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FREE PISTON GASIFIER STUDIES:

FURTHER DEVELOPMENT AND USE OF THE HYBRID SIMULATION

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SUMMARY

Previous work on the modelling of the free piston gasifier has been extended to include refined models of the combustion process and of engine friction. The model was used to study the sensitivity of the engine performance to environmental and design parameters.

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SYMBOLS

Symbol	Definition
A_d	double area of diesel cylinder
C	viscous friction coefficient
c_p	specific heat at constant pressure

Symbol	Definition
c_v	specific heat at constant volume
HV	lower heating value of the fuel
F	total friction force
F_c	Coulomb friction force
FLEAK	gasifier leakage factor
ghp	gas horsepower
IML	inner mechanical limit
J	mechanical equivalent of heat
m_a	mass of air trapped in the diesel cylinder
m_f	mass of fuel injected per stroke
m_{f56}	mass of fuel burned between beginning of combustion and IDP
n_c	combustion exponent
ODP	outer dead point
P	pressure
P_{amb}	atmospheric pressure
P_{bref}	bounce reference pressure
P_4	pressure in diesel cylinder at closing of intake ports
P_5	compression pressure in diesel cylinder
P_6	maximum pressure in diesel cylinder
Q	amount of heat
R	gas constant
SFC	specific fuel consumption

Symbol	Definition
T	temperature
T_{amb}	atmospheric temperature
T_4	temperature in diesel cylinder at closing of intake ports
T_5	compression temperature in diesel cylinder
T_6	temperature in diesel cylinder at the start of constant volume combustion
T_7	temperature in diesel cylinder at the end of combustion
U	internal energy
V	volume
V_r	gas receiver volume
V_5	volume of diesel cylinder at end of compression
V_6	diesel cylinder volume at IDP
V_7	diesel cylinder volume at end of combustion
$\dot{x}$	piston speed
x_{d_4}	position of diesel exhaust ports
x_{d_6}	diesel piston position at IDP
x_{d_7}	diesel piston position at end of combustion
γ	isentropic exponent (ratio of specific heat of constant pressure to that at constant volume)
γ_c	compressor polytropic exponent
γ_{45}	isentropic exponent during diesel compression process
ΔT	temperature difference
Δx_5	start of combustion measured from IDP
η	assumed efficiency factor assigned to the combustion process, but including heat losses and gas leakage for the entire cycle and excluding friction

FREE PISTON GASIFIER STUDIES:
FURTHER DEVELOPMENT AND USE OF THE HYBRID SIMULATION

1.0 INTRODUCTION

Reference 1 describes the development of a hybrid computer simulation of a free piston gasifier, and compares the gasifier performance with the output of the simulation under similar running conditions. One of the observations made in this reference was that, although generally good agreement was obtained, there were discrepancies in the shapes of some of the curves from the engine tests and the simulation. The most obvious of these was in the plot of inner dead point versus load, which was almost flat for the simulation, but increased sharply with load in the engine. The explanation offered for this was that in the engine, injection begins earlier with increasing fuel, thus causing a greater piston deceleration at higher load and moving the IDP outward. In the model this effect was absent because with the simple heat release model chosen (constant volume followed by constant pressure) no heat release could take place until IDP was reached.

It was thus argued that a modified heat release model, in which some of the heat was released before IDP, might give a more realistic variation of IDP with load. The present report describes this change in the model and its effect upon predicted performance. Also studied was an experimental change in the modelling of friction, and the effects on performance of small changes in environmental and design parameters.

2.0 THE ENGINE

The engine studied is a small free-piston gasifier of the outward-compressing type, and is described in some detail in Reference 1. Figure 1 shows a longitudinal section of the engine in diagrammatic form, and the principal dimensions and masses are given in Table 1. It will be appreciated that the simulation is by no means restricted to these values.

TABLE 1

ENGINE DATA

Maximum stroke between mechanical limits	5.20 in
Piston mass (2 piston assemblies)	58.00 lb_m
Compressor piston area (2 pistons)	127.60 in^2
Diesel piston area (2 pistons)	19.24 in^2
Negative bounce piston area (2 pistons)	123.31 in^2
Bounce piston area (2 pistons)	15.04 in^2
Compressor cylinder volume at IML (2 cylinders)*	746.90 in^3
Negative bounce volume at IML (2 cylinders)*	181.45 in^3
Bounce cylinder volume at IML (2 cylinders)*	90.86 in^3
Gas receiver volume	4.70 ft^3

*The volume of a cylinder at IML (inner mechanical limit) is its volume when the two diesel piston crowns are in contact.

3.0 THE MODEL AND ITS IMPLEMENTATION

A detailed description of the basic model and of its implementation on the EAI 690 hybrid computer is given in Reference 1 and will not be repeated here. An information flow chart for the simulation is, however, reproduced in Figure 2.

The sections that follow describe the changes introduced into the model.

3.1 Combustion

The combustion process is probably the least understood of all the processes occurring in reciprocating engines. This is supported by Streit and Borman[2] and Borman, Myers and Uyehara[3] in their development of engine models for simulation. The design and development of an actual combustion system is usually done experimentally and on the basis of past experience rather than from theoretical treatment.

The work described in Reference 1 embodies a simple combustion model that assumes constant volume heat release up to a previously selected maximum pressure, followed by constant pressure heat release for any remaining fuel. The oversimplification implied by this model was at first considered acceptable because the resulting pressure-volume diagram bore a close resemblance to the real one, and its area was correct. However, it made no provision for variable injection timing and its effect on piston dynamics, nor for the effect on the inner dead point position of variations in the amount of fuel injected per stroke.

In the real engine, the fuel for combustion is delivered by a jerk-type fuel pump actuated by a cam driven by the pistons themselves. Since the cam comes to rest at IDP, the pump delivery stroke must be completed at or before IDP. Some measure of controlled injection delay is introduced by the compressibility of the fuel trapped in the lines between the pump and the injection nozzles, so that the actual injection will occur later and may, in fact, continue beyond inner dead point. In addition, there is an uncontrolled ignition delay before the actual beginning of heat release.

It was therefore considered not unreasonable to postulate a heat release model in which heat release would begin at a selected point near IDP on the compression stroke and proceed while the piston continued moving toward IDP. Any remaining energy would then be released at constant pressure as before.

The simulation was so arranged that either combustion model could be used. The two schemes are hereafter referred to as Combustion Model I and Combustion Model II respectively, and are defined in some detail below.

3.1.1 Combustion Model I

The constant volume followed by constant pressure heat release model is shown in Figure 3, and is described fully in Reference 1. For the sake of completeness, the derivations of the equations are repeated here, in a simplified form consistent with the second combustion model.

The heat released by the combustion of a mass m_f of fuel,

$$m_f \cdot HV \cdot \eta = m_a \cdot c_v \cdot (T_6 - T_5) + m_a \cdot c_p \cdot (T_7 - T_6) \tag{1}$$

where the numbered subscripts refer to the states indicated in Figure 3, and

HV = lower heating value of the fuel,

η = an assumed efficiency factor assigned to the combustion process, but including heat losses and gas leakage for the entire cycle, and excluding friction,

m_a = mass of gas trapped in the cylinder,

c_v and c_p = specific heats at constant volume and constant pressure, respectively,

and T = temperature.

Equation (1) ignores the heat absorbed by the mass of fuel m_f. Since

$$P \cdot V = m_a \cdot R \cdot T$$

$$T_6 - T_5 = \frac{V_5}{m_a \cdot R} \cdot (P_6 - P_5)$$

and

$$T_7 - T_6 = \frac{P_6}{m_a \cdot R} \cdot (V_7 - V_6)$$

where P and V refer to pressure and volume, respectively, and R is the gas constant. Substituting these values in (1) gives

$$m_f \cdot HV \cdot \eta = \frac{m_a \cdot c_v \cdot V_5 \cdot (P_6 - P_5)}{m_a \cdot R} + \frac{m_a \cdot c_p \cdot P_6 (V_7 - V_6)}{m_a \cdot R}$$

$$= \frac{c_v}{R} \cdot x_{d_6} \cdot A_d \cdot (P_6 - P_5) + \frac{c_p}{R} \cdot A_d \cdot (x_{d_7} - x_{d_6}) \cdot P_6$$

Substituting $c_v = c_p/\gamma$ gives

$$x_{d_7} = x_{d_6} + \frac{1}{P_6}\left[\frac{m_f \cdot HV \cdot \eta \cdot R}{c_p \cdot A_d} - \frac{x_{d_6} \cdot (P_6 - P_5)}{\gamma}\right] \tag{2}$$

If the quantity of fuel injected is insufficient to raise the pressure to the pre-selected maximum value of P_6, all combustion will take place at constant volume, and

$$m_f \cdot HV \cdot \eta = m_a \cdot c_v \cdot (T_6 - T_5)$$

and

$$T_6 - T_5 = \frac{V_5}{m_a \cdot R} \cdot (P_6 - P_5)$$

Combining, and again substituting $c_v = c_p/\gamma$ gives

$$P_6 = P_5 + \frac{m_f \cdot HV \cdot \eta \cdot R \cdot \gamma}{c_p \cdot x_{d_6} \cdot A_d} \tag{3}$$

If, on the other hand, the compression pressure at IDP reaches or exceeds the assumed maximum value, all heat release will take place at the maximum compression pressure, and the constant volume portion of the process disappears. In this case,

$$m_f \cdot HV \cdot \eta = m_a \cdot c_p \cdot (T_7 - T_6)$$

$$= P_6 \cdot \frac{c_p}{R} \cdot A_d \cdot (x_{d_7} - x_{d_6})$$

Hence,

$$x_{d_7} = x_{d_6} + \frac{m_f \cdot HV \cdot \eta \cdot R}{c_p \cdot P_6 \cdot A_d} \tag{4}$$

3.1.2 Combustion Model II

The new combustion model, shown in Figure 4, postulates that heat release begin at a selected point 5 near the end of the compression stroke. This is modelled by switching from the isentropic exponent γ_{45} when the piston reach x_{d_5} to an arbitrarily chosen polytropic exponent n_c having a value much in excess of γ_{45} (typically 2.5). In other respects, the solution of the equation for compression,

$$\frac{dP_d}{dt} = -\gamma_{45} \cdot \left(\frac{P_d \cdot \dot{x}}{x} \right)$$

proceeds normally until the piston reaches IDP at 6.

For polytropic compression with heat addition, the heat added,

$$dQ = dU + P \cdot dV$$

where dU is the change in internal energy.

