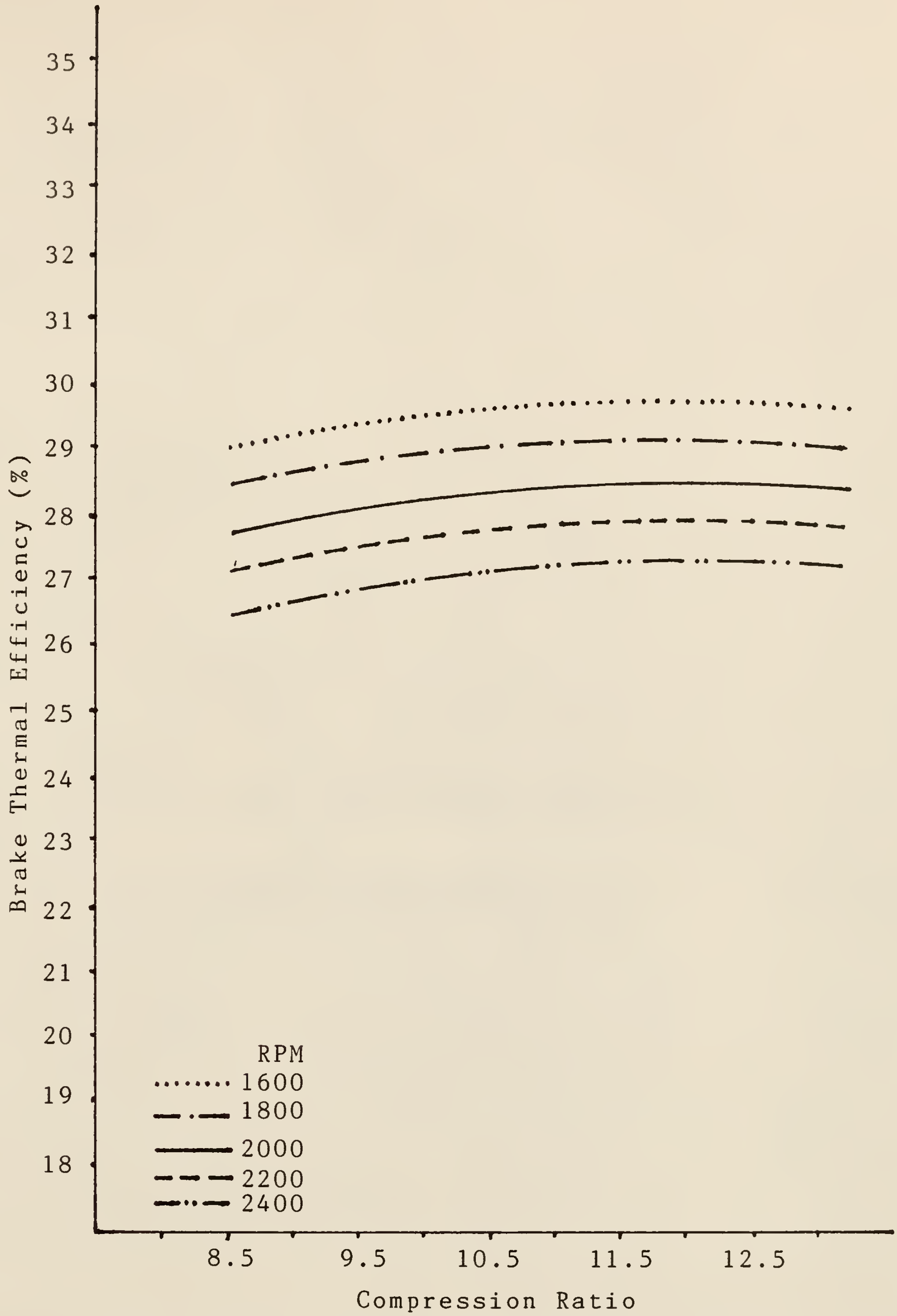


Figure 8a - Efficiency versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (early intake valve closing)

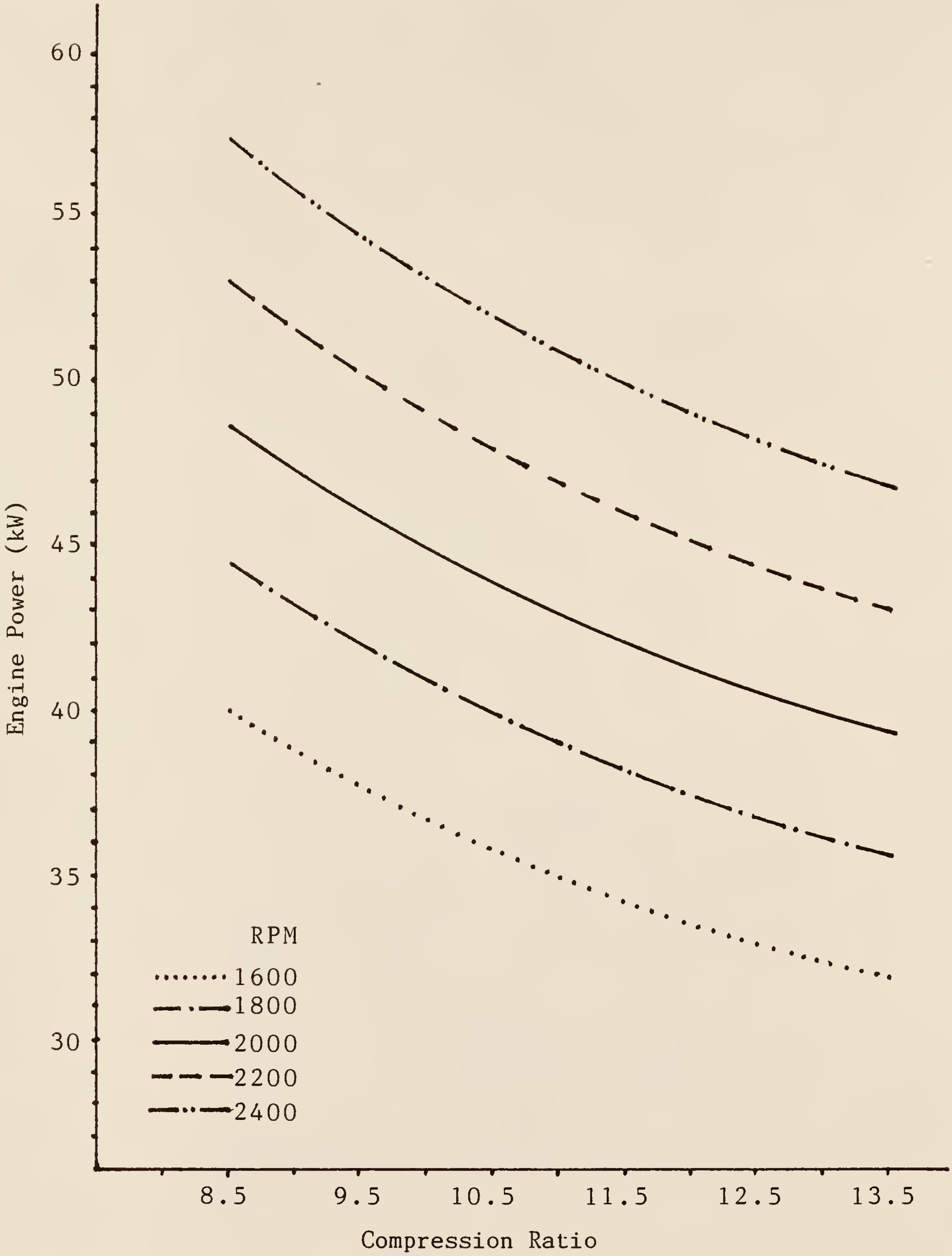


Figure 8b - Engine power versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (early intake valve closing)

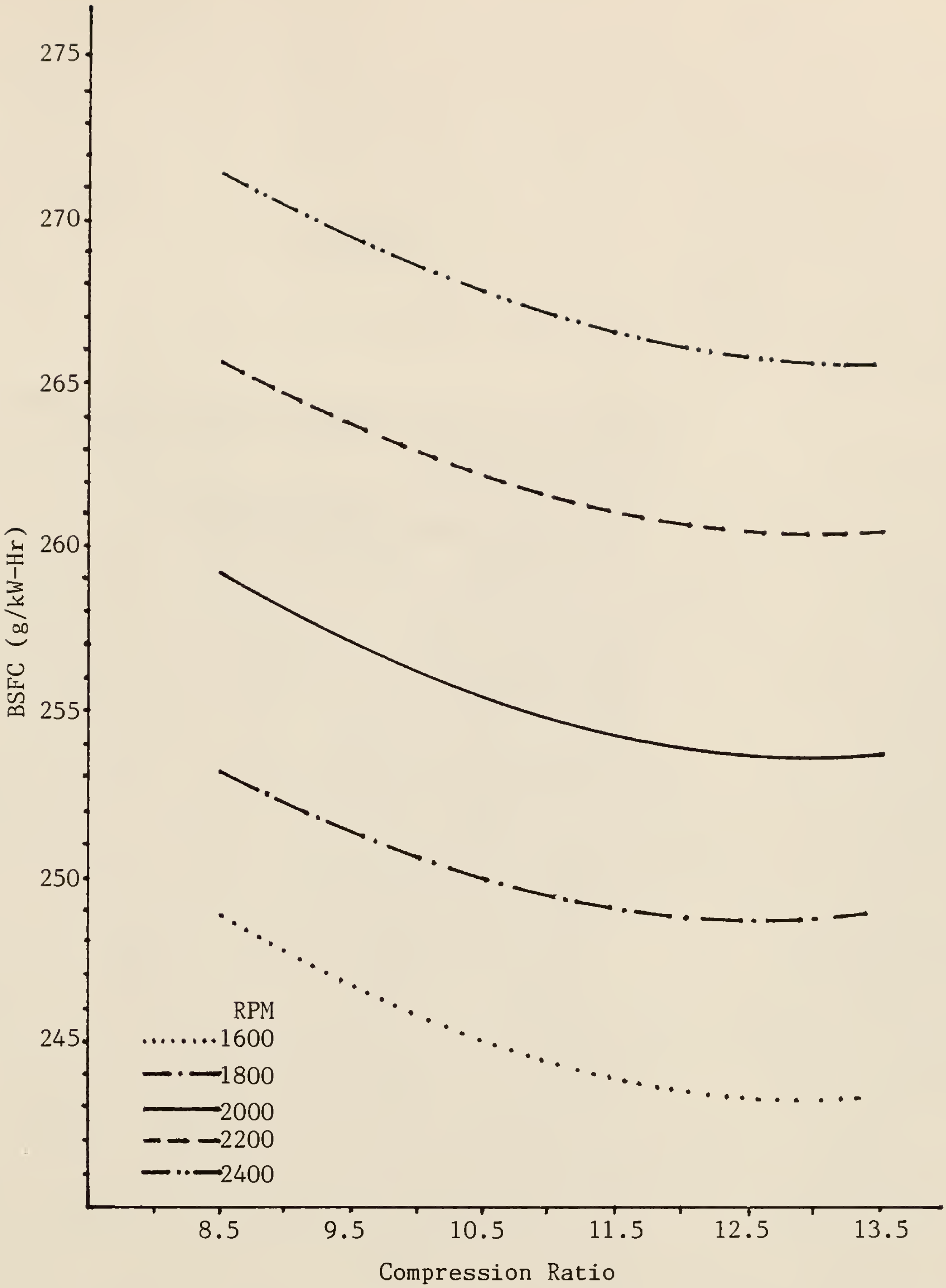


Figure 8c - Brake specific fuel consumption versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (early intake valve closing)

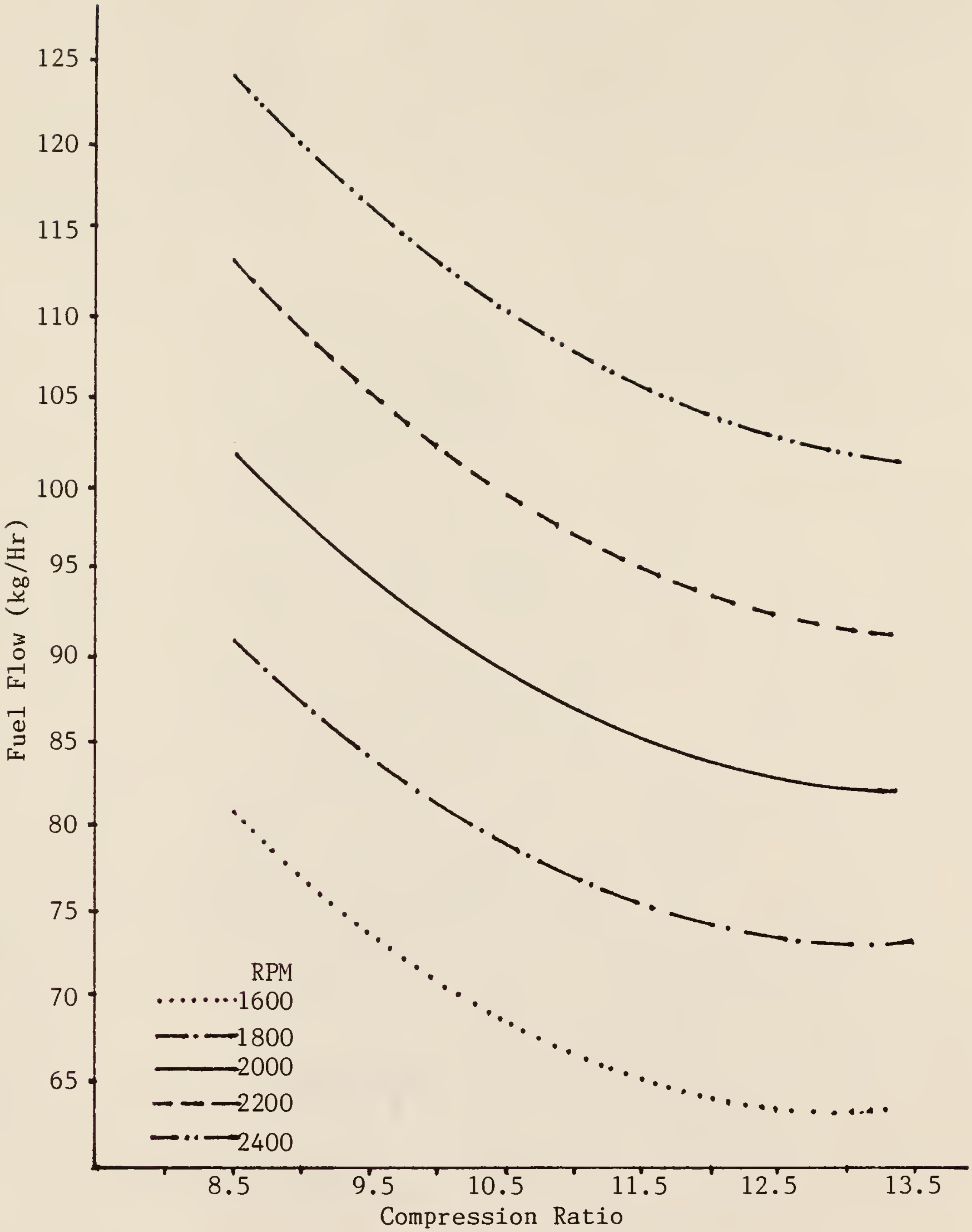


Figure 8d - Fuel flow versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (early intake valve closing)

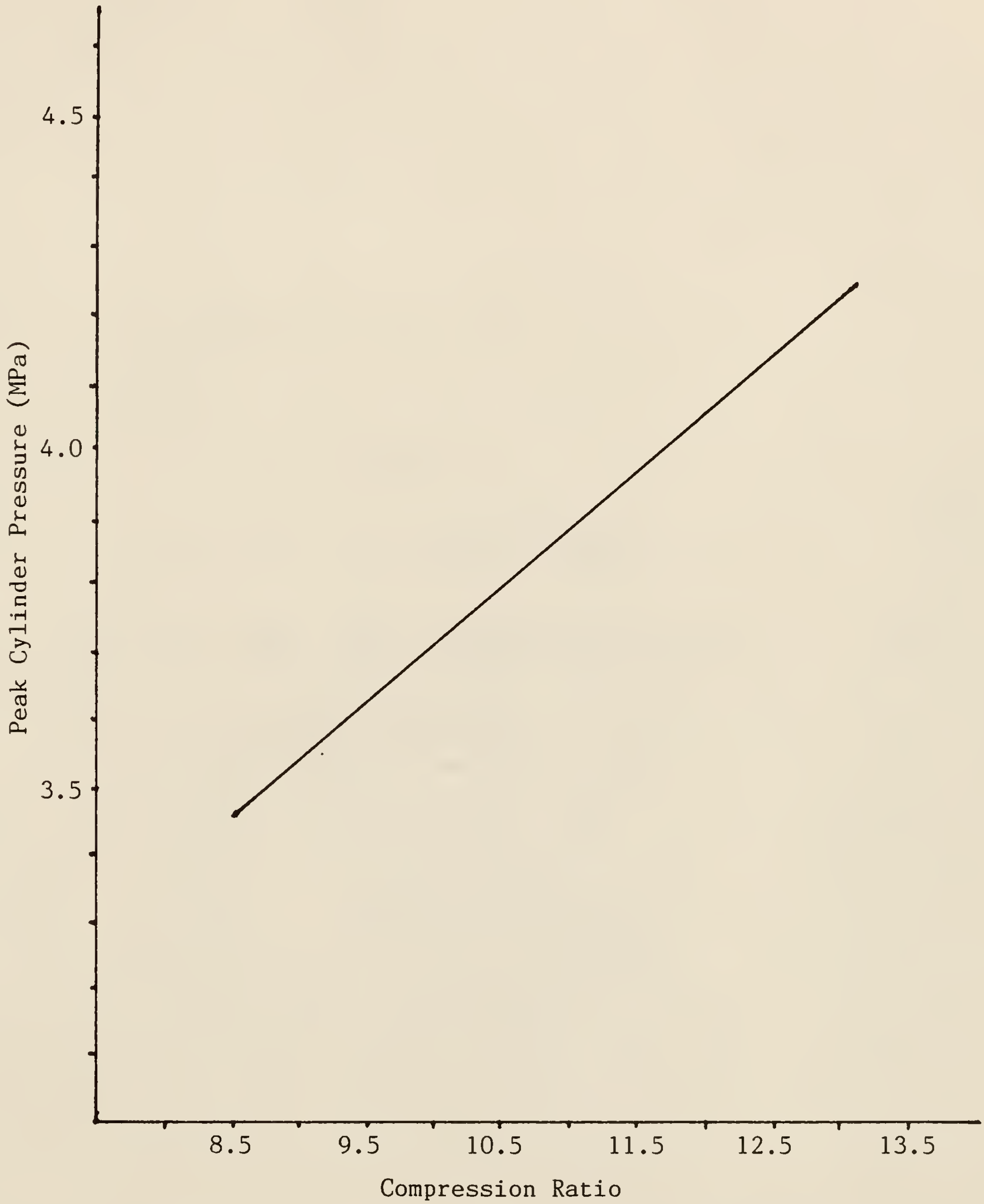


Figure 8e - Peak cylinder pressure versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (early intake valve closing)