Hence,

$$\begin{aligned} dQ &= m_a \cdot c_v \cdot dT + m_a \cdot \frac{R}{J} \cdot T \cdot \frac{dV}{V} \\ &= m_a \cdot c_v \cdot dT + m_a \cdot (c_p - c_v) \cdot T \cdot \frac{dV}{V} \end{aligned} \tag{5}$$

For adiabatic compression, $dQ = 0$,

and $$c_v \cdot \frac{dT}{T} = -(c_p - c_v) \cdot \frac{dV}{V}$$

$$\frac{dT}{T} = - \frac{c_p - c_v}{c_v} \cdot \frac{dV}{V}$$

$$\frac{dV}{V} = \frac{dT}{T} \cdot \frac{-1}{\gamma - 1}$$

Similarly, for the polytropic case,

$$\frac{dV}{V} = \frac{dT}{T} \cdot \frac{-1}{n_c - 1}$$

Substituting this in (5), putting $c_p - c_v = c_v \cdot (\gamma - 1)$, and integrating, gives

$$Q = m_a \cdot c_v \cdot \Delta T + m_a \cdot c_v \cdot (\gamma - 1) \cdot \Delta T \cdot \frac{1}{n_c - 1}$$

$$= m_a \cdot c_v \cdot \Delta T \cdot \left(1 - \frac{\gamma - 1}{n_c - 1}\right) \tag{6}$$

Also, $$\frac{T_6}{T_5} = 1 + \frac{\Delta T}{T_5} = \left(\frac{V_5}{V_6}\right)^{(n_c - 1)} = \left(\frac{x_{d_5}}{x_{d_6}}\right)^{(n_c - 1)}$$

$$\Delta T = T_5 \cdot \left[\left(\frac{x_{d_5}}{x_{d_6}}\right)^{(n_c - 1)} - 1\right]$$

Substituting in (6),

$$Q = m_a \cdot c_v \cdot T_5 \cdot \left[\left(\frac{x_{d_5}}{x_{d_6}}\right)^{(n_c - 1)} - 1\right] \cdot \left(1 - \frac{\gamma - 1}{n_c - 1}\right)$$

Substituting $c_v = c_p/\gamma$ and $Q = m_{f56} \cdot HV \cdot \eta$, where m_{f56} is the fuel burned from 5 to 6,

$$m_{f56} \cdot HV \cdot \eta = m_a \cdot \frac{c_p}{\gamma} \cdot T_5 \cdot \left[\left(\frac{x_{d_5}}{x_{d_6}}\right)^{(n_c - 1)} - 1\right] \cdot \left(1 - \frac{\gamma - 1}{n_c - 1}\right)$$

and

$$m_{f56} = \frac{m_a \cdot c_p \cdot T_5}{HV \cdot \eta \cdot \gamma} \cdot \left[\left(\frac{x_{d_5}}{x_{d_6}}\right)^{(n_c - 1)} - 1\right] \cdot \left(1 - \frac{\gamma - 1}{n_c - 1}\right)$$

Any remaining fuel, $m_{f67} = m_f - m_{f56}$, is burned at a constant pressure $P_6 = P_7$.

$$m_{f67} \cdot HV \cdot \eta = m_a \cdot c_p \cdot (T_7 - T_6)$$

But
$$T_7 = \frac{P_6 \cdot A_d \cdot x_{d_7}}{m_a \cdot R}$$

and
$$T_6 = \frac{P_6 \cdot A_d \cdot x_{d_6}}{m_a \cdot R}$$

Therefore
$$T_7 - T_6 = \frac{P_6 \cdot A_d}{m_a \cdot R} \cdot (x_{d_7} - x_{d_6})$$

Substituting,
$$m_{f67} \cdot HV \cdot \eta = m_a \cdot c_p \cdot \frac{P_6 \cdot A_d}{m_a \cdot R} \cdot (x_{d_7} - x_{d_6})$$

and
$$x_{d_7} = x_{d_6} + \frac{m_{f67} \cdot HV \cdot \eta \cdot R}{P_6 \cdot A_d \cdot c_p} \tag{7}$$

The temperature at the end of combustion,
$$T_7 = \frac{P_6 \cdot A_d \cdot x_{d_7}}{m_a \cdot R}$$

But
$$m_a = \frac{P_4 \cdot A_d \cdot x_{d_4}}{R \cdot T_4}$$

Hence,
$$T_7 = \frac{P_6 \cdot A_d \cdot x_{d_7} \cdot R \cdot T_4}{P_4 \cdot A_d \cdot x_{d_4} \cdot R}$$

$$= T_4 \cdot \frac{P_6 \cdot x_{d_7}}{P_4 \cdot x_{d_4}} \tag{8}$$

For a given setting of the injection pump timing, the pump delivery will begin at a fixed distance before IML, in terms of piston travel. However, a study of indicator cards taken during test runs on the engine suggests that, regardless of load, the distance from the beginning of heat release to IDP is nearly constant for a given setting of injection timing. This phenomenon has not been fully explained, but is presumably connected with the variations in injection and ignition delay with variations in speed and load. See also Reference 4.

In the model, therefore, it has been decided to select the point at which heat release begins on the basis of IDP rather than of IML. Timing is therefore set by selecting a distance Δx_5 (typically 0.1 in) from the beginning of heat release to IDP. Since the next IDP is not known at the time the value of x_{d_5} is required and the variations in IDP from one cycle to the next are small, the current value of x_{d_5} is obtained by adding Δx_5 to the previous IDP. In order to avoid hunting, it was found necessary in practice to allow a small tolerance so that a new value of x_5 would be generated only if the change in IDP exceeded a preset small amount, typically 0.003 in.

It is thus possible to study the effect of timing by varying Δx_5 and of the rate of heat release before IDP by varying n_c. The results of this study are illustrated in Figure 5, which also gives a comparison between the two combustion models, since Combustion Model I is in force when the abscissa is zero (beginning of heat release at IDP). Examination of Figure 5 suggests that an optimum timing position exists with the beginning of heat release about 0.07 inches before IDP, or 0.32 inches before IML. Unfortunately, only a limited study of the effect of timing in the real engine was made, since the beginning of the pump delivery stroke could not be advanced beyond 1.2 inches before IML. At this point, the specific fuel consumption appeared to have levelled off, although horsepower was still increasing with advancing timing. Because of the unknown injection and ignition delays, it is not possible to make a direct comparison of timing figures for the engine and the simulation.

Simulation performance with the two combustion models is compared in Figure 6. The dependence of IDP on load is only very slightly greater with Model II than Model I, and the new model, at least with constant values of Δx_5 and n_c, cannot be said to be significantly superior to the old. It may be inferred from Figure 5 that even with varying Δx_5 and n_c, the effect on IDP is not likely to be strong.

Aside from the absence of any clear superiority of Model II combustion over Model I, an additional argument against its adoption lies in the tendency toward instability exhibited in a hybrid simulation of another free piston engine concept using the same combustion model. In this case, the power take-off from the engine was by means of a mechanically coupled hydraulic pump, and this, together with the external load assumed, produced an inherently unstable system. It was almost impossible to run this model until the Model II combustion was replaced by Model I.

3.2 Friction

Experimental data on friction in diesel engines are singularly scarce (Reference 5), and for free piston engines appear to be totally lacking. In the case of the crankshaft engine, one can gain some insight into the magnitude of friction by means of a motoring test, but with the free piston machine this is not possible. The results of motoring tests must, in any case, by applied with extreme caution because the conditions during such a test are quite different from those that apply in a running engine.

Probably the best that one can do in the case of the free piston engine is to estimate the friction forces. Most of these are caused by ring friction in the diesel, compressor and bounce cylinders, with some additional losses in the synchronising mechanism and fuel pump drive. The sliding friction of rings against cylinder walls is affected by such operating conditions as temperature, pressure, and lubrication, and may vary considerably during any one cycle, as well as over the engine operating range.

In the earlier work (References 1 and 6) Coulomb friction only was modelled; piston motion was opposed by a friction force of constant magnitude but with its sign changing each time the piston assembly reversed direction. This oversimplification is now refined following Millington et al.[5] by the addition of a viscous friction term proportional to piston velocity.

The equation for friction force then assumes the form:

$$F_f = -F_c \cdot \mathrm{sgn}(\dot{x}) - c \cdot \dot{x} \tag{9}$$

where F_c is the assumed Coulomb friction force,

c is an assumed factor for viscous friction,

and $\dot{x}$ is the instantaneous piston speed.

A comparison of engine performance with and without viscous friction is given in Figure 7. Based on a typical mean piston speed of 1500 ft/min, the mean viscous friction force here amounts to about one-sixth of the total. Figure 8 shows performance as a function of the viscous friction factor c.

A study of Figures 7 and 20 suggests that the effect of viscous friction is nearly identical to that of Coulomb friction. This result is not entirely unexpected, since the frequency dependence of the viscous friction loss is largely nullified by the narrow frequency range of the free piston engine.

3.3 Effect of Small Changes

It is of interest to know the effects on engine performance of small changes in the operating conditions, such as ambient temperature and pressure, and in such estimated quantities as friction and leakage factors.

These influences were studied by the method used in Reference 6, varying each of the selected quantities in turn, and noting the effect on engine performance. The results of this study are given in Table 2. A number in the main body of the table gives the percentage change in the selected performance criterion for a one percent change in the engine variable in question. Thus, for example, a one percent increase in bounce reference pressure is seen to cause a decrease of almost 0.2% in gas horsepower and an increase of nearly 0.4% in frequency; and a small change in ambient pressure has no measurable effect at all on ODP or specific fuel consumption. The same results are presented in graphic form in Figures 9 to 20. In each of these figures, the standard value is indicated by a vertical line.

The strongest interactions are found between ambient pressure and IDP, gas horsepower, mass flow and fuel flow, and between bounce reference pressure and IDP. The assumed polytropic exponent for expansion in the compressor cylinder also has a strong effect on IDP, but the position of the beginning of heat release (injection timing in the real engine) has almost no influence. This bears out the observation made in 3.2 above, and suggests that improvement in the simulation of IDP must be sought elsewhere than in the combustion model. Decreasing the bounce reference pressure with increasing load may be justified on physical grounds, but can be applied only to a very limited extent because of the fairly strong dependence of frequency on bounce pressure level.

3.4 Stability and Receiver Volume

There is always a time lag of a number of cycles between a change in running conditions and the attainment of a new equilibrium running point. This is particularly evident each time the simulation is started.