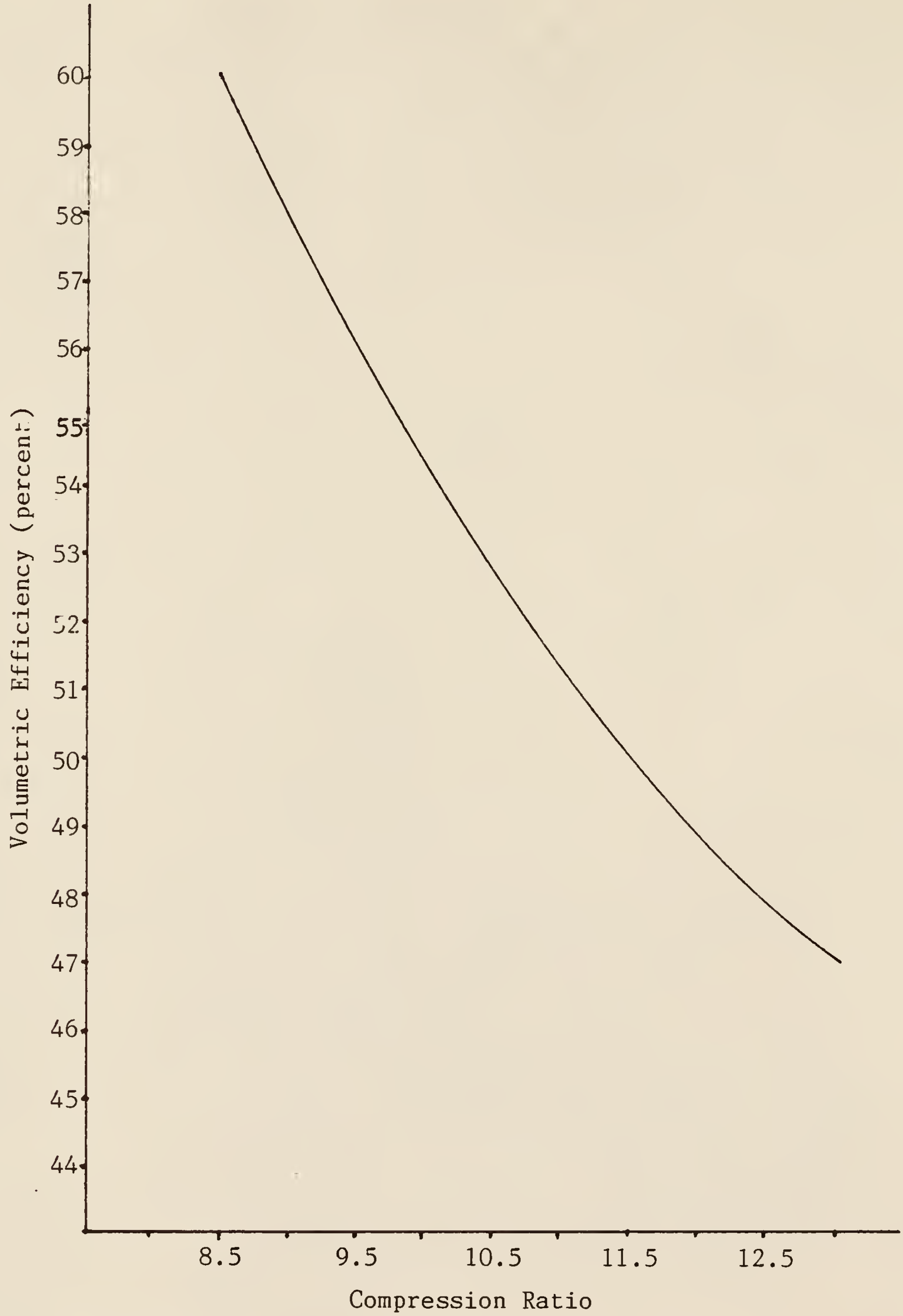


Figure 8f - Volumetric efficiency versus compression ratio for a Chevy 5.74 liter with matched ignition pressures. (early intake valve closing)

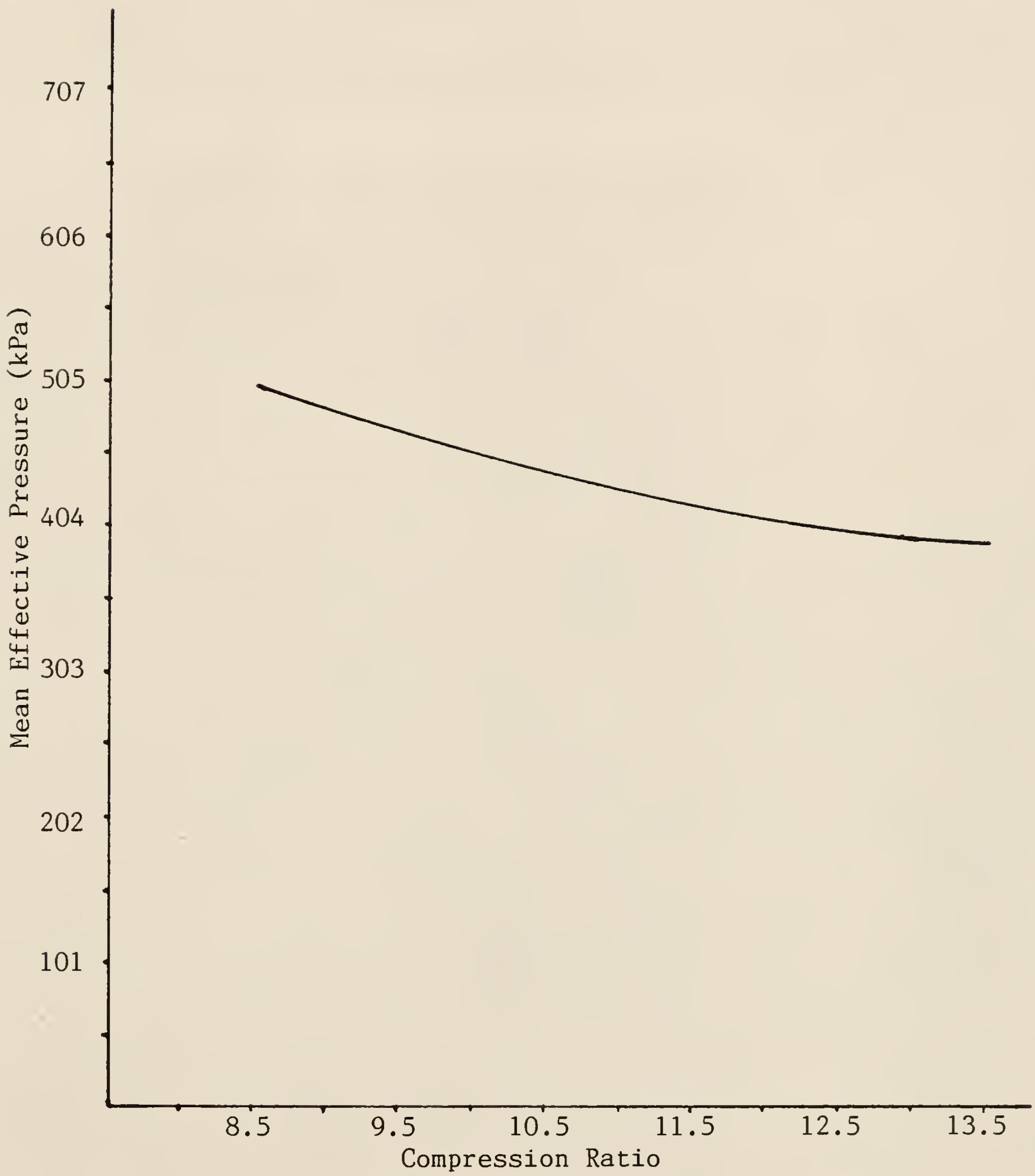


Figure 8g - Mean effective pressure versus compression ratio for Chevy 5.74 liter with matched ignition pressures. (early intake valve closing)

Figures 9a-9c are plots to determine the optimum exhaust valve opening. As before, the optimum time is 20 degrees BBDC. Table 3 shows the results of the proposed arrangement with early intake closing. Figure 10 is a plot of the P-V diagram of this cycle compared to the reference cycle.

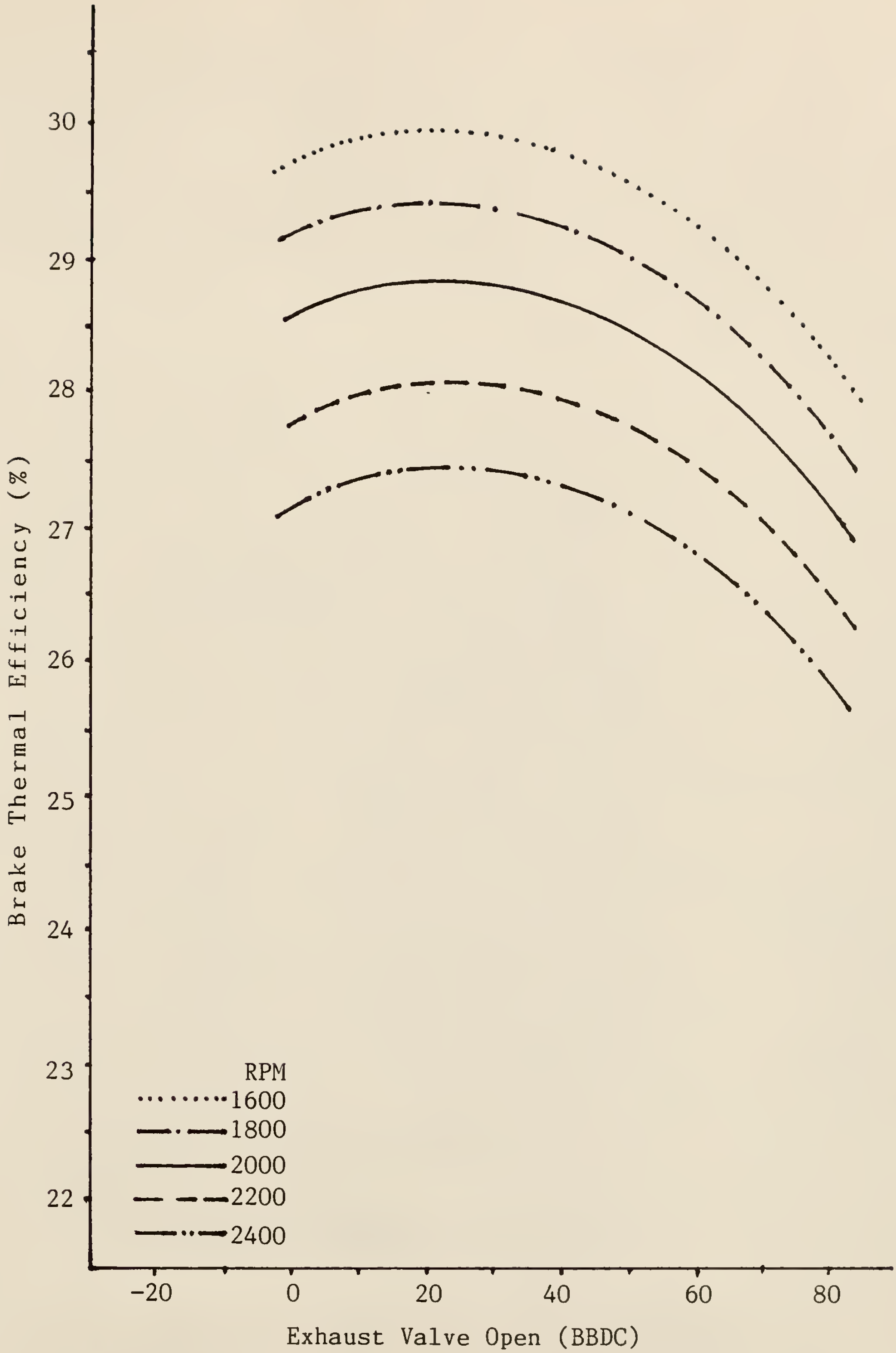


Figure 9a - Efficiency versus exhaust opening for a Chevy 5.74 liter with CR=12.5 and matched ignition pressures. (early intake valve closing)

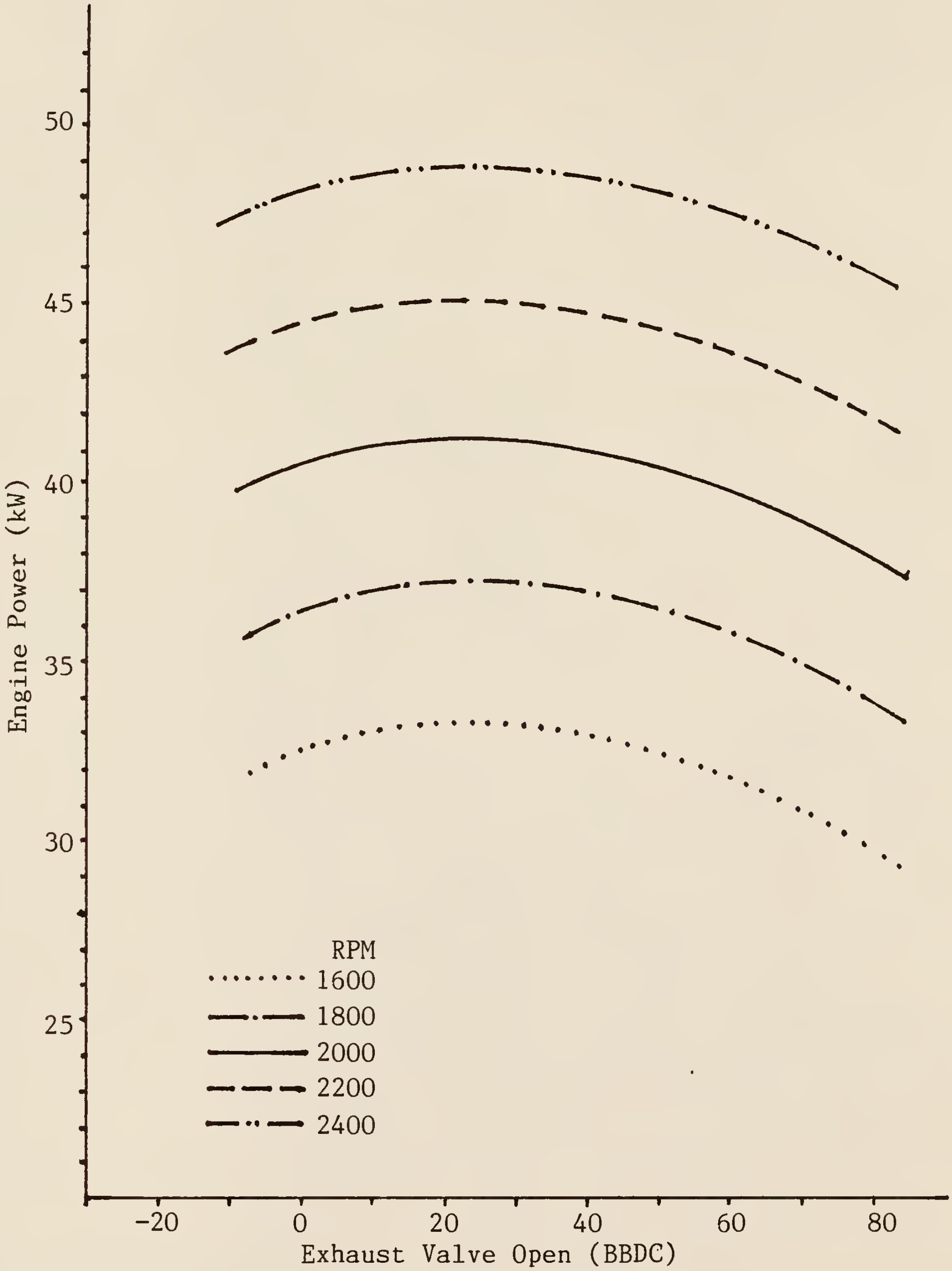


Figure 9b - Engine power versus exhaust opening for a Chevy 5.74 liter with CR=12.5 and matched ignition pressures. (early intake valve closing)