Figure 21 presents the results of an experiment in which readings were taken at intervals after simulation starts with various receiver volumes, but with the engine controls and ambient conditions held constant. As expected, stable running was achieved in fewer cycles with the smaller receiver volumes. It is noteworthy, however, that the same equilibrium point was reached in each case, regardless of receiver volume. This suggests that an economy in simulation running time may be effected by reducing the receiver volume to a value lower than that used in the real engine, if the interest is only in steady-state results.

A receiver volume of 2.0 ft^3 was selected as the standard value for subsequent tests, and a stabilization period of 100 cycles provided an adequate margin of error for the attainment of repeatable readings.

4.0 CONCLUSIONS

1. A combustion model has been produced that permits the evaluation of combustion effects prior to IDP. However, performance is not strongly affected by a considerable simulated advance in injection timing.

2. Further experimental data would be essential for a detailed assessment of combustion modelling.

3. An improved piston friction model has been produced that permits the incorporation of viscous components along with Coulomb friction.

4. A study of the effects of small changes indicates that it is possible to predict the trend and character of variations in performance as the result of small changes in operating conditions.

5. Stability tests indicate that simulation performance is not affected by reasonable reductions in the receiver volume in order to avoid excessive running times.

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TABLE 2

EFFECTS OF SMALL CHANGES

PARAMETER	STANDARD OPERATING VALUE	UNITS	IDP	ODP	GAS HORSEPOWER	FREQUENCY	SFC	GAS DELIVERY TEMPERATURE	MASS FLOW	FUEL FLOW	SEE ALSO FIGURE	REMARKS
			in	in	hp	min^{-1}	$lb_m/ghph$	°K	lb_m/s	lb_m/h	No.	
Ambient Pressure	14.70	lb_f/in^2	1.11	-0.01	1.00	0.20	0.00	0.02	1.00	1.02	9	Constant delivery pressure ratio
Ambient Temperature	288	°K	-0.28	0.08	0.49	0.02	0.02	0.91	-0.48	0.53	10	Constant delivery pressure = 46 lb_f/in^2
Diesel Efficiency	0.75	-	0.00	0.00	0.00	0.00	-1.12	0.00	-0.03	-1.09	11	Constant delivery pressure = 46 lb_f/in^2
Leakage Factor	0.77	-	0.27	-0.15	-0.22	-0.01	-0.32	-0.48	0.27	-0.56	12	Constant delivery pressure = 46 lb_f/in^2
Compressor Polytropic Exponent: Compr.	1.27	-	0.00	-0.06	0.43	0.04	0.47	0.63	-0.30	0.78	13	Constant delivery pressure = 46 lb_f/in^2
Compressor Polytropic Exponent: Exp.	1.27	-	0.99	-0.16	-0.08	-0.19	-0.15	-0.12	0.12	-0.27	13	Constant delivery pressure = 46 lb_f/in^2
Diesel Combustion Exponent	2.50	-	-0.22	-0.03	-0.08	0.03	0.07	0.06	-0.06	0.04	14	Constant fuel per cycle
Beginning of Heat Release	0.10	in from IDP	0.09	-0.00	-0.00	0.00	0.01	0.01	-0.02	0.01	15	Constant fuel per cycle
Bounce Reference Pressure	60	lb_f/in^2	-1.33	-0.08	-0.07	0.36	-0.09	-0.09	0.07	-0.15	16	Constant delivery pressure = 46 lb_f/in^2
Compressor Valve Loss: Deliv.	15	%	-0.09	0.00	0.02	0.04	0.03	0.04	-0.02	0.05	17	Constant fuel per cycle
Compressor Valve Loss: Inlet	5	%	0.00	0.01	-0.04	-0.01	0.03	0.02	-0.04	-0.01	18	Constant fuel per cycle
Coulomb Friction	160	lb_f	0.15	-0.01	0.07	-0.02	0.08	0.11	-0.06	0.13	19	Constant fuel per cycle
Viscous Friction Coefficient	30*	$lb_f s/in$	0.13	-0.01	-0.05	-0.01	0.02	0.02	-0.04	-0.01	20	Constant fuel per cycle

* Although the simulation was normally run without viscous friction, values in the table are based on a datum of 30 $lb_f s/in$.