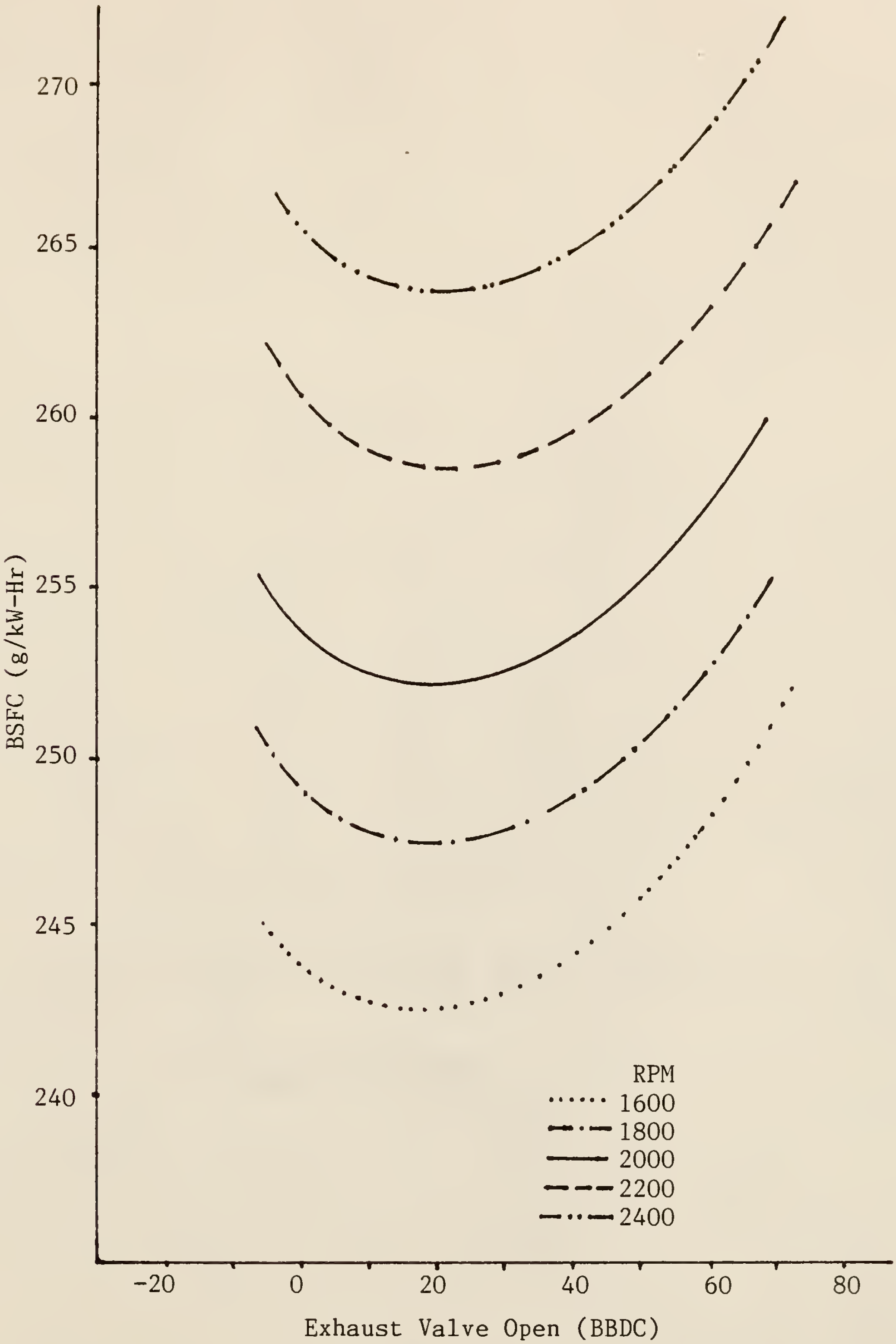


Figure 9c - Brake specific fuel consumption versus exhaust valve opening for a Chevy 5.74 liter with CR=12.5 and matched ignition pressures. (early intake valve closing)

TABLE 3

Computer generated performance data for a Chevy 5.74 liter engine operating at full-throttle, same ignition pressures as part-throttle case, and standard atmospheric conditions. CR=12.5, A/F=2.2, TM=300 K, and Methane Fuel.

RPM	SA DEG BTDC	MEAN PISTON SPEED (m/s)	VALVE TIMING EXH OPEN DEG BBDC	VALVE TIMING INT CLOSE DEG ABDC	MEP (kPa)	kW	BSFC (g/kW-Hr)	FF (kg/Hr)	VOL EFF (%)	PEAK PRESSURE (MPa)	EFF (%)
1600	27.2	4.71	20.0	-94.5	427	32.7	242	63	47	4.0	29.8
1700	27.6	5.01	20.0	-93.5	429	34.8	244	68	47	4.0	29.5
1800	28.0	5.30	20.0	-92.5	428	36.8	247	73	48	4.0	29.2
1900	28.4	5.60	20.0	-91.5	428	38.8	249	78	48	4.0	29.0
2000	28.9	5.89	20.0	-90.5	428	40.9	252	82	48	4.0	28.7
2100	29.3	6.19	20.0	-89.5	428	42.9	254	87	49	4.1	28.4
2200	29.7	6.48	20.0	-88.5	425	44.7	258	92	49	4.1	28.0
2300	30.1	6.78	20.0	-87.5	425	46.7	260	98	50	4.1	27.7
2400	30.5	7.07	20.0	-86.5	424	48.6	263	102	50	4.2	27.4
2500	30.9	7.37	20.0	-85.5	421	50.3	267	108	51	4.2	27.0

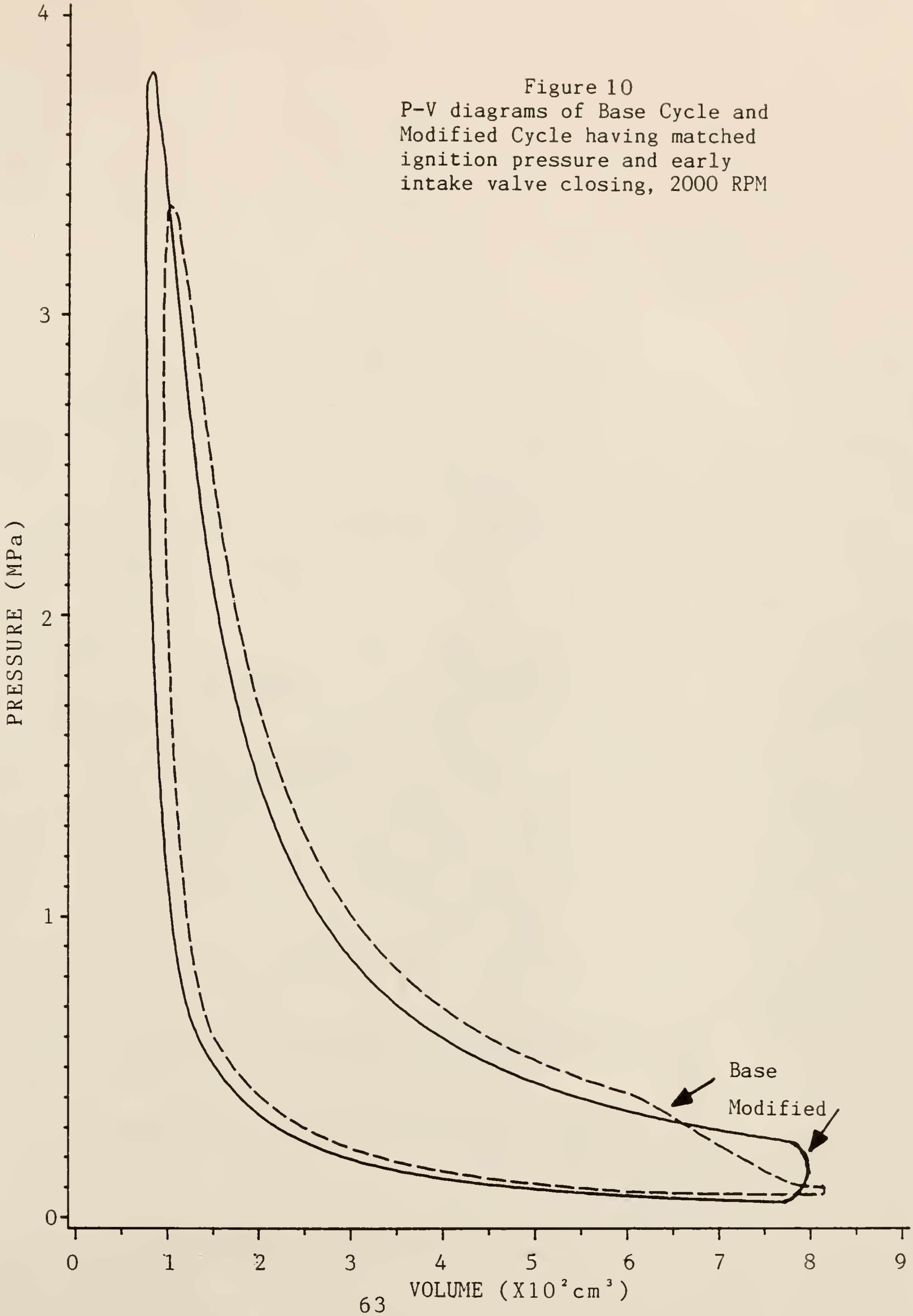


Figure 10
P-V diagrams of Base Cycle and Modified Cycle having matched ignition pressure and early intake valve closing, 2000 RPM

5.3 Matched power output, late intake valve closing:

The purpose of this segment in the analysis is to determine the advantage of implementing the more-complete-expansion idea to an actual irrigation pump arrangement. Irrigation pump gear heads usually require a specified amount of power at a certain speed. It is essential that the modified engine produce the required pumping conditions. These conditions are met by matching the predicted power output from the modified model to those of the reference model which was operated at 178 mm Hg. This also requires that the engine speed (RPM) be equivalent for an accurate prediction.

The procedure used for matching the power output is much the same as the two previous cases. The match was achieved by varying the intake valve closing while maintaining the exhaust valve opening at its optimum time of 20 degrees BBDC.

Simulations were produced to examine the effects of changing the compression ratio and intake valving on the performance information. In this case the intake valve was closed late allowing a large portion of the freshly inducted charge to be pushed out of the cylinder resulting in a reduced power output. The results of the simulations can be seen in Figures 11a-11e.

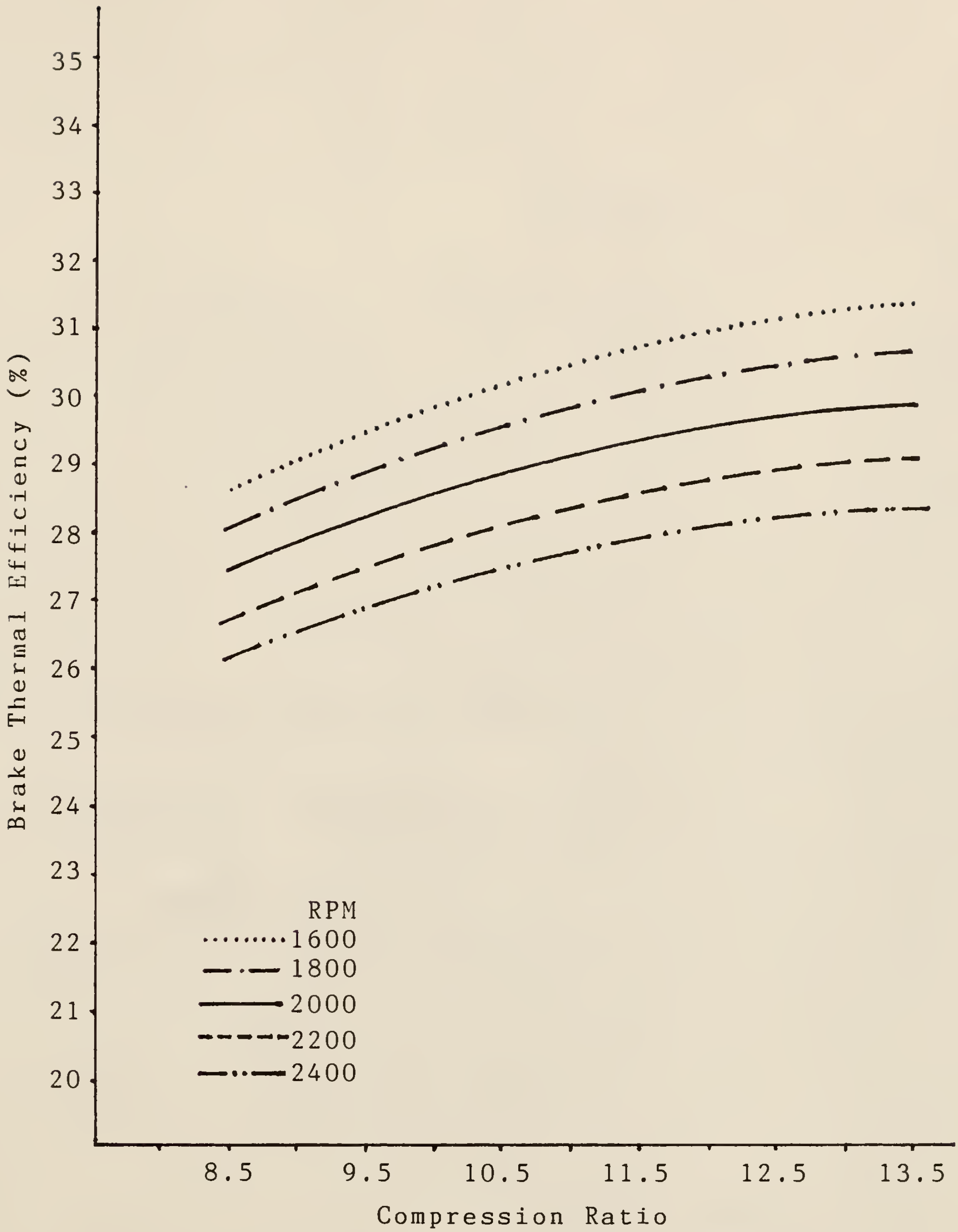


Figure 11a - Efficiency versus compression ratio for a Chevy 5.74 liter with matched power output. (late intake valve closing)

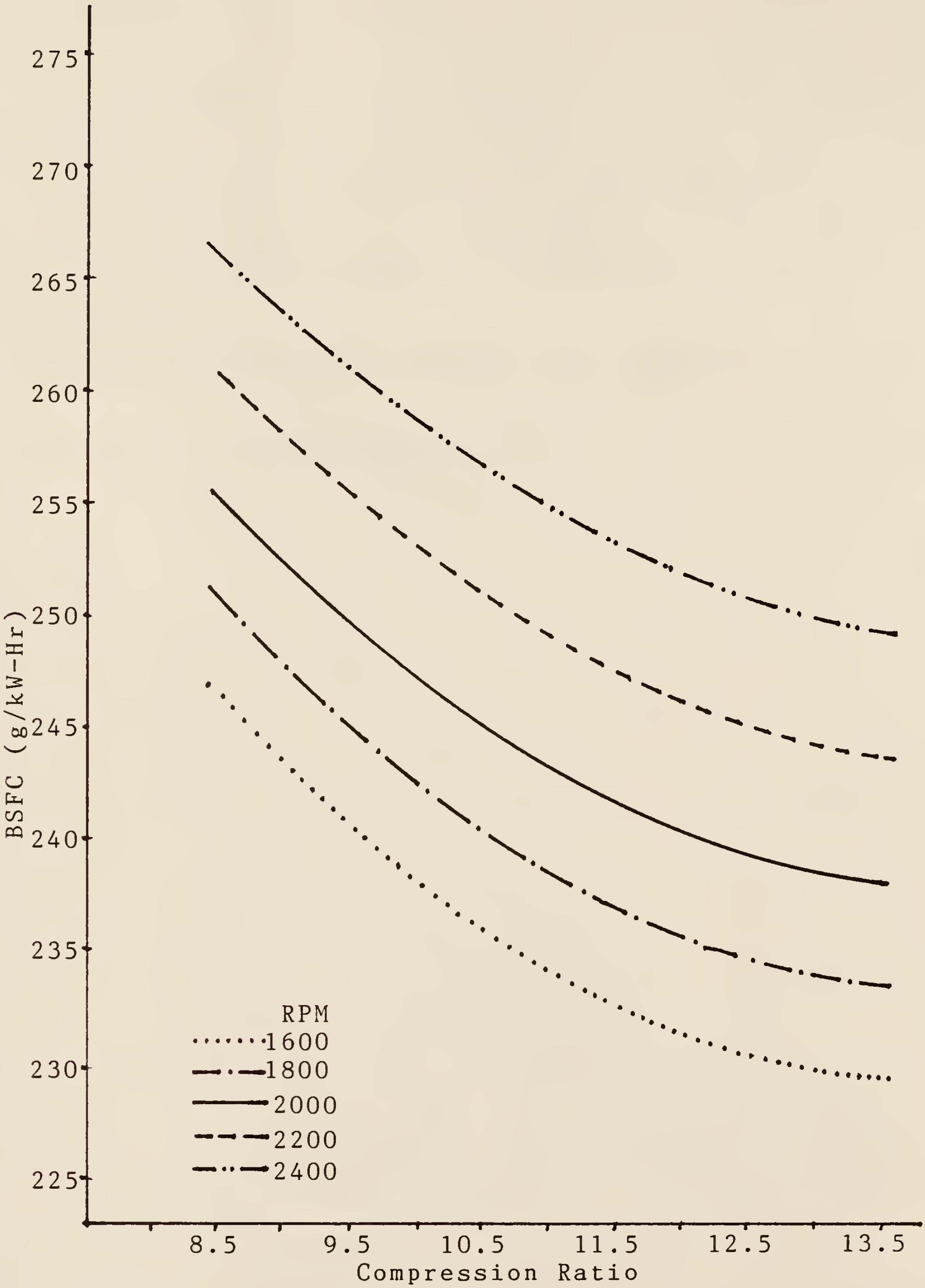


Figure 11b - Brake specific fuel consumption versus compression ratio for a Chevy 5.74 liter with matched power output. (late intake valve closing)