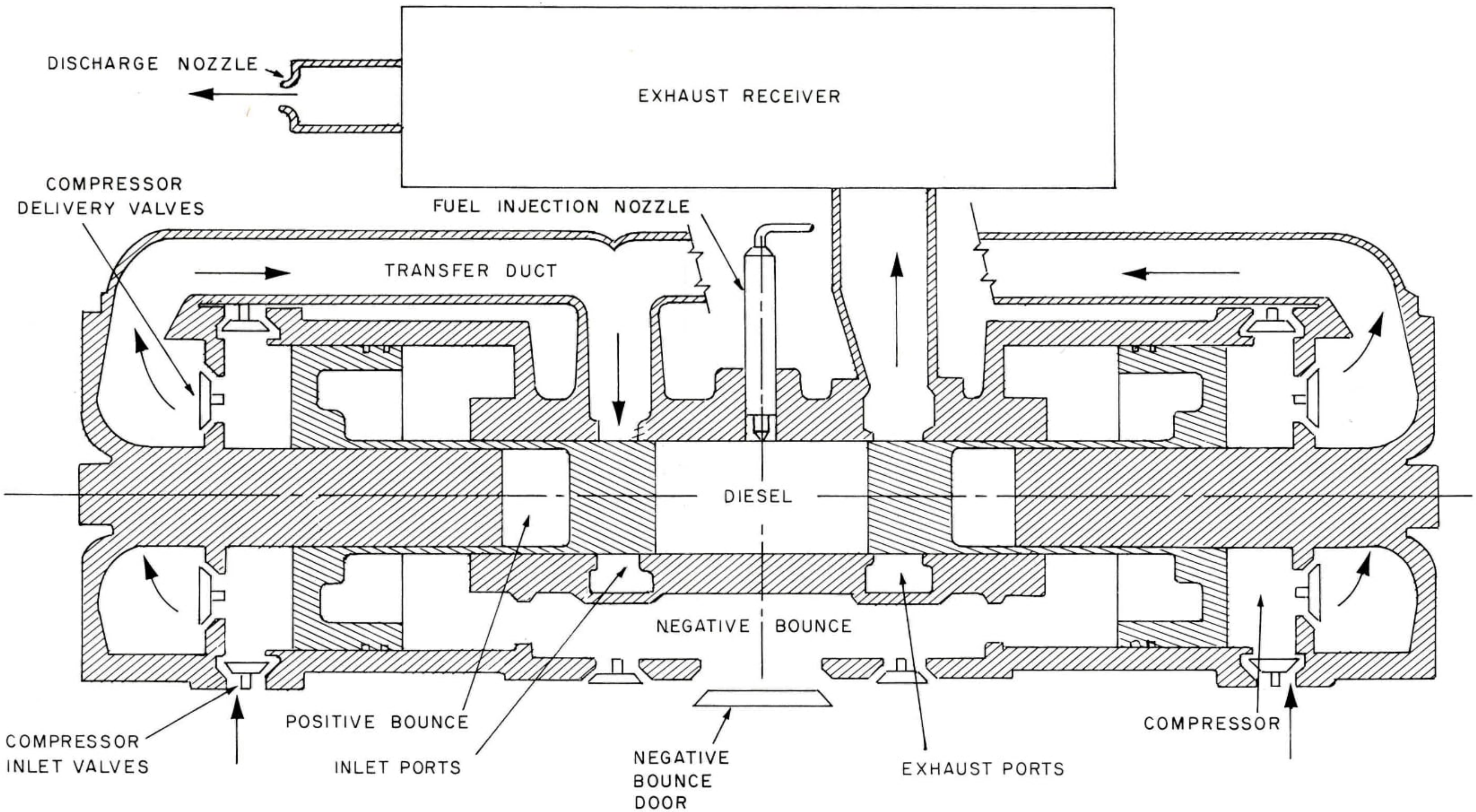

FIG.I: LONGITUDINAL SECTION OF ENGINE

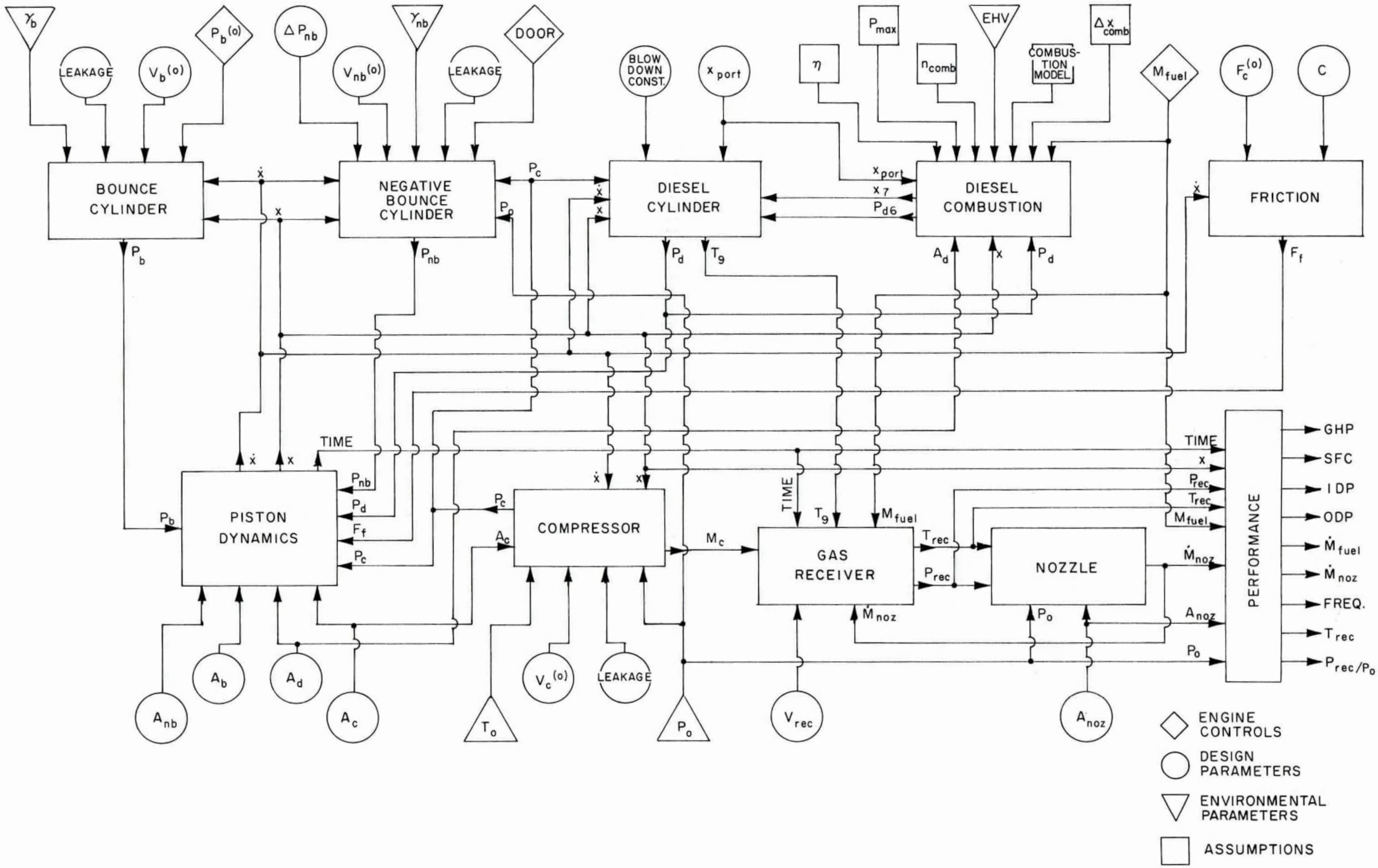

FIG. 2: INFORMATION FLOW DIAGRAM

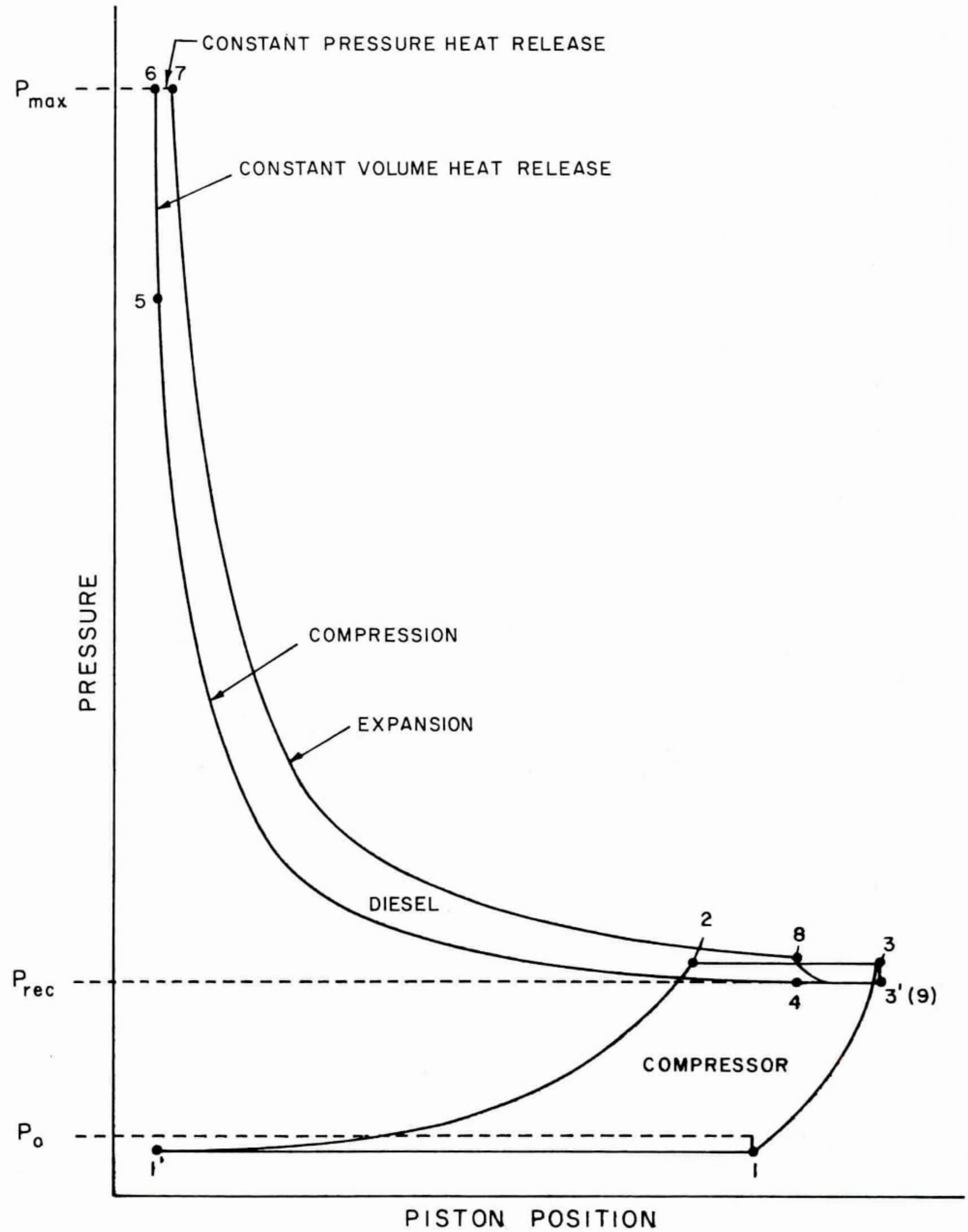

FIG. 3: DIESEL ENGINE CYCLE WITH MODEL I COMBUSTION

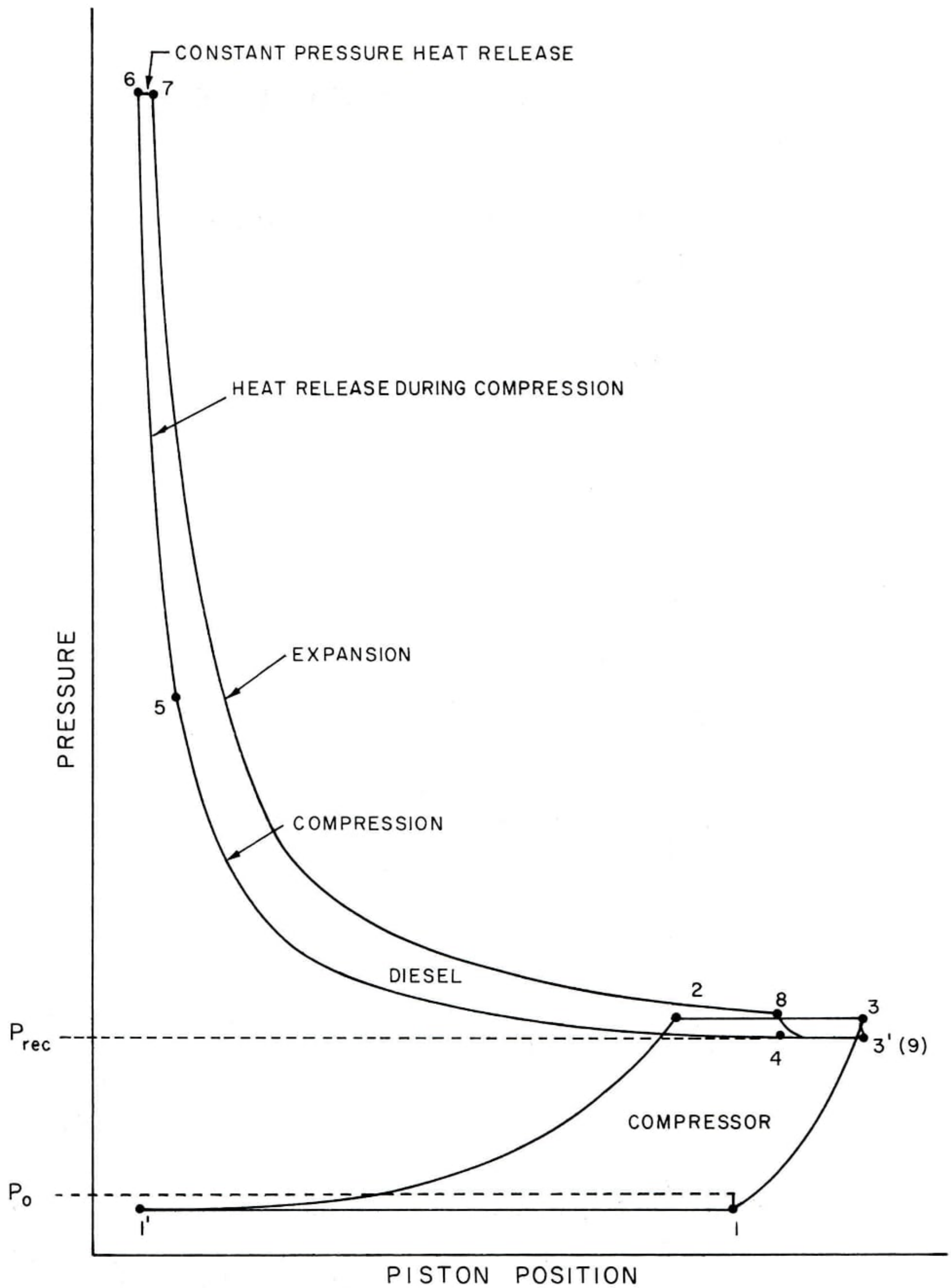

FIG. 4: DIESEL ENGINE CYCLE WITH MODEL II COMBUSTION

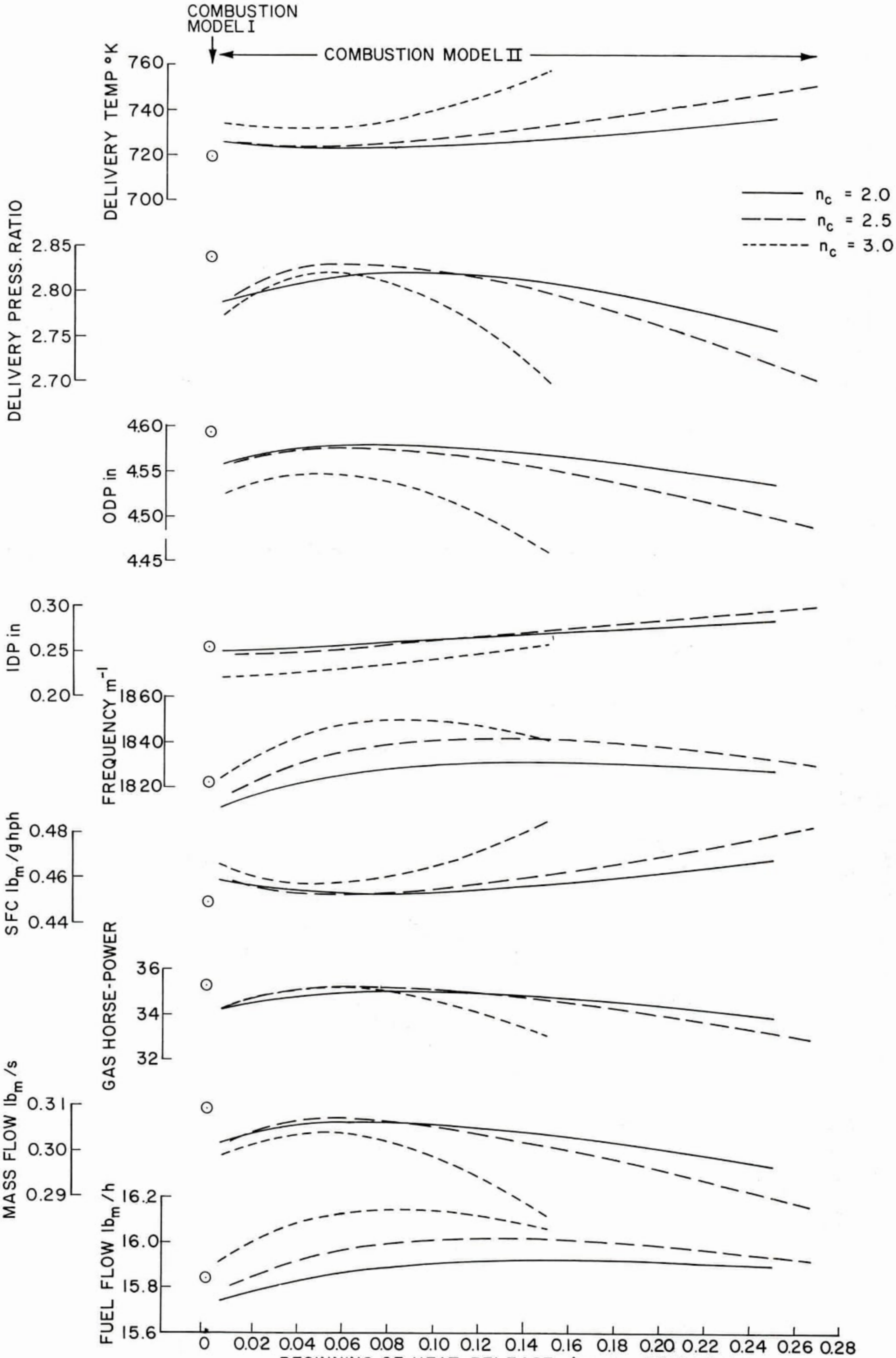

FIG. 5: INJECTION TIMING

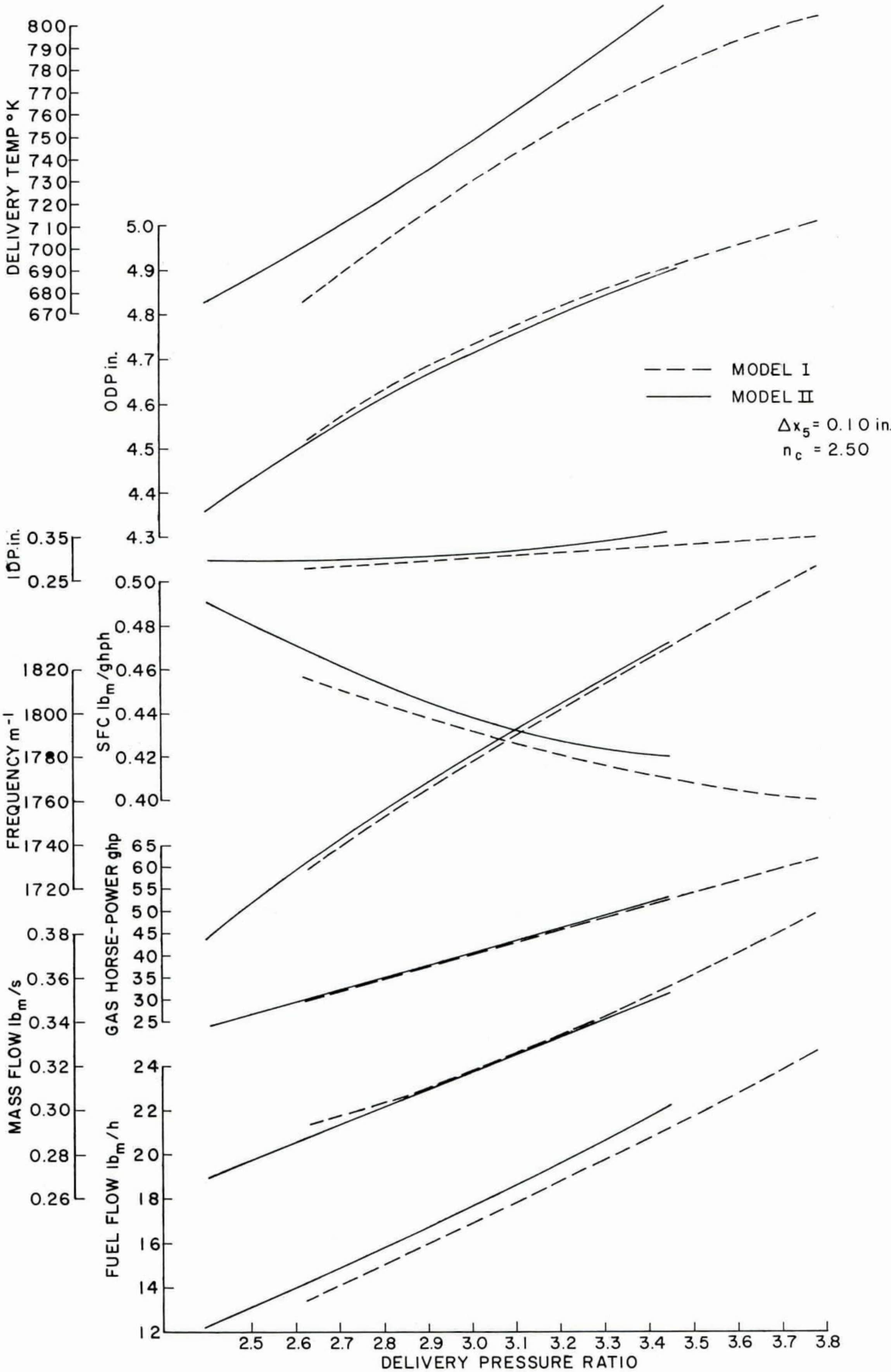

FIG.6: COMPARISON OF COMBUSTION MODELS I AND II

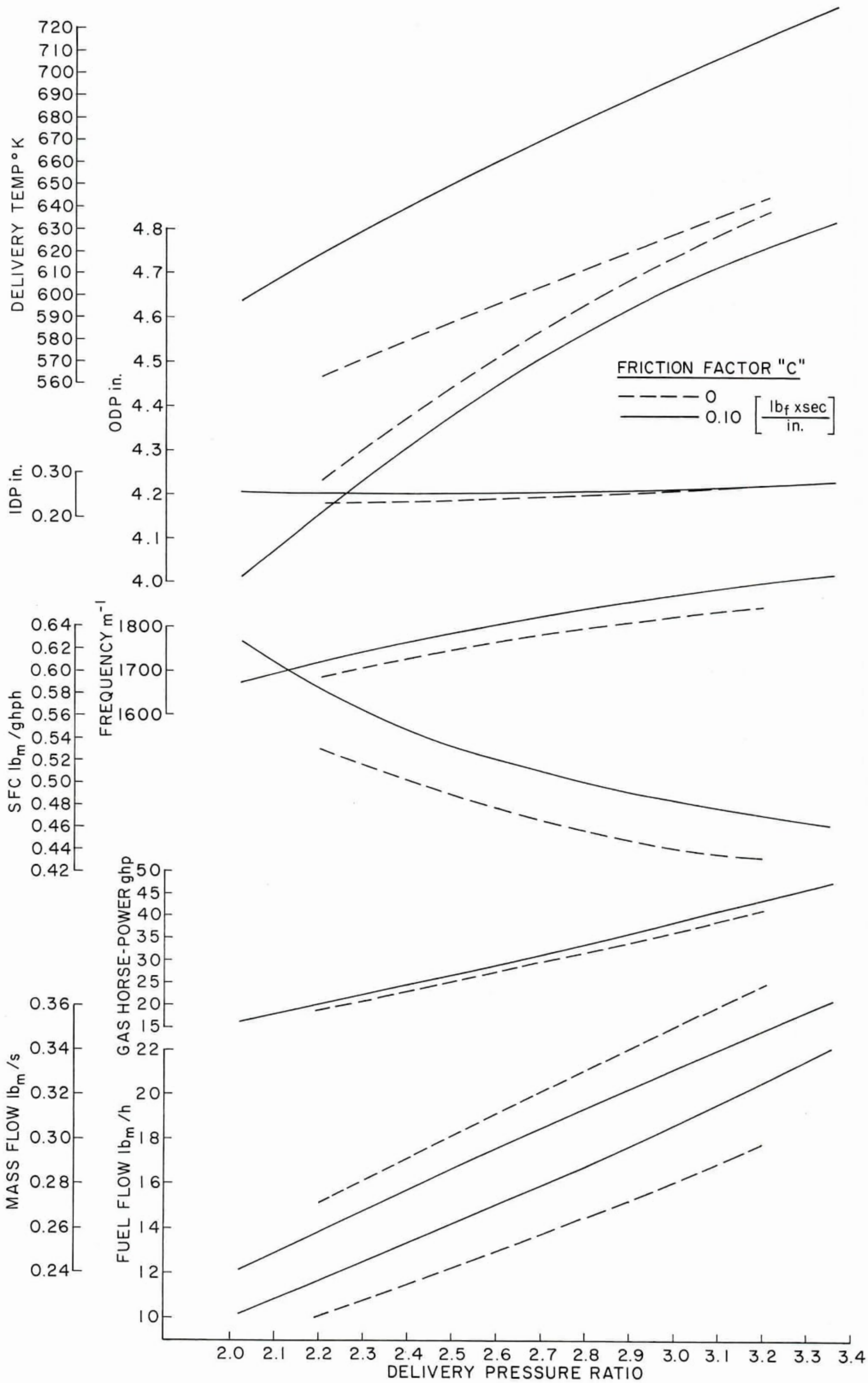