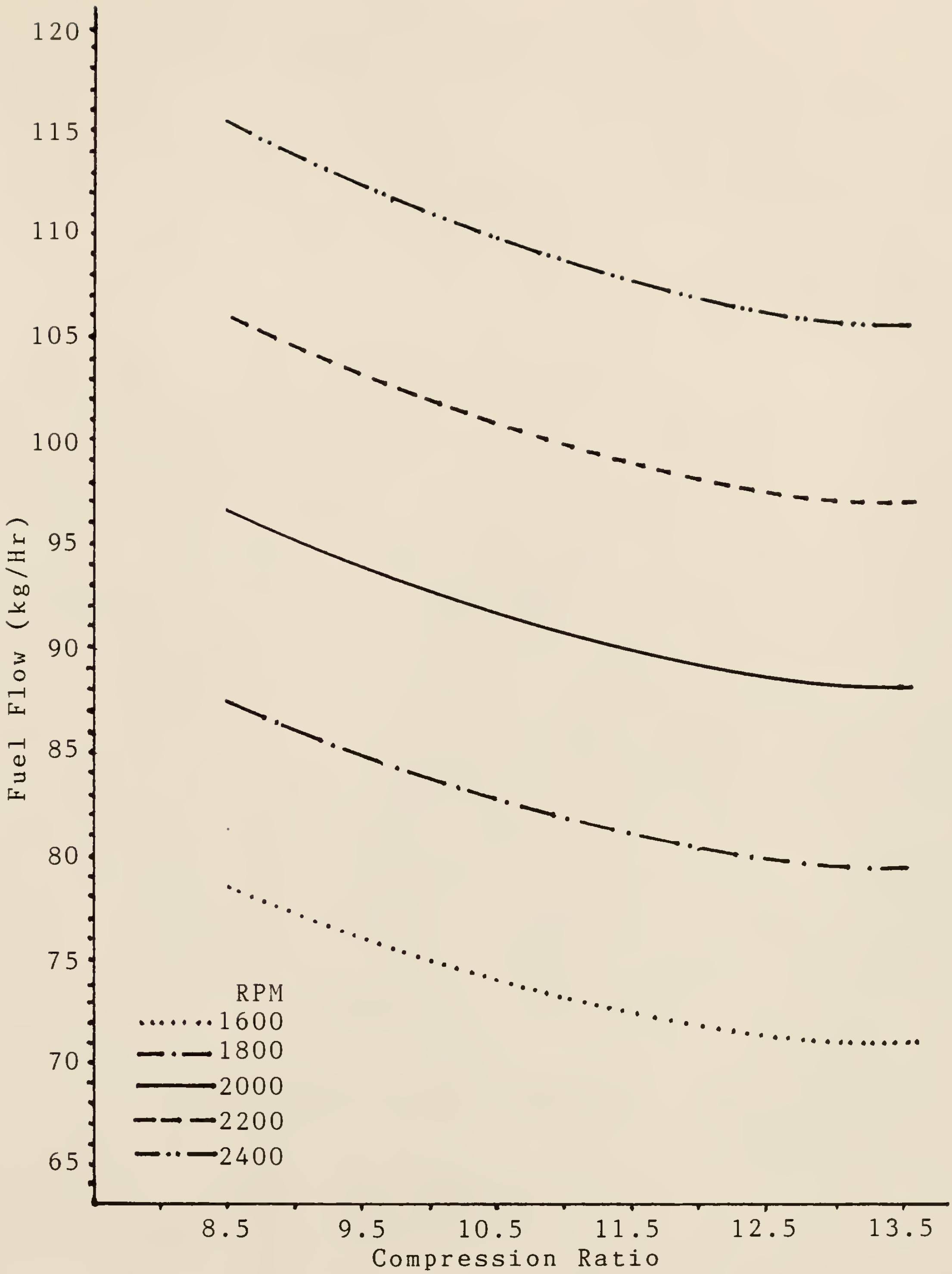


Figure 11c - Fuel flow versus compression ratio for a Chevy 5.74 liter with matched power output. (late intake valve closing)

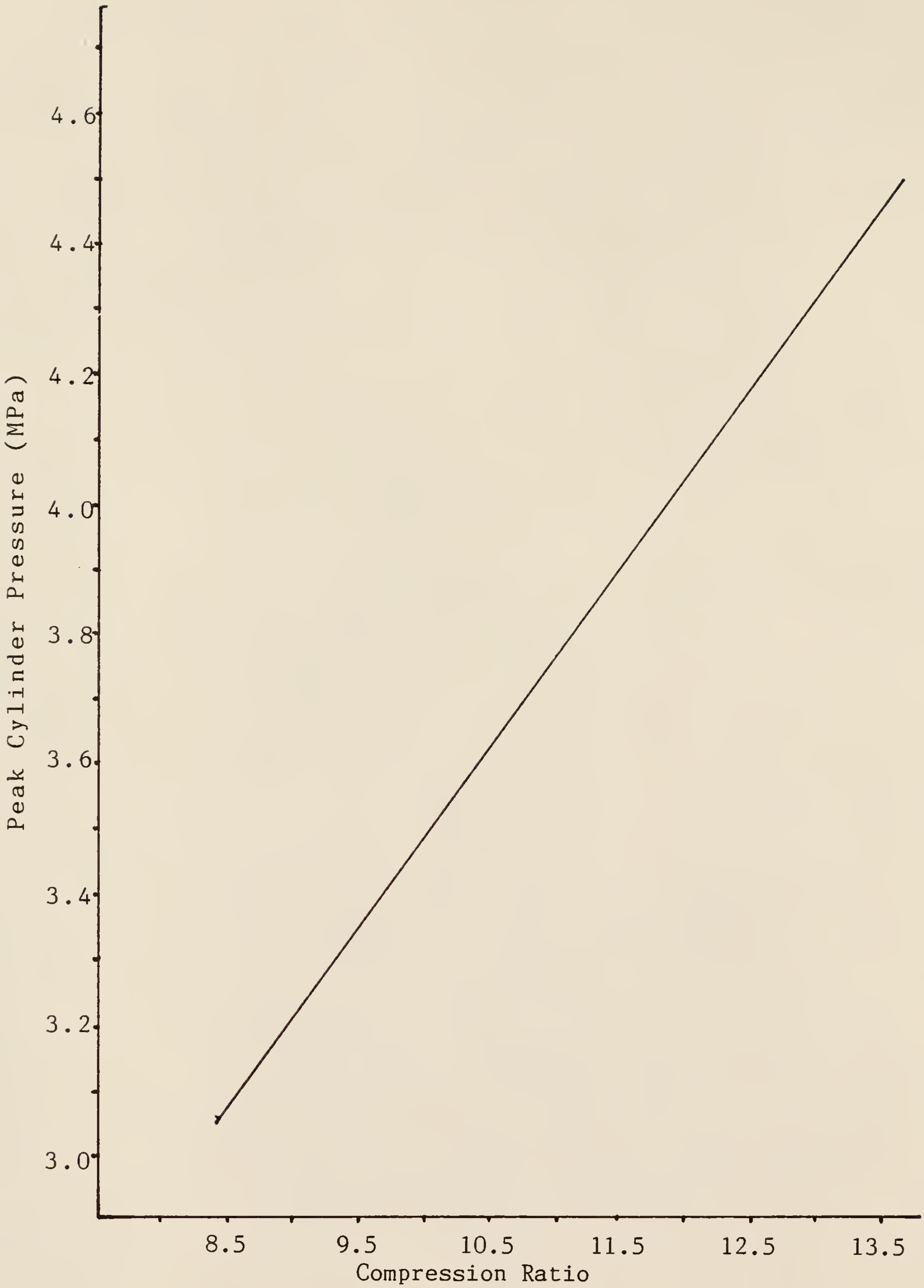


Figure 11d- Peak cylinder pressure versus compression ratio for a Chevy 5.74 liter with matched power output. (late intake valve closing)

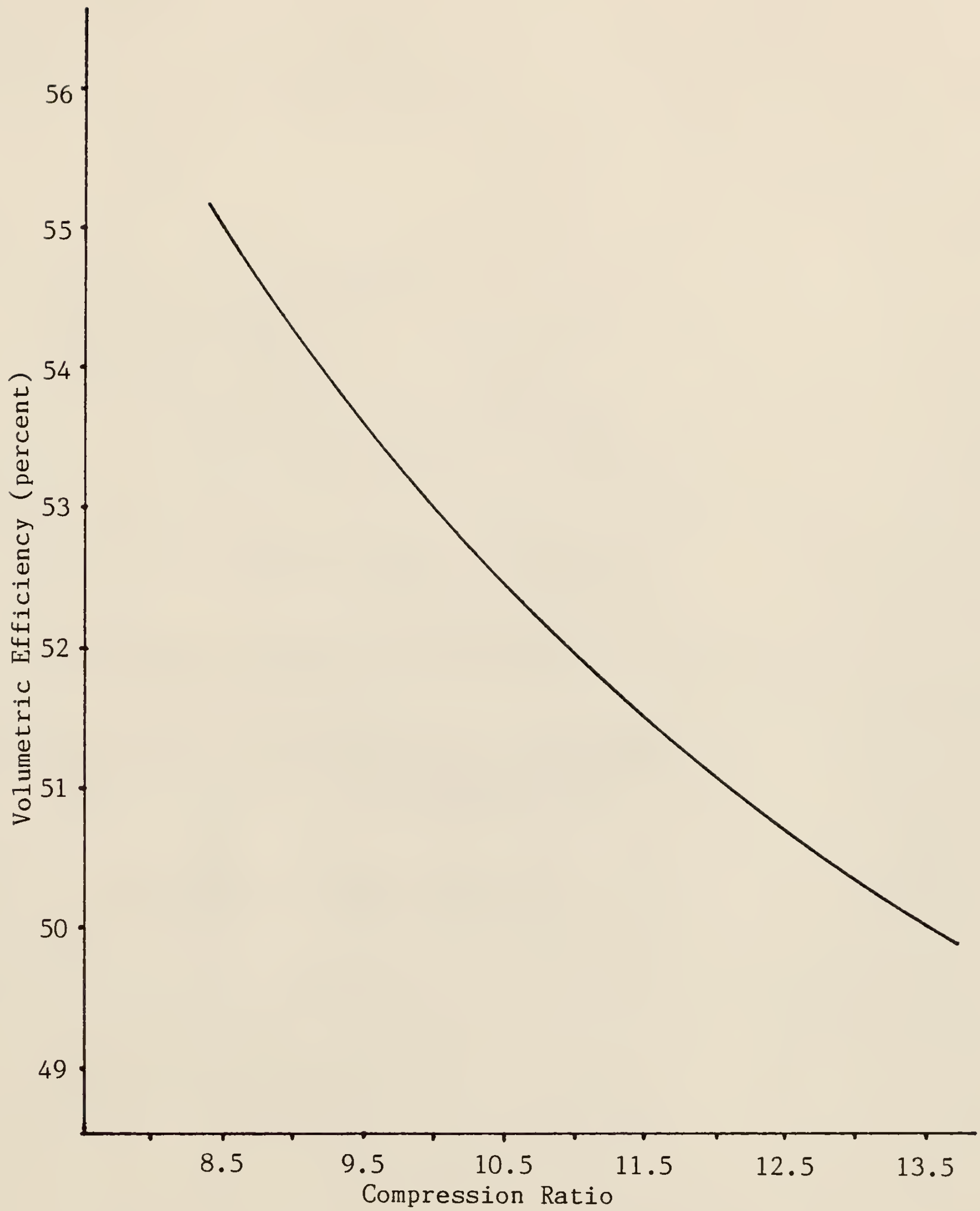


Figure 11e- Volumetric efficiency versus compression ratio for a Chevy 5.74 liter with matched power output. (late intake valve closing)

From these figures it is observed that a significant increase in engine performance cannot be attained by increasing the compression ratio beyond 12.5:1. This condition was chosen as the maximum performance arrangement for matching power output. Table 4 shows the performance data for a simulated engine with matched power output, late intake valve closing and 12.5:1 compression ratio. The information produced will be useful in the comparison between the modified engine and the reference engine. Also, a P-V diagram of the modified cycle and reference cycle is included in Figure 12 to demonstrate the changes in the two cycles.

TABLE 4

Computer generated performance data for a Chevy 5.74 liter engine operating at full-throttle, same power output (kW) as part-throttle case, and standard atmospheric conditions. CR=12.5, A/F=2.2, TM=300 K, and Methane Fuel.

RPM	SA DEG BTDC	MEAN PISTON SPEED (m/s)	VALVE TIMING EXH OPEN DEG BBDC	INT CLOSE DEG ABDC	MEP (kPa)	kW	BSFC (g/kW-Hr)	FF (kg/Hr)	VOL EFF (%)	PEAK PRESSURE (MPa)	EFF (%)
1600	27.2	4.71	20.0	116.0	464	35.8	234	67	49	4.0	30.9
1700	27.6	5.01	20.0	116.3	464	38.0	236	72	49	4.1	30.6
1800	28.0	5.30	20.0	116.7	464	39.8	239	76	50	4.1	30.2
1900	28.4	5.60	20.0	117.0	464	41.9	241	81	50	4.1	29.9
2000	28.9	5.89	20.0	117.3	454	43.5	244	86	50	4.1	29.6
2100	29.3	6.19	20.0	117.7	454	45.6	247	90	51	4.1	29.2
2200	29.7	6.48	20.0	118.0	454	47.4	251	95	51	4.1	28.8
2300	30.1	6.78	20.0	118.3	444	49.2	254	100	51	4.1	28.4
2400	30.5	7.07	20.0	118.7	444	50.8	258	105	51	4.2	28.0
2500	30.9	7.37	20.0	119.0	434	52.4	262	110	52	4.2	27.6

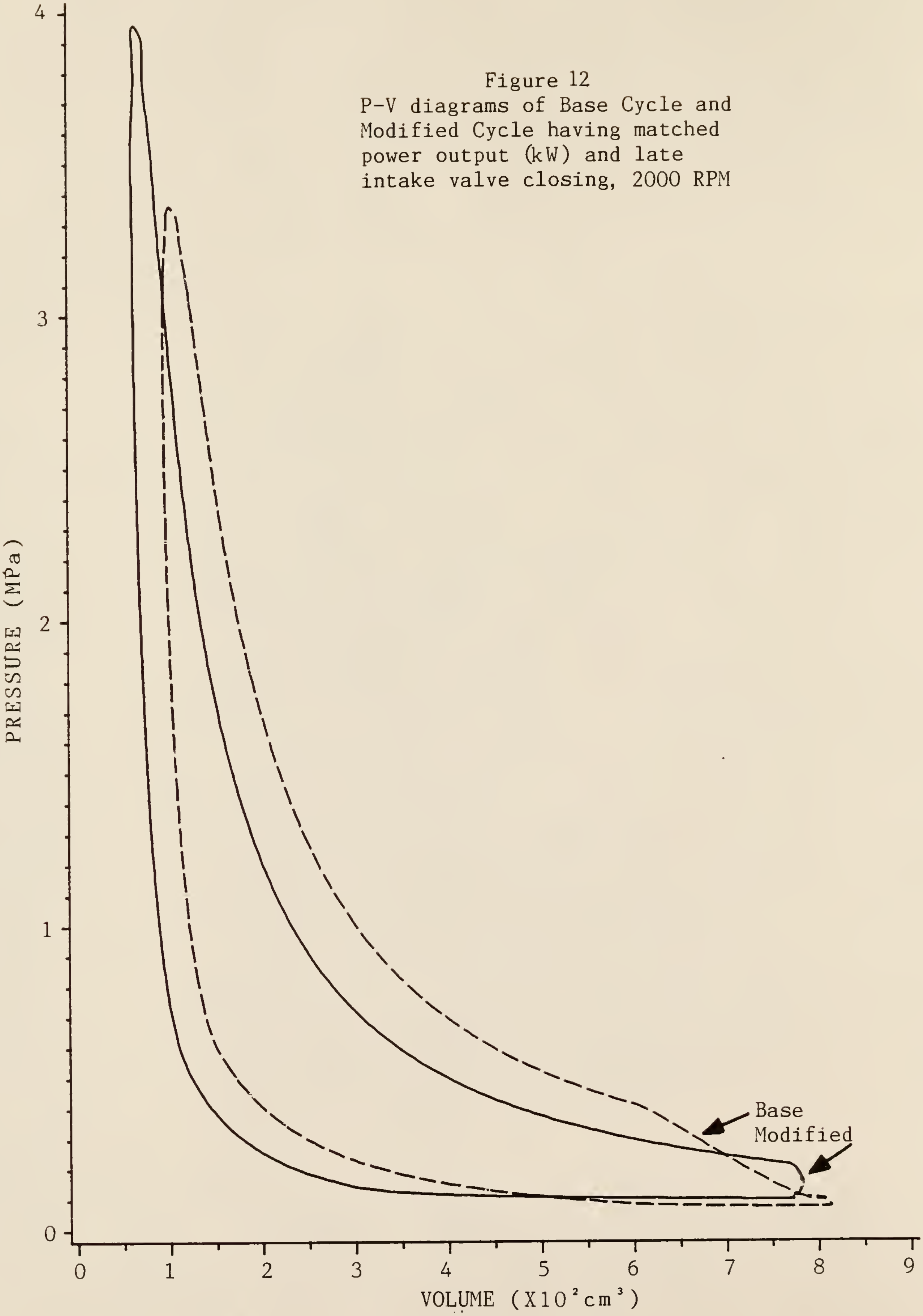


Figure 12
P-V diagrams of Base Cycle and Modified Cycle having matched power output (kW) and late intake valve closing, 2000 RPM

5.4 Matched power output, early intake closing:

This section involves determining the effects on engine performance when closing the intake valve early while holding the power output the same as the reference case. The method used in this case is the same as before and the plotted results are shown in Figures 13a-13e. These figures show that the best compression ratio for maximum performance under matched power output constraints is 12.5:1. This value is used to produce Table 5 which lists the performance data resulting from the change. A P-V diagram is included in Figure 14 to show the modifications between the reference and modified cases.

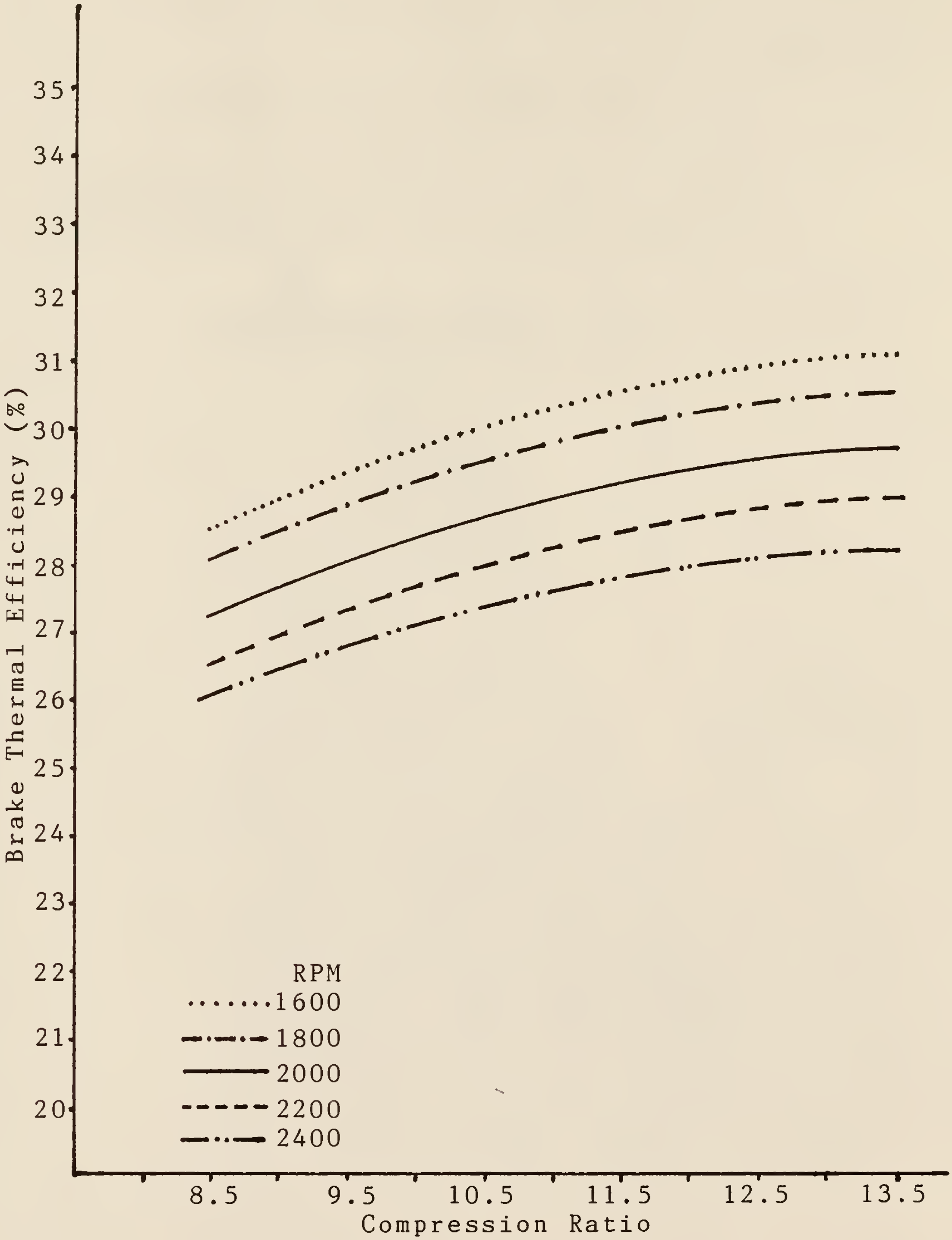


Figure 13a - Efficiency versus compression ratio for a Chevy 5.74 liter with matched power output. (early intake valve closing)

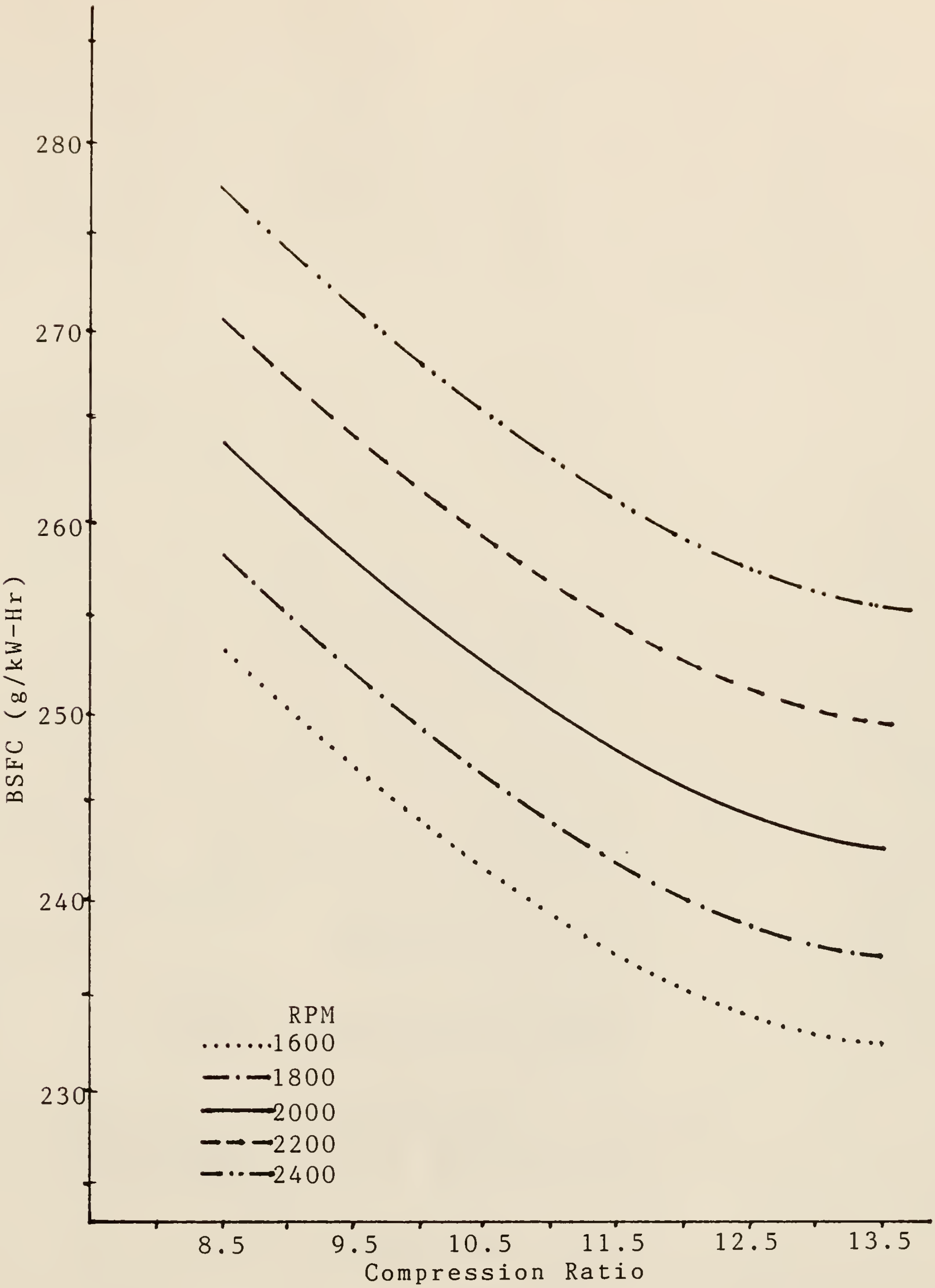


Figure 13b - Brake specific fuel consumption versus compression ratio for a Chevy 5.74 liter with matched power output. (early intake valve closing)

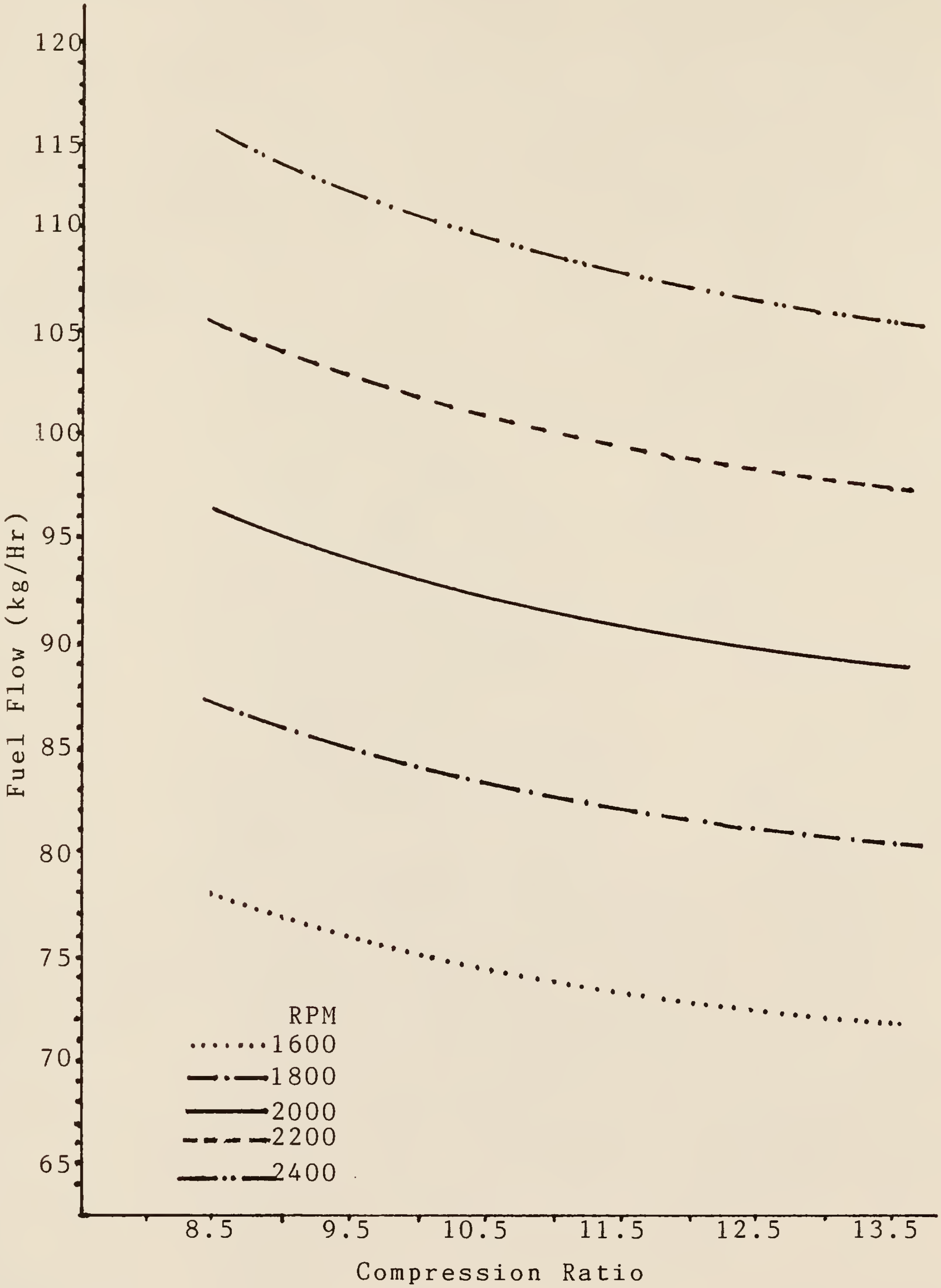


Figure 13c - Fuel flow versus compression ratio for a Chevy 5.74 liter with matched power output. (early intake valve closing)

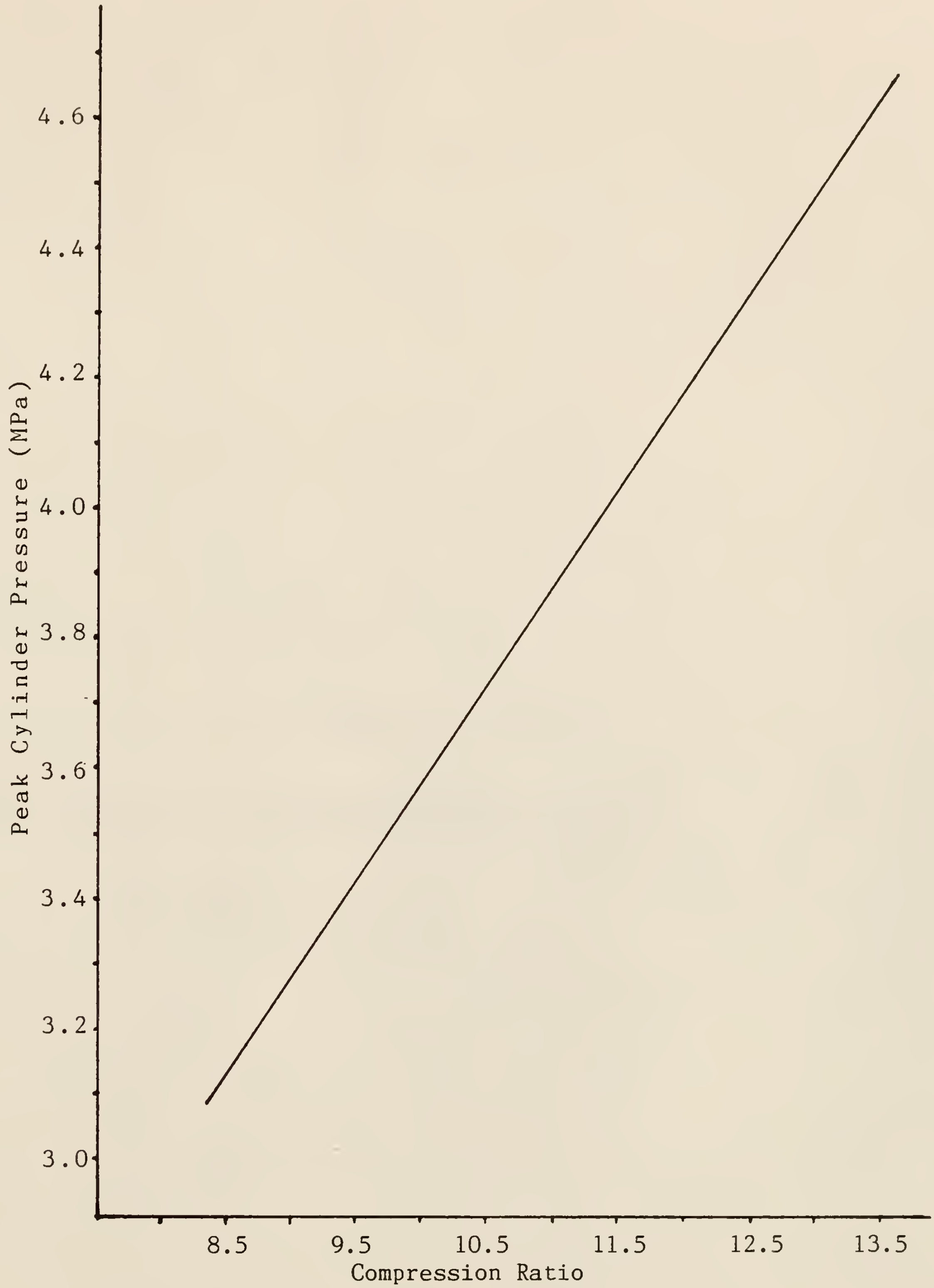


Figure 13d- Peak cylinder pressure versus compression ratio for a Chevy 5.74 liter with matched power output. (early intake valve closing)