FIG. 7: PERFORMANCE WITH AND WITHOUT VISCOUS FRICTION

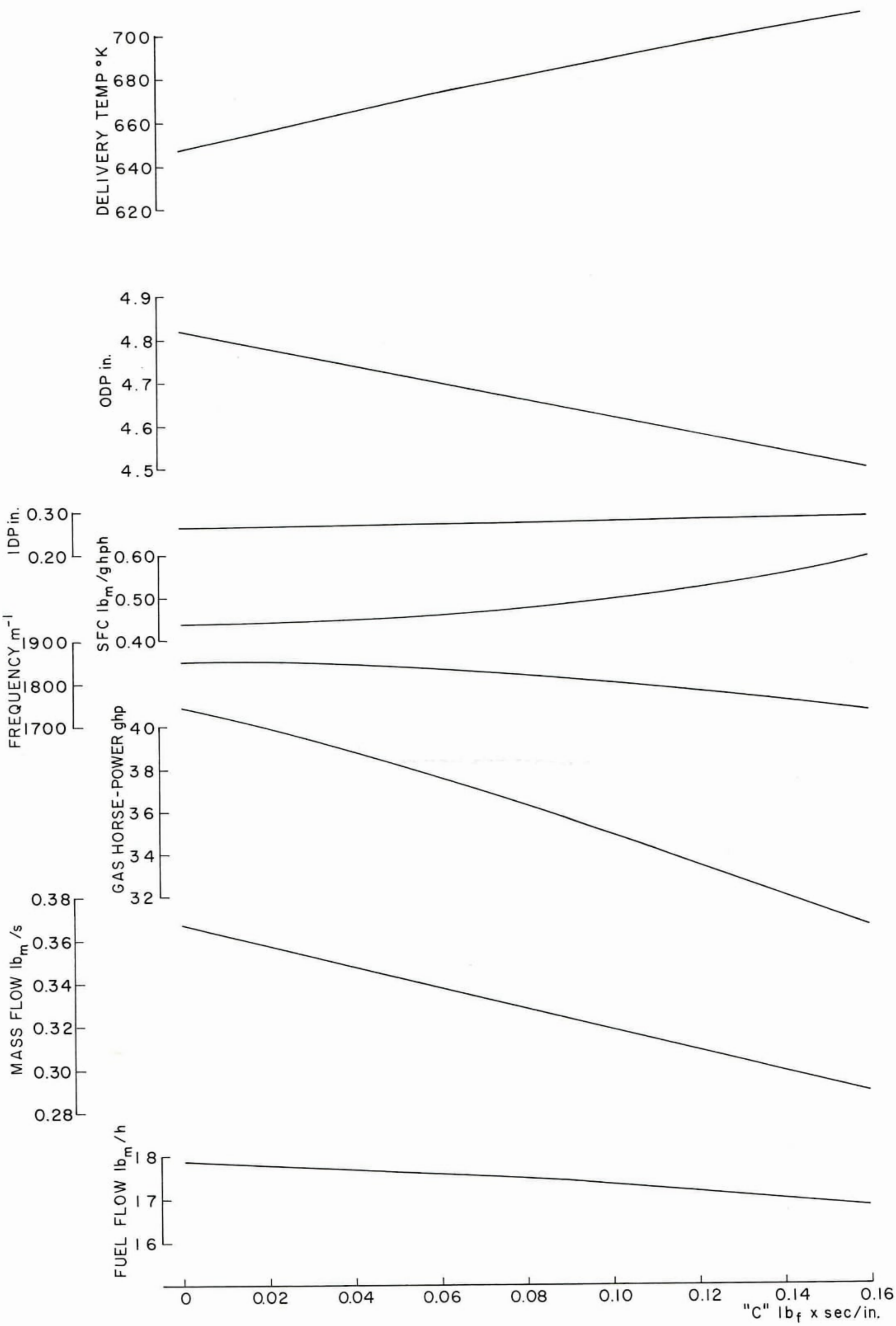

FIG.8: VISCOUS FRICTION COEFFICIENT

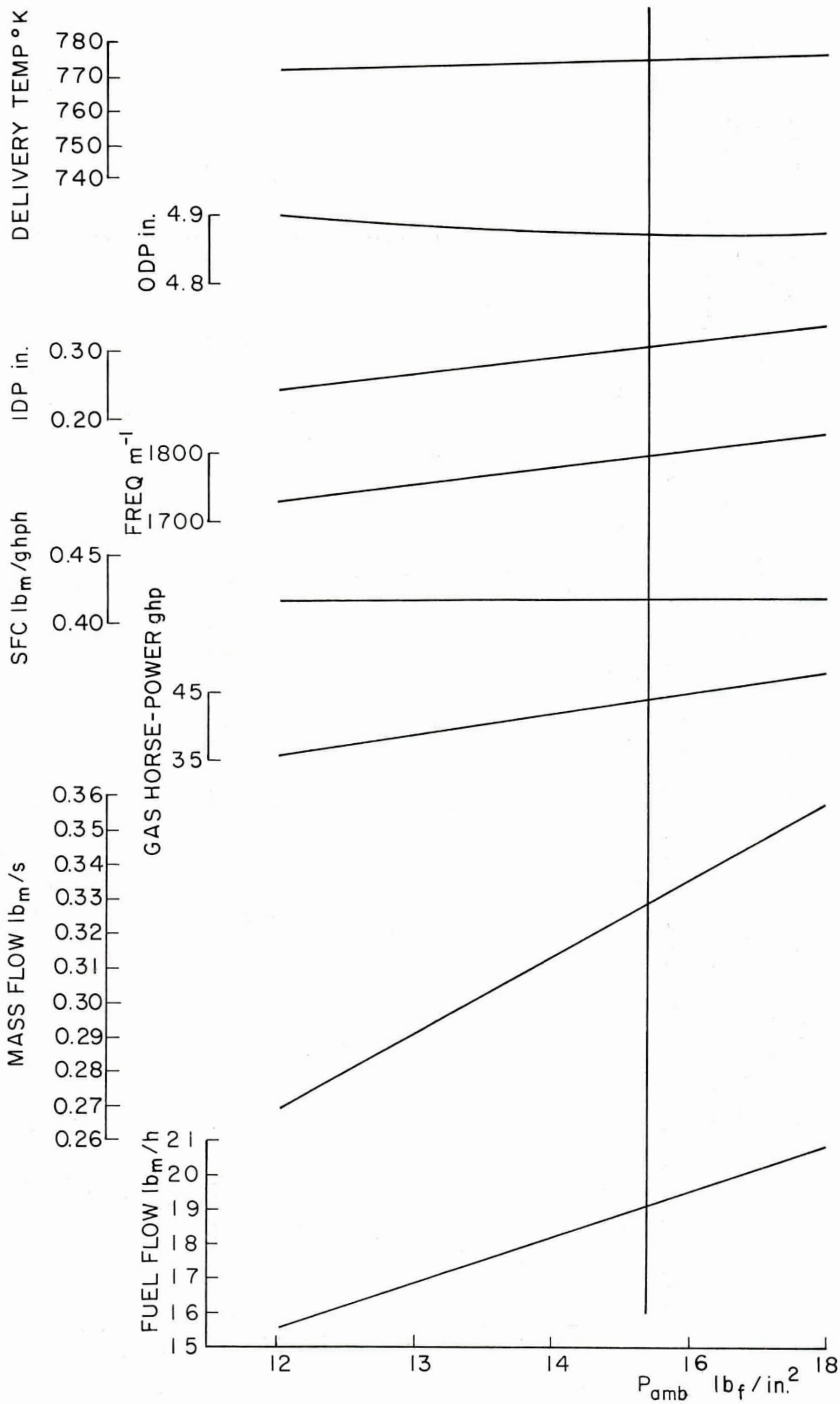

FIG. 9: EFFECT OF SMALL CHANGES : AMBIENT PRESSURE

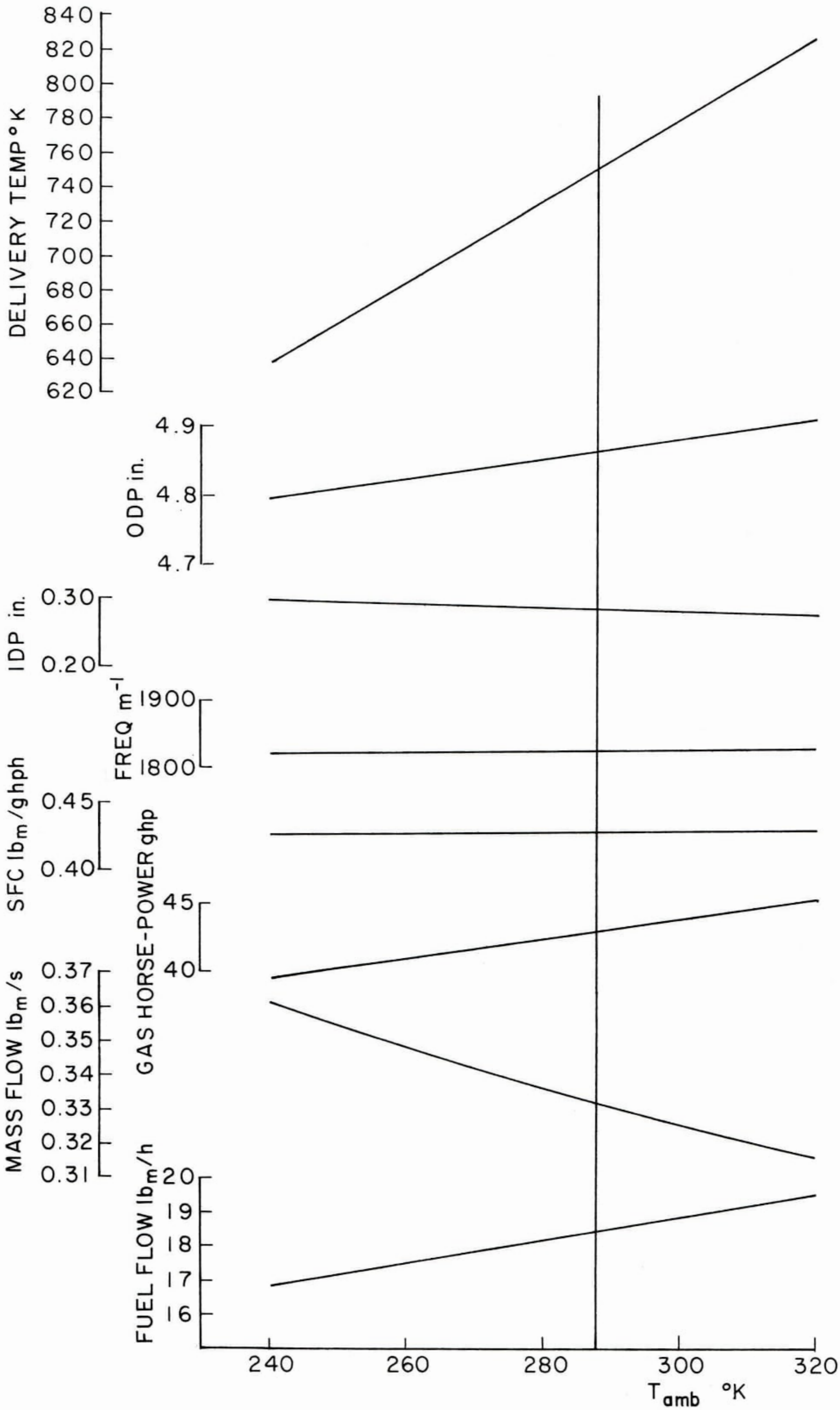