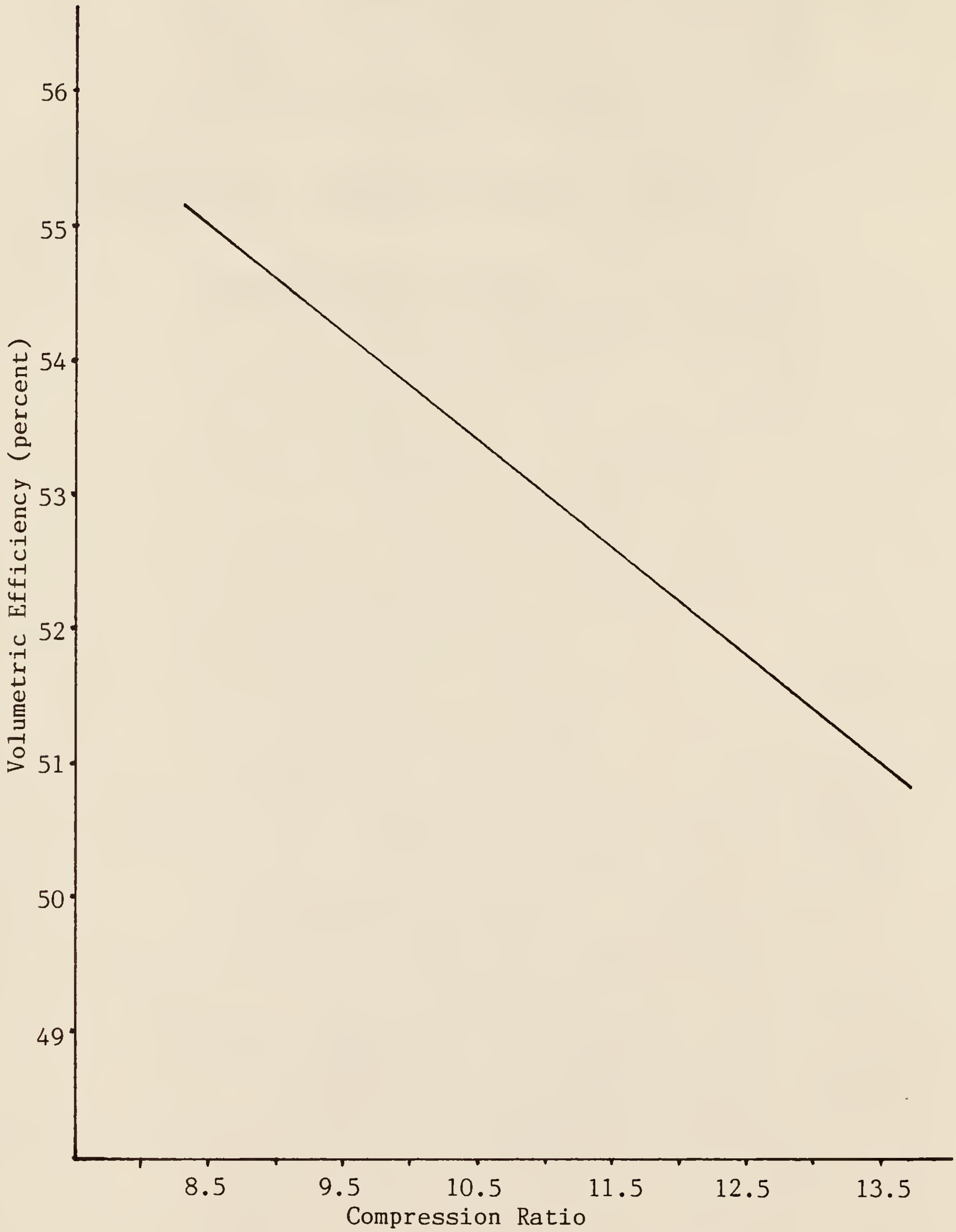


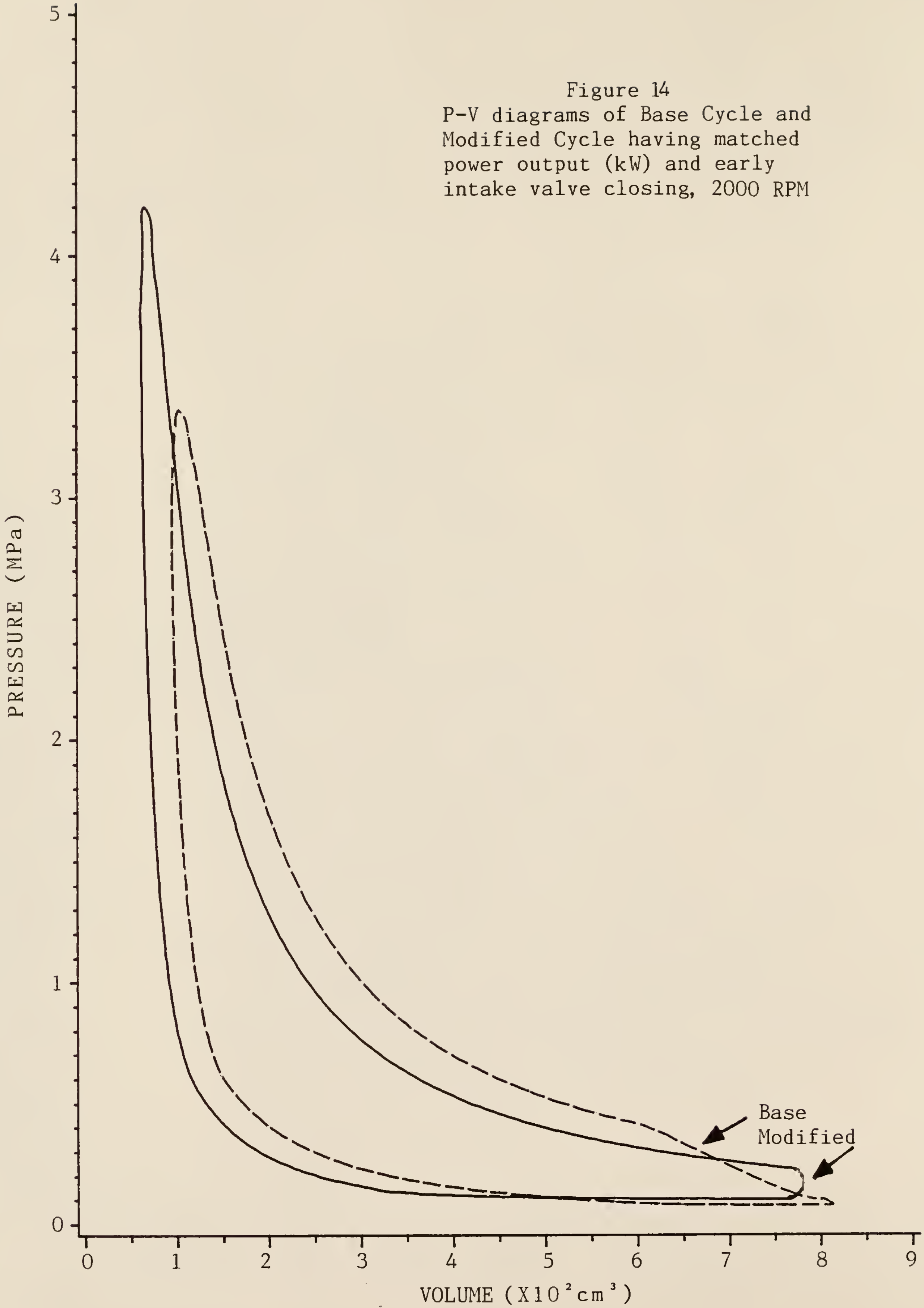
Figure 13e- Volumetric efficiency versus compression ratio for a Chevy 5.74 liter with matched power output. (early intake valve closing)

TABLE 5

Computer generated performance data for a Chevy 5.74 liter engine operating at full-throttle, same power output (kW) as part-throttle case, and standard atmospheric conditions. CR=12.5, A/F=2.2, TM=300 K, and Methane Fuel.

RPM	SA DEG BTDC	MEAN PISTON SPEED (m/s)	VALVE TIMING EXH OPEN DEG BBDC	VALVE TIMING INT CLOSE DEG ABDC	MEP (kPa)	kW	BSFC (g/kW-Hr)	FF (kg/Hr)	VOL EFF (%)	PEAK PRESSURE (MPa)	EFF (%)
1600	27.2	4.71	20.0	-90.0	465	36.0	236	68	50	4.2	30.6
1700	27.6	5.01	20.0	-89.1	465	37.9	239	72	50	4.2	30.2
1800	28.0	5.30	20.0	-88.2	465	39.8	241	77	50	4.2	30.0
1900	28.4	5.60	20.0	-87.3	465	42.1	244	82	51	4.2	29.6
2000	28.9	5.89	20.0	-86.4	455	44.0	247	87	51	4.2	29.3
2100	29.3	6.19	20.0	-85.6	455	45.8	250	92	51	4.2	28.9
2200	29.7	6.46	20.0	-84.7	455	47.7	252	96	52	4.2	28.6
2300	30.1	6.78	20.0	-83.8	444	49.5	256	101	52	4.2	28.2
2400	30.5	7.07	20.0	-82.9	444	50.9	261	106	52	4.2	27.7
2500	30.9	7.37	20.0	-82.0	434	52.5	264	111	52	4.3	27.3

Figure 14
P-V diagrams of Base Cycle and Modified Cycle having matched power output (kW) and early intake valve closing, 2000 RPM



6.0 DISCUSSION OF IMPLEMENTATION OF A MORE-COMPLETE-EXPANSION CYCLE

The reason for this investigation was to determine the effects of the more-complete-expansion cycle and the feasibility of utilizing such a cycle in the application of irrigation water pumping. The results from the present study appear promising. It must be noted that an even greater advantage can be achieved in an engine where the displacement volume is much greater than the portion used for output. In this case the more-complete-expansion cycle will not only use the entire displacement more efficiently, but can also cut fuel consumption in half while reducing power output by 75%. The same is true for the engine used in this study if it were operated at half-throttle or below rather than at three-forths throttle.

Perhaps the question remains as to why the Chevy 5.74 liter engine was chosen for this research. As mentioned before, this is a popular engine with many modified parts available. If experimental tests are conducted at Kansas State University in the future, this engine will be used to minimize experimental costs. Should the more-complete-expansion cycle warrant further development beyond experimental testing on an actual Chevy 5.74 liter engine, a more realistic engine can be purchased and modified.

6.1 Comparison of simulated data to the reference case:

Table 6 lists the percent change in some of the important performance parameters that resulted from the engine modification. As can be seen in the table, all of the modified engines exhibited approximately the same power output as the reference engine. The modification also caused a 13-20% increase in the thermal efficiency which was expected due to the increased expansion/compression ratio.

The results shown in Table 6 are necessary to determine the effects of the more-complete-expansion cycle. Each of these cycles will be examined in the coming sections to determine the cost effectiveness of implementing the modifications.

TABLE 6

Comparison of the modified cases to the reference case for determination of cost effectiveness.

Case 1: matched ign. press., late int. closing
Case 2: matched ign. press., early int. closing
Case 3: matched kW, late int. closing
Case 4: matched kW, early int. closing

		Case 1	Case 2	Case 3	Case 4
% change in kW:					
RPM:	1600	7.0	9.4	0.0	0.0
	1800	6.1	8.3	0.0	0.0
	2000	7.9	7.0	0.0	0.0
	2200	6.9	6.1	0.0	0.0
	2400	5.9	4.6	0.0	0.0
% change in Eff:					
RPM:	1600	20.2	13.1	17.5	16.3
	1800	19.4	12.8	17.1	16.3
	2000	19.4	13.0	17.0	15.8
	2200	18.5	12.5	15.7	14.9
	2400	18.2	12.9	15.7	14.5
% change in FF:					
RPM:	1600	-11.1	-12.4	-15.8	-14.5
	1800	-11.0	-11.8	-15.4	-14.0
	2000	-10.2	-11.2	-14.5	-13.3
	2200	- 9.7	-10.8	-14.1	-13.1
	2400	-10.2	-10.4	-13.6	-12.6

6.2 Cost analysis of engine modification:

An important feature of implementing this idea is the cost of the modifications. In order to incorporate the more-complete-expansion cycle on an existing irrigation engine, the engine must be modified in the following ways. The compression ratio needs to be increased to the appropriate value and the cam must be replaced by a new one ground to the correct specifications.

The compression ratio is changed by planing the cylinder heads down to reduce the volume at top dead center. A price estimate of $26 per head was received from Billy Graham Performance Machine, Manhattan, Kansas. If the compression ratio required cannot be attained by planing the cylinder heads, a set of high compression pistons can be purchased at a cost of $400.

It would be necessary for the new cam to be ground by a professional tool and part shop. A few companies were contacted to determine the price of the cam. (See Appendix II) Each of the shops estimated that a new master would be needed to mill the correct profiles. The cost of this procedure usually runs $340-350 per profile (i.e. intake valving and exhaust valving) and is only incurred once if several cams are ordered. If the company happened to have a satisfactory master in stock this cost would be eliminated. The other costs include the cast iron core at $40-50 and the actual grinding at $250.

Another cost created by the modification is the replacement of the gaskets. The gaskets needed for the Chevy 5.74 liter engine are priced at about $50. Therefore, the complete cost of the modification with the two required masters and high compression pistons would be approximately $1500. If the tool shop had the masters in stock, the cost would be reduced to $800. Since mechanic rates vary, a generous labor cost of $800 will be assumed. Therefore, the overall cost of the modification would be between approximately $1600-2300.

Table 7 shows the amount of money saved per year in fuel for each of the modified cases. An irrigation engine usually operates an estimated 1000-2000 hours per year. If the engine is run for 2000 hours per year, the amount of money saved would be significant and could pay for the modification in less than a year. Whichever modified engine is used the payback period would be less than two years.

On the basis of this analysis, the use of the

TABLE 7

Fuel cost saved per year by implementing the more complete expansion cycle. Fuel costs based on $0.097 per Kg of methane gas.

	Case 1	Case 2	Case 3	Case 4
RPM: 1600				
Hr/yr operation:				
1000	$ 854	$ 984	$1220	$1116
1500	1281	1476	1831	1673
2000	1709	1968	2440	2232
RPM: 1800				
Hr/yr operation:				
1000	$ 960	$1043	$1342	$1220
1500	1438	1565	2013	1831
2000	1918	2086	2685	2440
RPM: 2000				
Hr/yr operation:				
1000	$ 994	$1064	$1412	$1290
1500	1491	1596	2118	1935
2000	1987	2128	2825	2580
RPM: 2200				
Hr/yr operation:				
1000	$1046	$1100	$1516	$1412
1500	1569	1650	2275	2119
2000	2092	2200	3033	2825
RPM: 2400				
Hr/yr operation:				
1000	$1202	$1250	$1603	$1482
1500	1804	1875	2404	2225
2000	2404	2500	3206	2965

more-complete-expansion cycle is warranted. It is also recommended that this idea be pursued through the experimental testing stage to determine the actual costs and advantages. However, different engines must be examined individually by a computer simulation or by experimentation to correctly incorporate a particular engine's features.

7.0 REFERENCES

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8.0 APPENDIX I

Engine characteristic parameters for the reference Chevy 5.74 liter engine.

Fuel	Methane
Bore, cm	10.16
Stroke, cm	8.84
Connecting Link, cm	14.50
Compression Ratio	8.50
RPM	1600-2500
Intake Man. Pressure, kPa	76.00
Intake Man. Temperature, K	300.00
Exhaust Man. Pressure, kPa	101.40
Lift, cm	1.14
Valve Thickness, cm	0.64
Valve Seat Angle, deg	45.00
Valve Diameter (small)	
Intake, cm	4.45
Exhaust, cm	3.81
Valve Timing, deg	
Intake : Open (BTDC)	0.00
Close (ABDC)	76.00
Exhaust: Open (BBDC)	77.00
Close (ATDC)	0.00

APPENDIX II

List of cam grinding companies used in cost estimates for modifications.

Ed Iskenderian, Racing Cams
16020-T.S. Broadway
Gardena, California
(213) 770-0930

Andrew Products
Chicago, Illinois 60618
(312) 992-4014

EONIC Inc.
466 East Hollywood
Detroit, Michigan 48203
(313) 893-8100

Appendix III

List of Variables used in Computer

Model and Complete Program Listing

LIST OF VARIABLES

		Units
AA	: fuel equation constant	
AEO	: max. flow area of exhaust valve	m^2
AIO	: max. flow area of intake valve	m^2
AREA	: inst. flow area for valving	m^2
AH(5)	: gas equation constants	
AL(5)	: gas equation constants	
B	: bore	cm
BB	: fuel equation constant	
BH(5)	: gas equation constants	
BL(5)	: gas equation constants	
CD	: inst. flow coefficient (drag)	
CM	: scaling factor for chemistry section	
CP	: inst. constant pressure specific heat	kJ/kmol-K
CPM	: constant pressure specific heat for mixture	kJ/kmol-K
CR	: compression ratio	
CPR	: reactant gas specific heat	kJ/kmol-K
CPX	: residual gas specific heat	kJ/kmol-K
CRIT	: critical pressure ratio	
CSUB	: subsonic flow equation	
CSUP	: supersonic flow equation	
CVR	: reactant gas specific heat	kJ/kmol-K
CUT	: divisor of increment equation	
C1	: constant (2*L/s)	
CH(5)	: gas equation constants	
CL(5)	: gas equation constants	
DN	: fraction of burned gas	
DP	: change in inst. pressure	kPa
DPH	: change in pressure due to heat transfer	kPa
DPH1	: pressure change of hot gases	kPa
DPH2	: pressure change of cold gases	kPa
DT	: change in inst. time increment	s
DV	: change in inst. volume increment	m^3
DW	: inst. wotk increment	kJ
DWI	: ideal inst. work increment	kJ
DZ	: inst. crank angle increment	rad
DTMAX	: max. time allowed for integration step	s
DTMIN	: min. time allowed for integration step	s
DH(5)	: gas equation constants	
DL(5)	: gas equation constants	
E	: constant (k-1/k)	
EFF	: brake thermal efficiency	%
EK	: constant (1/3)	
EO	: energy available for work	kJ
EVLD	: exhaust valve large diameter	cm
EVSD	: exhaust valve small diameter	cm
F	: fraction of do loop duration	
FF	: fuel flow	kg/hr
FMEP	: frictional mean effective pressure	kPa

LIST OF VARIABLES

		Units
HC	: inst. heat transfer coefficient	kW/m^2-K
HC1	: inst. heat transfer coefficient for hot gases	kW/m^2-K
HC2	: inst. heat transfer coefficient for cold gases	kW/m^2-K
HRP	: constant pressure heat of reaction for fuel	kJ/kmol
I	: do loop variable	cm
IVLD	: intake valve large diameter	cm
IVSD	: intake valve small diameter	
K	: specific heats ratio	
KB	: inst. specific heat ratio used in progressive combustion	
KP	: specific heat ratio for product gases	
KK	: do loop variable	
KR	: specific heat ratio for reactant gases	
L	: connecting length	cm
LFT	: inst. valve lift	m
LIFT	: max. valve lift	cm
M	: inst. mass in cylinder	kmol
MDOT	: change of mass flow w.r.t. time	kmol/s
MEP	: mean effective pressure	kPa
MF	: mass of fuel	kg
MC	: number of carbon atoms in fuel	
MH	: number of hydrogen atoms in fuel	
MO	: number of oxygen atoms in fuel	
MPS	: mean piston speed	m/s
NC	: do loop variable	
NCYL	: number of cylinders in engine	
ND	: do loop variable	
NM	: number of kmol of fuel vapor	kmol
NMO	: number of kmol of fuel at stoich. conditions	kmol
NP	: number of kmol of product gases formed by comb.	kmol
NPO	: number of kmol of product gases at stoich. cond.	kmol
NW	: counter constant	
NX	: number of kmol of residual gases	kmol
NXO	: old number of kmol of residual gases	kmol
N(1)	: kmol of carbon monoxide	kmol
N(2)	: kmol of carbon dioxide	kmol
N(3)	: kmol of water vapor	kmol
N(4)	: kmol of oxygen	kmol
N(5)	: kmol of nitrogen	kmol
P	: inst. cylinder pressure	kPa
PC	: crank case pressure	kPa
PE	: exhaust manifold pressure	kPa
PH	: peak cylinder pressure	mPa
PI	: constant (3.14159)	
PM	: intake manifold pressure	kPa
PNEW	: new pressure calculated for a process	kPa
PO	: atmospheric pressure	kPa
POLD	: old pressure following a process	kPa
POWER	: engine power produced from cycle	kW
PR	: pressure ratio	
PX	: pressure of residual gases	kPa
P1NEW	: new pressure before compression stroke	kPa
P1	: pressure before compression stroke	kPa
P2	: pressure after compression stroke	kPa
P3	: pressure following combustion	kPa
P4	: pressure following expansion stroke	kPa