FIG.10: EFFECT OF SMALL CHANGES : AMBIENT TEMPERATURE

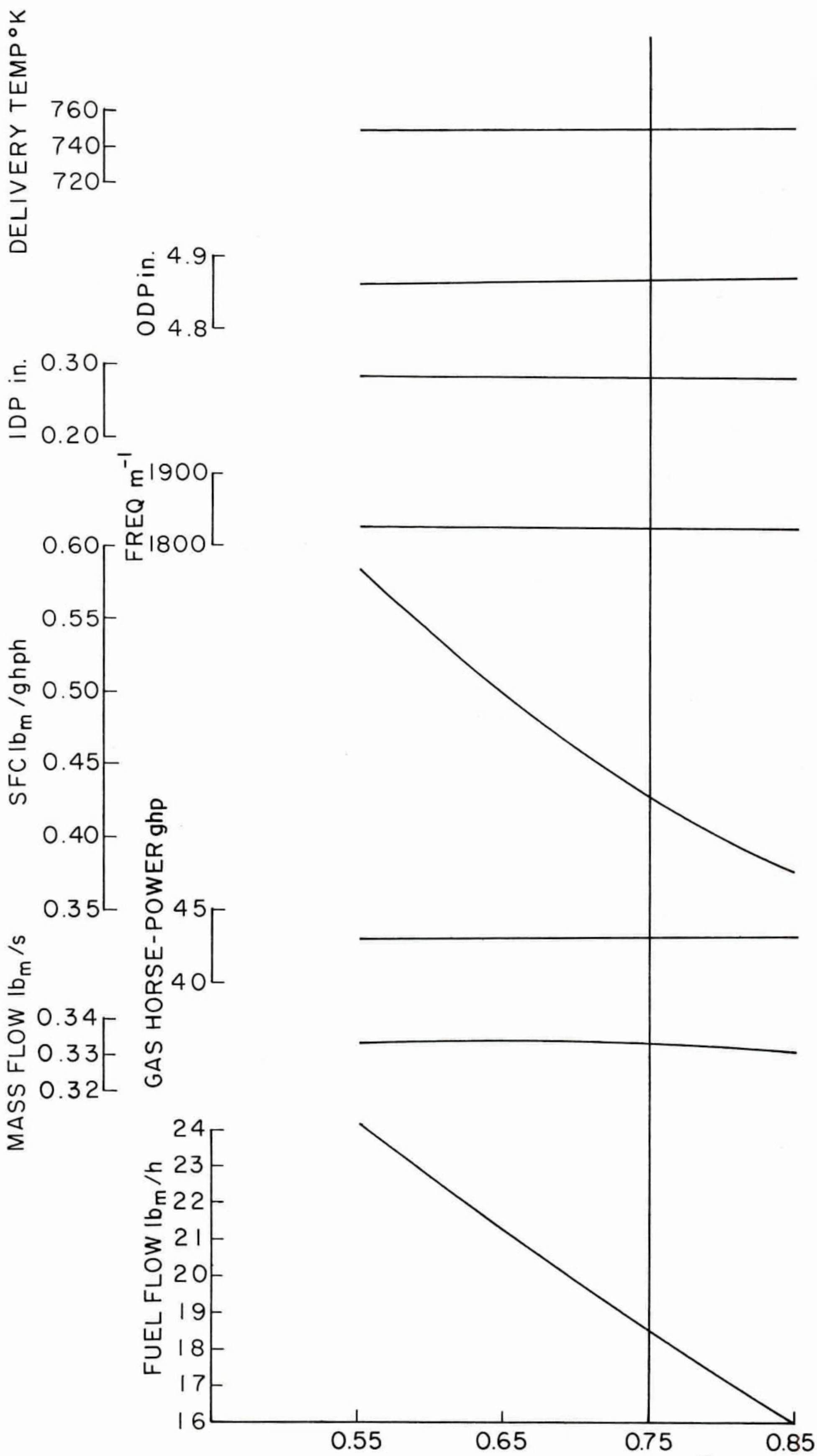

FIG.11: EFFECT OF SMALL CHANGES: DIESEL EFFICIENCY

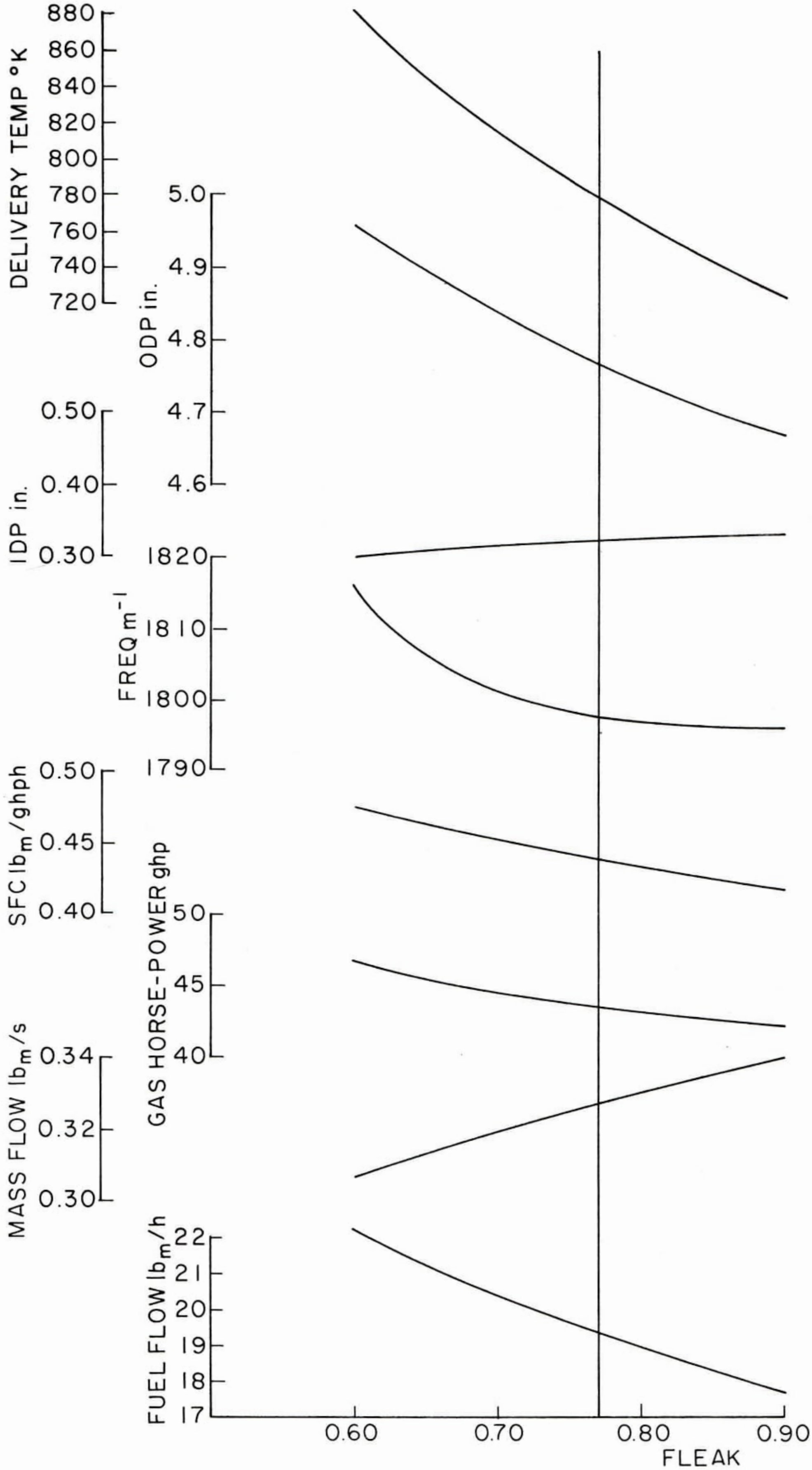

FIG.12: EFFECT OF SMALL CHANGES: LEAKAGE FACTOR