LIST OF VARIABLES

		Units
Q	: mass flow rate through valve	kPa/s
QV	: total amount of energy available for work	KJ
R	: universal gas constant (8.314)	kJ/kmol-K
RPM	: engine speed	rev-min
S	: stroke length	cm
SA	: spark advance	deg
SETANG	: seat angle of valve	deg
SFC	: brake specific fuel consumption	g/kW-Hr
SN	: total fraction of gases burned during combustion	
T	: inst. cylinder temperature	K
TCOLD	: inst. temperature of cold gases	K
THNESS	: valve thickness	cm
THOT	: inst. temperature of hot gases	K
TM	: intake manifold temperature	K
TNEW	: new temperature calculated from a process	K
TOLD	: old temperature following a process	K
TPOWER	: total power output for all cylinders	kW
TW	: temperature of cylinder walls	K
TX	: temperature of residual gases	K
T1NEW	: new temperature before compression stroke	K
T1	: temperature before compression stroke	K
T2	: temperature after compression stroke	K
T3	: temperature after combustion	K
T4	: temperature after expansion stroke	K
URP	: constant volume heat of reaction for fuel	kJ/kmol
V	: inst. cylinder volume	m^3
VBDC	: cylinder volume at bottom dead center	m^3
VDISP	: cylinder displacement volume	m^3
VDOT	: change in volume w.r.t. time	m^3/s
VE	: volumetric efficiency	%
VEO	: volume of cylinder when exhaust valve opens	m^3
VIC	: volume of cylinder when intake valve closes	m^3
VOL	: old volume following a process	m^3
WC	: compression work	kJ
WCOMP	: compression work done during combustion	kJ
WCOMP1	: compression work done during compression	kJ
WEXP	: expansion work	kJ
WL	: work done during valving	kJ
WLOOP	: work done during valving	kJ
WNET	: net work	kJ
Y	: air/fuel ratio	
YCC	: stoich. air/fuel ratio	
YMIN	: min. air/fuel ratio for combustion	
Z	: inst. crank angle	rad
ZC	: combustion duration	deg
ZCR	: combustion duration in radians	rad
ZDOT	: change in crank angle w.r.t. time	rad/s
ZE	: exhaust valve opening angle	deg
ZI	: intake valve closing angle	deg
ZIG	: crank angle at spark ignition	deg
ZIO	: intake valve opening angle	deg

```
FILE: CYCLE     WATFIV   A   KANSAS STATE UNIVERSITY VM/SP CMS


$JOB            ,P=20,T=(2,),NOLIST
C
C      OTTO CYCLE (METHANE FUEL) WITH PROGRESSIVE COMBUSTION
C      AND REALISTIC VALVING FOR THE AN IRRIGATION ENGINE
C
C
      REAL IVSO,IVLO,K,KB,KP,KR,L,LIFT,MC,MEP,MF,MH,MO,MPS
      REAL N,NM,NP,NX,NMO,NPO,NXO
      COMMON B,K,L,F,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLO,IVLO,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
C               W(Z)= VOLUME AT CRANK ANGLE Z
C               G(Z)= ADDITIONAL TERM IN F(Z)
C               F(Z)= DERIVITIVE OF VOLUME WRT. CRANK ANGLE
C               A(Z)= HEAT TRANSFER AREA AT CRANK ANGLE Z
      W(Z)=VBDC-VDISP*(1.-COS(Z)+SQRT(C1**2-(SIN(Z))**2)-C1)/2.
      G(Z)=2.*SQRT(C1**2-(SIN(Z))**2)
      F(Z)=VDISP*(SIN(2.*Z)/G(Z)-SIN(Z))/2.
      A(Z)=PI*B**2/4.+4.*W(Z)/B
C
C
C           READ IN COEFFICIENTS FOR GAS PROPERTIES
C
      DO 900 I=1,5
  900 READ(5,901) GAS,AL(I),BL(I),CL(I),DL(I),AH(I),BH(I),CH(I),DH(I)
  901 FORMAT(A4,F6.0,F6.2,F9.1,F8.2,F8.0,F6.2,F9.1,F8.2)
C
C           INPUT OF COEFFICIENTS COMPLETE
C
C           READING OF FUEL AND HEATING VALUE USED IN THE CYCLE
C
      READ(5,175)GAS,AA,BB,MC,MH,MO,HRP
  175 FORMAT(A6,F5.1,F7.4,3F4.0,F9.0)
C
C           INPUT OF FUEL DATA COMPLETE
C
C           FORMAT OF HEADING FOR OUTPUT DATA
C
      WRITE(6,202)
  202 FORMAT(1H1,2X,1HY,4X,2HCR,4X,2HSA,4X,3HRPM,3X,4HMEAN,5X,
     C   12HVALVE TIMING,4X,3HMEP,5X,2HKW,7X,4HBSFC,3X,7HVOL EFF,
     C   3X,2HPE,5X,2HPM,6X,2HP1,5X,2HPH,4X,2HWL,4X,3HEFF/
     C   26X,6HPISTON,3X,3HEXH,5X,3HINT,5X,3HKPA,11X,9H(G/KW-HR),
     C   11X,3HKPA,5X,3HKPA,5X,3HKPA,4X,3HMPA,3X,2HKW,5X,1H%/
     C   26X,5HSPEED,4X,4HOPEN,4X,5HCLOSE/
     C   12X,7HDEGREES,7X,5H(M/S),7X,7HDEGREES/
     C   13X,4HBTDC,18X,4HBBDC,4X,4HABDC)
C
C           HEADING COMPLETE
C
      PI=3.14159
```

```
      R=8.314
C
C             INITIALIZATION OF ENGINE SPECIFICATIONS
C
      B=10.16/100.
      S=8.84/100.
      L=14.5/100.
      CR=8.5
      NCYL=8.
      RPM=1500.
      LIFT=1.14/100.
      THNESS=.64/100.
      SETANG=45.*PI/180.
      IVSD=4.45/100.
      EVSD=3.81/100.
C
C             ENGINE SPECS. COMPLETE
C
      C1=2.*L/S
      VDISP=PI*B**2*S/4.
      VBDC=VDISP/(1.-1./CR)
      VTDC=VBDC-VDISP
      YCC=MC+MH/4.-MO/2.
      YMIN=YCC-MC/2.
      URP=-HRP-2480.*(MC+MH/2.-YCC-1.)
      IVLD=IVSD+2.*THNESS/TAN(SETANG)
      EVLD=EVSD+2.*THNESS/TAN(SETANG)
      AIO=PI/2.*(IVLD+IVSD)*SQRT((IVLD-IVSD)**2/4.+(LIFT-THNESS)**2)
      AEO=PI/2.*(EVLD+EVSD)*SQRT((EVLD-EVSD)**2/4.+(LIFT-THNESS)**2)
C
C             BEGINNING OF CYCLE CALCULATIONS
C
      READ(5,400) ZED,ZIO,ZID
  400 FORMAT(3F10.0)
      DO 700 KK=1,10
      PM=101.4
      PC=250.
      PE=101.4
      PO=101.4
      TM=300.
      TW=423.
      RPM=RPM+100.
      ZDOT=2.*PI*RPM/60.
      Y=YCC+.2
      ZE=ZED*PI/180.
      ZI=ZID*PI/180.+2.*PI
      VEO=W(-ZE)
      VIC=W(ZI)
C
      P1=PM
      T1=TM
      NMO=1.+4.76*YCC
      NPO=MC+MH/2.+3.76*YCC
      CPR=(4.76*YCC*29.3+AA+BB*T1)/NMO
      CVR=CPR-R
```

```
      KR=CPR/CVR
   14 NM=P1*VIC/R/T1
      VE=1.
      NX=0.
      NX0=0.
      NW=0.
C
C
C
   20 NW=NW+1
      RN=NM/(NM+NX)
      CM=NM/NMO+NX/NPO
      IF(Y-YCC) 3,2,2
    2 N(1)=0.
      N(2)=MC*CM
      N(3)=MH/2.*CM
      N(4)=(Y-YCC)*CM
      N(5)=3.76*Y*CM
      GO TO 4
    3 N(1)=(2.*(YCC-Y))*CM
      N(2)=2.*(YCC-YMIN)*CM
      N(3)=MH/2.*CM
      N(4)=0.
      N(5)=3.76*CM*Y
    4 QV=-(URP+(N(1)/CM)*281400.)*NM/NMO
      NP=N(1)+N(2)+N(3)+N(4)+N(5)
      ZC=40.+156.*(YCC/Y-1.1)**2+5.*(RPM/600.-1.)
C
C             COMPRESSION STROKE BEGINS
C
      Z=ZIO*PI/180.
      WCOMP1=0.
      CALL COMP
C
C             COMPRESSION COMPLETE
C
      T=2000.
    6 TNEW=T-(U(T)-U(T2*(V/VTDC)**(KR-1.))-QV)/CV(T)
      IF(ABS(T-TNEW)-5.) 9,9,8
    8 T=TNEW
      GO TO 6
    9 T3=TNEW
      P3=P2*(V/VTDC)**KR*NP*T3/(T2*(V/VTDC)**(KR-1.))/(NM+NX)
      DELP=P3-P2*(V/VTDC)**KR
      KP=(CV(T3)+R*NP)/CV(T3)
C
C             PROGRESSIVE COMBUSTION BEGINS HERE
C
      ZIG=Z
      ZCR=ZC*PI/180.
      V=W(Z)
      P=P2
      TCOLD=T2
      POLD=P
      TOLD=TCOLD
```

```
      WCOMP=0.
      WEXP=0.
      DPH1=0.
      DPH2=0.
      DWI=0.
      NC=40
      PH=P2
      DZ=ZC*PI/NC/180.
      DT=DZ/ZDOT
      SN=0.
      DO 300 I=1,NC
      Z=Z+DZ
      DV=W(Z)-V
      DN=PI*SIN((Z-ZIG)*PI/ZCR)/NC/2.
      SN=SN+DN
      KB=KR-(KR-KP)*SN
      DP=(-KB*P*DV+DELP*VTDC*DN)/V
      DWI=DWI+((P-PC)+DP/2.)*DV
C
      CALL TEMP(QV,TCOLD,THOT,I,NC,DWI)
C
      HC1=.0006*S*RPM*(CV(THOT)/NP+R)*(P+DP)/THOT/R*
     C  (W(ZI)/V)**(1./3.)
      DPH1=(P+DP)*HC1*(1.*I/NC)*A(Z)*(TW-THOT)*DT/(CV(THOT)*THOT)
      CP=(4.76*YCC*29.3+AA+BB*TCOLD)/NMO
      HC2=.0006*S*RPM*CP*(P+DP)/TCOLD/R*(W(ZI)/V)**(1./3.)
      DPH2=(P+DP)*HC2*(1-I*1./NC)*A(Z)*(TW-TCOLD)*DT/(NM*(CP-R)*TCOLD)
      DP=DP+DPH1+DPH2
      DW=((P-PC)+DP/2.)*DV
      IF(DW.LT.0) WCOMP=WCOMP-DW
      IF(DW.GT.0) WEXP=WEXP+DW
  303 P=P+DP
      IF(P.GT.PH) PH=P
      V=V+DV
      TCOLD=TOLD*(P/POLD)**((KR-1.)/KR)
      POLD=P
      TOLD=TCOLD
  300 CONTINUE
C
C               PROGRESSIVE COMBUSTION COMPLETE
C
C
C               EXPANSION STROKE BEGINS
C
      CALL EXP
C
C               EXPANSION COMPLETE
C
      K=(CV(T4)+R*NP)/CV(T4)
C
C               EXHAUST AND INTAKE BEGINS HERE
C
      CALL EXHINT
C
C               EXHAUST AND INTAKE COMPLETE
```

```
C
      IF(ABS((NXO-NX)/NX).LT..01) GO TO 39
      NXO=NX
      P1=P1NEW
      T1=T1NEW
      CPM=(4.76*YCC*29.3+AA+BB*T1)*NM/NMO
      CPX=N(1)*29.3+N(2)*37.2+N(3)*33.5+N(4)*29.3+N(5)*29.7
      CPX=CPX*NX/NP
      CPR=(CPM+CPX)/(NM+NX)
      CVR=CPR-R
      KR=CPR/CVR
      GO TO 20
C
C
   39 MF=(NM/NMO*(MC*12.+MH*1.+MO*16.))*1000.
      VE=NM*R*TM/PM/VDISP
      SA=SA*180./PI
      FF=60./2.*MF*RPM
      MPS=2.*S*RPM/60.
      FMEP=((.076*MPS**2+.92*MPS+7.23)/14.7+.5)*101.4
      WNET=WEXP-WCOMP+WLOOP+WCOMP1-(FMEP*VDISP)
      MEP=WNET/VDISP
      EFF=-100.*WNET*NMO/NM/URP
      POWER=WNET*RPM/120.
      WL=WLOOP*NCYL*RPM/120.
      WC=WCOMP1*NCYL*RPM/120.
      SFC=FF/POWER
      TPOWER=NCYL*POWER
C
C             CYCLE CALCULATION COMPLETE
C
      PH=PH/1000.
C
C             PRINT OUT CYCLE CALCULATIONS
C
      WRITE(6,201) Y,CR,SA,RPM,MPS,ZED,ZID,MEP,TPOWER,SFC,VE,
     C  PE,PM,P1,PH,WL,EFF
  201 FORMAT(F6.2,F5.1,F5.1,F7.0,F7.2,2F8.1,F5.0,F8.2,F10.2,F8.2,
     C  3F8.1,F7.3,F5.1,F7.2)
C
C             PRINTOUT COMPLETE
C
  700 CONTINUE
  800 STOP
      END
C
C
C
C     ************************************************************************
C                 FUNCTION U(X) USED TO DETERMINE THE
C                 INTERNAL ENERGY OF THE PRODUCT GASES
C     ************************************************************************
C
      FUNCTION U(X)
      REAL N
```

```
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      U=0.
      IF(X.GT.1600.) GO TO 3
      DO 2 I=1,5
    2 U=U+N(I)*(AL(I)+(BL(I)-R)*X+CL(I)*ALOG(X))
      RETURN
    3 DO 4 I=1,5
    4 U=U+N(I)*(AH(I)+(BH(I)-R)*X+CH(I)*ALOG(X))
      RETURN
      END
C
C     ******************************************************************
C                    FUNCTION UR(X) USED TO DETERMINE THE
C                    INTERNAL ENERGY OF THE REACTANT GASES
C     ******************************************************************
C
      FUNCTION UR(X)
      REAL NM,NMO
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      UR=0.
      CP=(1.*(38.4+.429*X)+YCC*(27+.0079*X)+3.76*YCC*(27.6+.0051*X))
      CP=CP/NMO
      UR=(CP-R)*NM
      RETURN
      END
C
C     ******************************************************************
C                    FUNCTION CV(X) USED TO FIND THE CONSTANT
C                    VOLUME SPECIFIC HEAT OF THE PRODUCT GASES
C     ******************************************************************
C
      FUNCTION CV(X)
      REAL N
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      CV=0.
      IF(X.GT.1600.) GO TO 3
      DO 2 I=1,5
```

```
    2 CV=CV+N(I)*(BL(I)-R+CL(I)/X)
      RETURN
    3 DO 4 I=1,5
    4 CV=CV+N(I)*(BH(I)-R+CH(I)/X)
      RETURN
      END
C
C     ******************************************************************
C                     SUBROUTINE TEMP USED TO FIND THE TEMPERATURE
C                     OF THE HOT AND COLD GASES DURING THE
C                     PROGRESSIVE COMBUSTION PROCESS
C     ******************************************************************
C
      SUBROUTINE TEMP(QV,TCOLD,THOT,I,NC,DWI)
      REAL K,KP,KR,M,MDOT,NMO,NM,NP,NX
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLO,IVLO,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      F=1.*I/NC
      TCOLD=TCOLD*((P+DP)/P)**((KR-1.)/KR)
   10 EO1=U(TCOLD)+QV
      EO=(EO1+F*UR(T2)*T2+F*QV-F*U(TCOLD)-F*QV-DWI)
      TT=2000.
    6 TNEW=TT-(U(TT)-EO)/CV(TT)
      IF(ABS(TT-TNEW)-5.) 9,9,8
    8 TT=TNEW
      GO TO 6
    9 THOT=TNEW
      RETURN
      END
C
C     ******************************************************************
C                     SUBROUTINE COMP USED TO SIMULATE THE
C                     COMPRESSION PROCESS WITH HEAT TRANSFER
C     ******************************************************************
C
      SUBROUTINE COMP
      REAL K,KP,KR,M,MDOT,NM,NMO,NX
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLO,IVLO,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      W(Z)=VBDC-VDISP*(1.-COS(Z)+SQRT(C1**2-(SIN(Z))**2)-C1)/2.
      A(Z)=PI*B**2/4.+4.*W(Z)/B
      NC=50
      SA=(ZC/2.)*PI/180.
      DZ=(PI-Z-SA)/NC
      DT=DZ/ZDOT
```

```
      PNEW=P1
      TNEW=T1
      VOL=W(Z)
      DPH=0.
      DO 1 I=1,NC
      Z=Z+DZ
      V=W(Z)
      P=PNEW*(VOL/V)**KR
      T=P*V/NM/R
      CP=(4.76*Y*29.3+AA+BB*T)/NMO
      HC=.0006*S*RPM*CP*P/T/R*(W(ZI)/V)**(1./3.)
      DPH=P*HC*A(Z)*(TW-T)*DT/(NM*(CP-R)*T)
      DV=V-VOL
      DP=(P-PNEW)+DPH
      DW=((PNEW-PC)+DP/2.)*DV
      WCOMP1=WCOMP1+DW
      PNEW=PNEW+DP
      VOL=V
      TNEW=PNEW*VOL/R/NM
    1 CONTINUE
      P2=PNEW
      T2=TNEW
      RETURN
      END
C
C     ************************************************************************
C                       SUBROUTINE EXP USED TO SIMULATE THE
C                       EXPANSION PROCESS WITH HEAT TRANSFER
C     ************************************************************************
C
      SUBROUTINE EXP
      REAL K,KP,KR,M,MDOT,NM,NP,NX
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIC
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      W(Z)=VBDC-VDISP*(1.-COS(Z)+SQRT(C1**2-(SIN(Z))**2)-C1)/2.
      A(Z)=PI*B**2/4.+4.*W(Z)/B
      NO=80
      DZ=(2.*PI-Z-ZE)/NO
      DT=DZ/ZDOT
      VOL=V
      DPH=0.
    1 Z=Z+DZ
      V=W(Z)
      PNEW=P*(VOL/V)**(KP)
      TNEW=PNEW*V/NP/R
      HC=.0006*S*RPM*(CV(TNEW)/NP+R)*PNEW/TNEW/R*(W(ZI)/V)**(1./3.)
      DPH=PNEW*HC*A(Z)*(TW-TNEW)*DT/(CV(TNEW)*TNEW)
      DV=V-VOL
      DP=(PNEW-P)+DPH
      DW=((P-PC)+DP/2.)*DV
```

```
      WEXP=WEXP+DW
      P=P+DP
      VOL=V
      T=P*VOL/R/NP
      IF((Z+DZ).LT.(2*PI-ZE)) GO TO 1
      T4=T
      P4=P
      RETURN
      END
C
C     ************************************************************************
C                    SUBROUTINE EXHINT USED AS THE CONTROL
C                    STRUCTURE FOR THE VALVING ROUTINES
C     ************************************************************************
C
      SUBROUTINE EXHINT
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      CUT=10.
      DTMAX=3.*PI/180./ZDOT
      DTMIN=.25*PI/180./ZDOT
      WLOOP=0.
      EK=.333
      Z=-ZE
      CALL EXH
      CALL TDC
      CALL SUC
      RETURN
      END
C
C     ************************************************************************
C                    SUBROUTINE EXH USED TO SIMULATE THE
C                    EXHAUST PROCESS DURING THE CYCLE
C     ************************************************************************
C
      SUBROUTINE EXH
      REAL K,LFT,LIFT,M,MDOT,NM,NX
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PF,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      G(Z)=2.*SQRT(C1**2-(SIN(Z))**2)
      F(Z)=VDISP*(SIN(2.*Z)/G(Z)-SIN(Z))/2.
      W(Z)=VBDC-VDISP*(1.-COS(Z)+SQRT(C1**2-(SIN(Z))**2)-C1)/2.
      E=(K-1.)/K
      CRIT=((K+1.)/2.)**(K/(K-1.))
      CSUB=-SQRT(2000./E/R/28.97)
```

```
      CSUP=(2./(K+1.))**((K+1.)/(K-1.))
      CSUP=-SQRT(CSUP*K*1000./28.97/R)
      P=P4
      T=T4
      V=W(Z)
      M=P*V/T/R
    1 AREA=AEO*(ABS(SIN(PI*(Z+ZE)/(PI+ZE))))**EK
      LFT=LIFT*(ABS(SIN(PI*(Z+ZE)/(PI+ZE))))**EK
      CD=-20./3.*(LFT/EVLD-.3)**2+1./3.*(LFT/EVLD-.3)+.7
      VDOT=F(Z)*ZDOT
      IF(P.LT.PE) GO TO 4
      DP=(P-PE)/CUT
      IF(P/PE.GT.CRIT) GO TO 2
      PR=(P/PE)**E
      MDOT=CD*AREA*CSUB*SQRT(PR*(PR-1.)/T)*PE
      GO TO 3
    4 IF(PE/P.GT.CRIT) GO TO 5
      PR=(PE/P)**E
      MDOT=-CD*AREA*CSUB*SQRT(PR*(PR-1.)/T)*PE
      GO TO 6
    5 MDOT=-CD*AREA*CSUP*PE/SQRT(T)
    6 DP=(PE-P)/CUT
      Q=K*(T*MDOT*R-P*VDOT)/V
      GO TO 7
    2 MDOT=CD*AREA*CSUP*P/SQRT(T)
    3 Q=K*P*(MDOT/M-VDOT/V)
    7 CALL STEP
      P=P+DP
      Z=Z+ZDOT*DT
      V=W(Z)
      T=T4*(P/P4)**E
      M=P*V/T/R
      IF(Z.LT.PI) GO TO 1
      PX=P
      TX=T
      NX=M
      RETURN
      END
C
C     ****************************************************************
C                     SUBROUTINE TDC USED TO DECREASE THE
C                     CYLINDER PRESSURE BELOW THE MANIFOLD
C                     PRESSURE SO THAT THE INTAKE STROKE
C                     CAN BEGIN
C     ****************************************************************
C
      SUBROUTINE TDC
      REAL IVLD,K,LFT,LIFT,M,MDOT,NX
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BS,C1,CP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIC,CUT,CVR,NMO,RPM,YCO,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
```

```
      G(Z)=2.*SQRT(C1**2-(SIN(Z))**2)
      F(Z)=VDISP*(SIN(2.*Z)/G(Z)-SIN(Z))/2.
      W(Z)=VBDC-VDISP*(1.-COS(Z)+SQRT(C1**2-(SIN(Z))**2)-C1)/2.
      IF(ZIO.NE.0.) GO TO 4
      IF(PM.LT.P) GO TO 1
      RETURN
    4 ZO=PI+ZIO*PI/180.
      NC=30
      VOL=W(Z)
      DZ=(ZO-PI)/NC
      DO 5 I=1,NC
      Z=Z+DZ
      V=W(Z)
      PNEW=P*(VOL/V)**K
      DP=PNEW-P
      DV=V-VOL
      DW=((P-PC)+DP/2)*DV
      WLOOP=WLOOP+DW
      VOL=V
      P=P+DP
    5 CONTINUE
      RETURN
    1 E=(K-1.)/K
      CSUB=-SQRT(2000./E/28.97/R)
      T=TX
      P=PX
      M=NX
      V=W(Z)
    2 DP=(P-PM)/CUT
      PR=(P/PM)**E
      AREA=AIO*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      LFT=LIFT*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      CD=-20./3.*(LFT/IVLD-.3)**2+1./3.*(LFT/IVLD-.3)+.7
      MDOT=CD*AREA*CSUB*SQRT(PR*(PR-1.)/T)*P
      VDOT=F(Z)*ZDOT
      Q=K*P*(MDOT/M-VDOT/V)
      CALL STEP
      P=P+DP
      Z=Z+ZDOT*DT
      V=W(Z)
      T=TX*(P/PX)**E
      M=P*V/T/R
      IF(P-PM.GT.0.68) GO TO 2
      P=PM-.68
      CSUB=-CSUB
    3 DP=(PM-P)/CUT
      PR=(PM/P)**E
      AREA=AIO*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      LFT=LIFT*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      CD=-20./3.*(LFT/IVLD-.3)**2+1./3.*(LFT/IVLD-.3)+.7
      MDOT=CD*AREA*CSUB*SQRT(PR*(PR-1.)/TM)*PM
      VDOT=F(Z)*ZDOT
      Q=K*(TM*MDOT*R-P*VDOT)/V
      CALL STEP
      P=P+DP
```

```
      Z=Z+ZDOT*DT
      V=W(Z)
      M=M+MDOT*DT
    6 IF(P.LT.PM.AND.M.LT.NX) GO TO 3
      IF(P.GT.PM.AND.M.LT.NX) GO TO 7
      RETURN
    7 DP=(P-PM)/CUT
      PR=(P/PM)**E
      AREA=AIO*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      LFT=LIFT*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      CD=-20./3.*(LFT/IVLD-.3)**2+1./3.*(LFT/IVLD-.3)+.7
      T=TX*(P/PX)**E
      MDOT=-CD*AREA*CSUB*SQRT(PR*(PR-1.)/T)*P
      VDOT=F(Z)*ZDOT
      Q=K*P*(MDOT/M-VDOT/V)
      CALL STEP
      P=P+DP
      Z=Z+ZDOT*DT
      V=W(Z)
      M=M+MDOT*DT
      IF(M.LT.NX) GO TO 6
      RETURN
      END
C
C     ************************************************************************
C                   SUBROUTINE SUC USED TO SIMULATE THE
C                   INTAKE STROKE DURING THE CYCLE
C     ************************************************************************
C
      SUBROUTINE SUC
      REAL IVLD,K,KR,LFT,LIFT,M,MDOT,NM,NX
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AO,BB,C1,DP,DT,EK,KP,KR,NM,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      G(Z)=2.*SQRT(C1**2-(SIN(Z))**2)
      F(Z)=VDISP*(SIN(2.*Z)/G(Z)-SIN(Z))/2.
      W(Z)=VBDC-VDISP*(1.-COS(Z)+SQRT(C1**2-(SIN(Z))**2)-C1)/2.
      M=NX
      K=KR
      E=(K-1.)/K
      CRIT=((K+1.)/2.)**(K/(K-1.))
      CSUP=(2./(K+1.))**((K+1.)/(K-1.))
      CSUP=SQRT(CSUP*K*1000./28.97/R)
      CSUB=SQRT(2000./E/28.97/R)
      V=W(Z)
      L=1
    1 AREA=AIO*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      LFT=LIFT*(ABS(SIN(PI*(Z-ZI)/(PI-ZI))))**EK
      CD=-20./3.*(LFT/IVLD-.3)**2+1./3.*(LFT/IVLD-.3)+.7
      VDOT=F(Z)*ZDOT
      GO TO (2,3),L
```

```
    2 IF(Z.LE.2.*PI) GO TO 3
      IF(P.LT.PM-.68) GO TO 4
      L=2
      P=PM+.68
    3 IF(P.LT.PM) GO TO 4
      DP=(P-PM)/CUT
      T=P*V/M/R
      IF(P/PM.GT.CRIT) GO TO 8
      PR=(P/PM)**E
      MDOT=-CD*AREA*CSUB*SQRT(PR*(PR-1.)/T)*P
      GO TO 9
    8 MDOT=-CD*AREA*CSUP*P/SQRT(T)
    9 Q=K*P*(MDOT/M-VDOT/V)
      GO TO 7
    4 DP=(PM-P)/CUT
      IF(PM/P.GT.CRIT) GO TO 5
      PR=(PM/P)**E
      MDOT=CD*AREA*CSUB*SQRT(PR*(PR-1.)/TM)*PM
      GO TO 6
    5 MDOT=CD*AREA*CSUP*PM/SQRT(TM)
    6 Q=K*(TM*MDOT*R-P*VDOT)/V
    7 CALL STEP
      P=P+DP
      Z=Z+ZDOT*DT
      V=W(Z)
      M=M+MDOT*DT
      IF(Z.LT.ZI) GO TO 1
      T1NEW=P*V/R/M
      P1NEW=P
      NM=M-NX
      RETURN
      END
C
C     *******************************************************************
C                     SUBROUTINE STEP USED TO DETERMINE THE
C                     APPROPRIATE STEP SIZE FOR THE INTEGRATION
C                     TECHNIQUE IN THE VALVING SUBROUTINES
C     *******************************************************************
C
      SUBROUTINE STEP
      COMMON B,K,L,P,Q,R,S,T,V,Y,Z
      COMMON AA,AD,EB,C1,DP,DT,EK,KP,KR,N4,NP,NW,NX,PC,PE,PI,PM,PO,PX
      COMMON P1,P2,P4,SA,TM,TW,TX,T1,T2,T4,ZC,ZE,ZI
      COMMON AEO,AIO,CUT,CVR,NMO,RPM,YCC,ZIO
      COMMON EVLD,IVLD,LIFT,VBDC,VDOT,ZDOT,WEXP
      COMMON DTMAX,DTMIN,P1NEW,T1NEW,VDISP,WLOOP,WCOMP1
      COMMON N(5),AH(5),AL(5),BH(5),BL(5),CH(5),CL(5),DH(5),DL(5)
      IF(Q) 2,5,6
    2 DT=-DP/Q
      IF(DT.LT.DTMAX) GO TO 3
      DT=DTMAX
      DP=Q*DT
      GO TO 9
    3 IF(DT.LT.DTMIN) GO TO 4
      DP=-DP
```

```
      GO TO 9
    4 DT=DTMIN
      DP=Q*DT
      GO TO 9
    5 DP=0.
      DT=DTMAX
      GO TO 9
    6 DT=DP/Q
      IF(DT.LT.DTMAX) GO TO 8
      DT=DTMAX
      DP=Q*DT
      GO TO 9
    7 DT=DTMIN
      DP=Q*DT
      GO TO 9
    8 IF(DT.LT.DTMIN) GO TO 7
    9 WLOOP=WLOOP+((P-PC)+DP/2.)*VDOT*DT
      RETURN
      END
$ENTRY
 CO 299180. 37.85  -4571.9  -31.10 309070. 39.29  -6201.9  -42.77
 CO2 56835. 66.27 -11634.0 -200.00  93048. 68.53 -16979.0 -220.40
 H2O 88923. 49.36  -7940.3 -117.00 154670. 50.45 -19212.0 -204.60
 O2  43386. 42.27  -6635.4  -55.15 127010. 46.25 -15798.0  -92.15
 N2  31317. 37.46  -4659.3  -34.82  44639. 39.32  -6753.4  -50.24
 CH4   20.1 0.0052  1.   4.   0.   797570
 77.       0.        76.
```

THE COMPUTER MODELING OF A

MORE-COMPLETE-EXPANSION CYCLE WITH

ITS APPLICATION TO AN IRRIGATION ENGINE

by

GEORGE ROSS EAKIN

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AN ABSTRACT OF A MASTER'S THESIS

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ABSTRACT

The more-complete-expansion cycle can be applied to a stationary irrigation engine commonly operated at constant partial-load burning liquid fuels. The analysis involved the development of a digital computer model to simulate a real engine.

The Chevrolet 5.74 liter (350 cu. in.) engine was simulated to determine the performance changes and possible advantages of applying the more-complete-expansion cycle features. The fuel used in this model was methane gas. The generated performance data is presented in figures. The modified engine specifications are results of these figures and are used to determine the cost effectiveness of the concept.

The performance results in conjunction with a cost analysis suggest that a fuel saving greater than 11% can be achieved by utilizing the more-complete-expansion cycle. If the engine is operated frequently, the amount of money saved in fuel is significant and could pay off the cost of the engine modification in less than a year. Therefore, this idea is recommended for further development and testing.