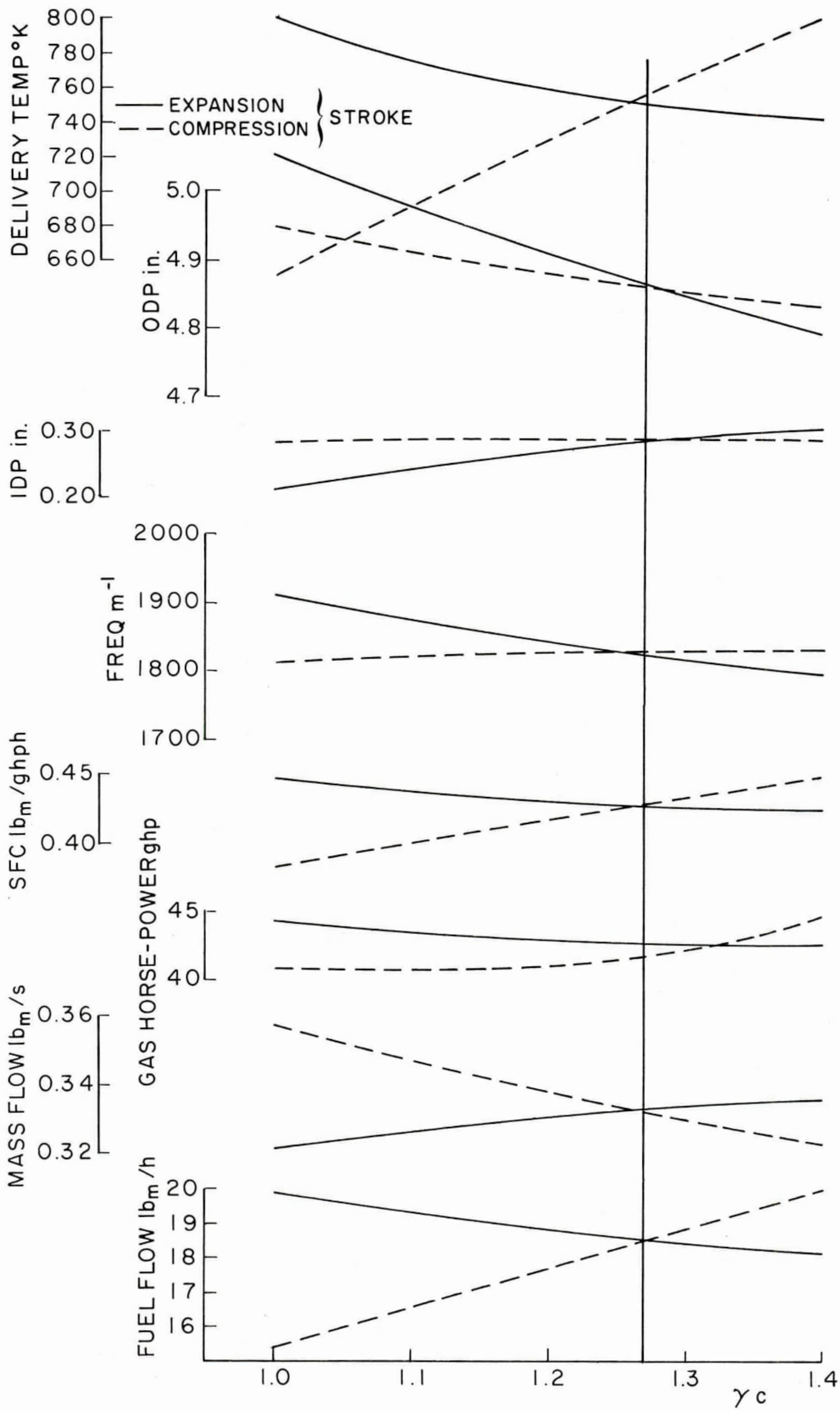

FIG.13: EFFECT OF SMALL CHANGES: COMPRESSOR POLYTROPIC EXPONENT

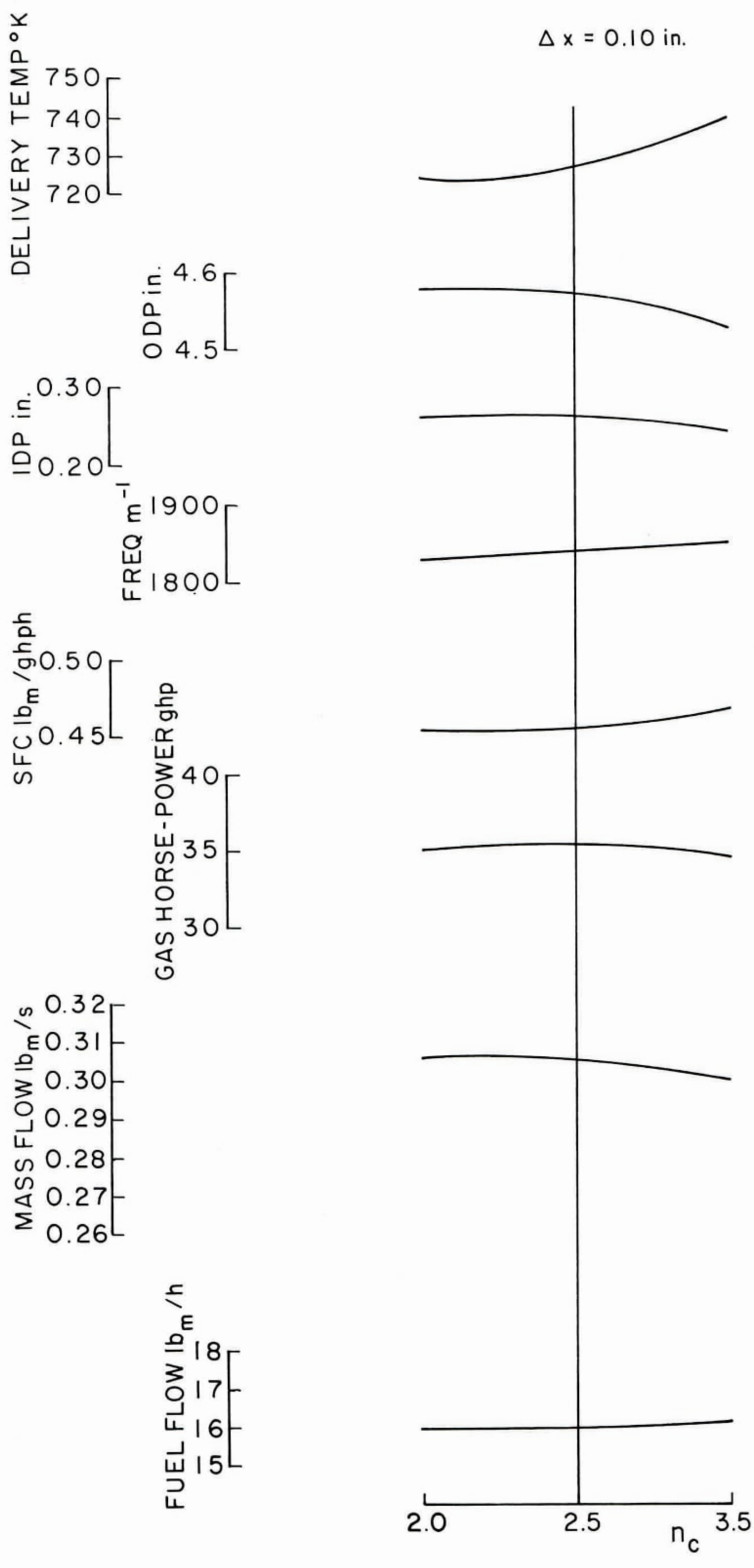

FIG.14: EFFECT OF SMALL CHANGES: COMBUSTION EXPONENT

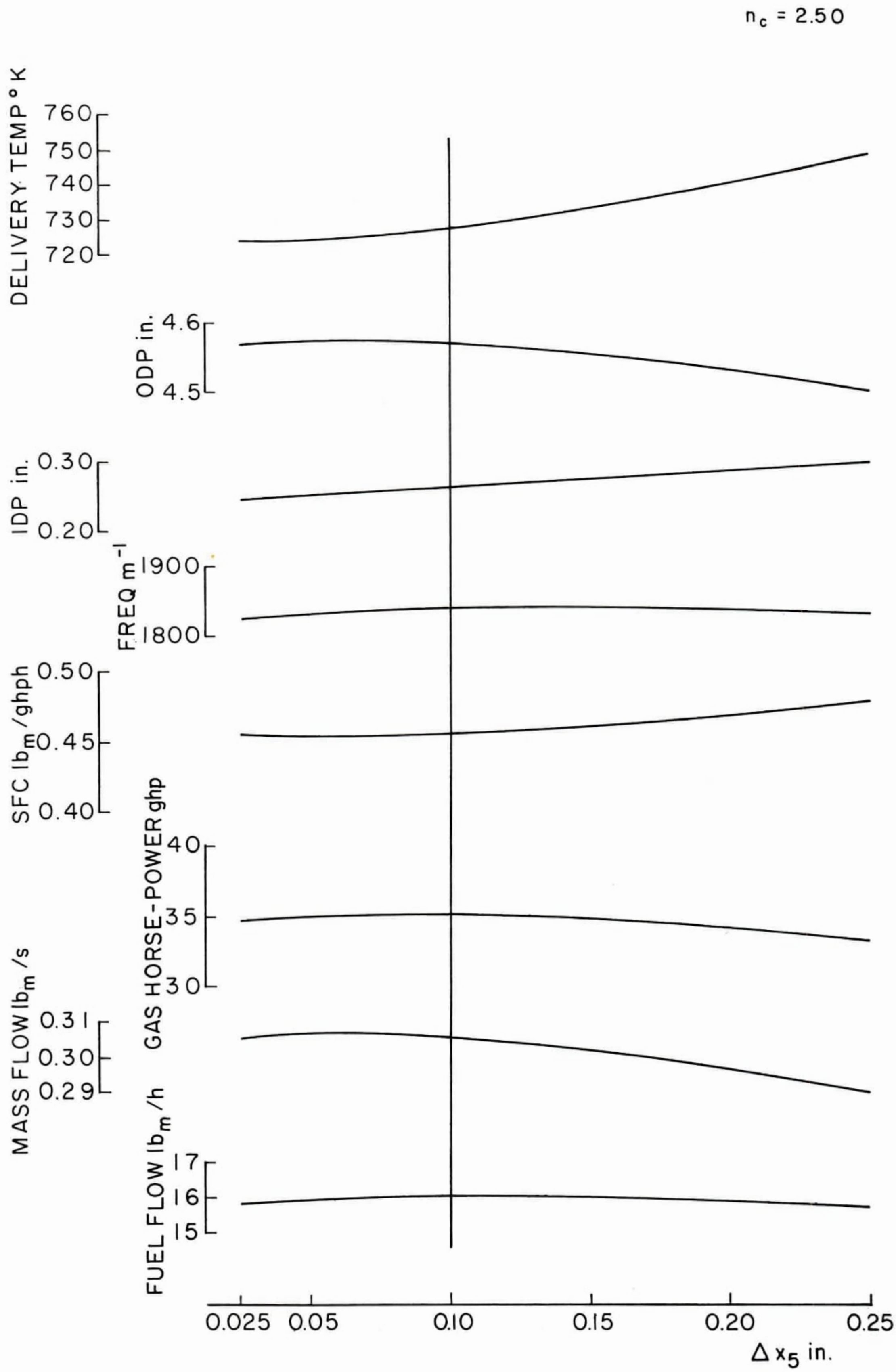

FIG.15: EFFECT OF SMALL CHANGES : TIMING

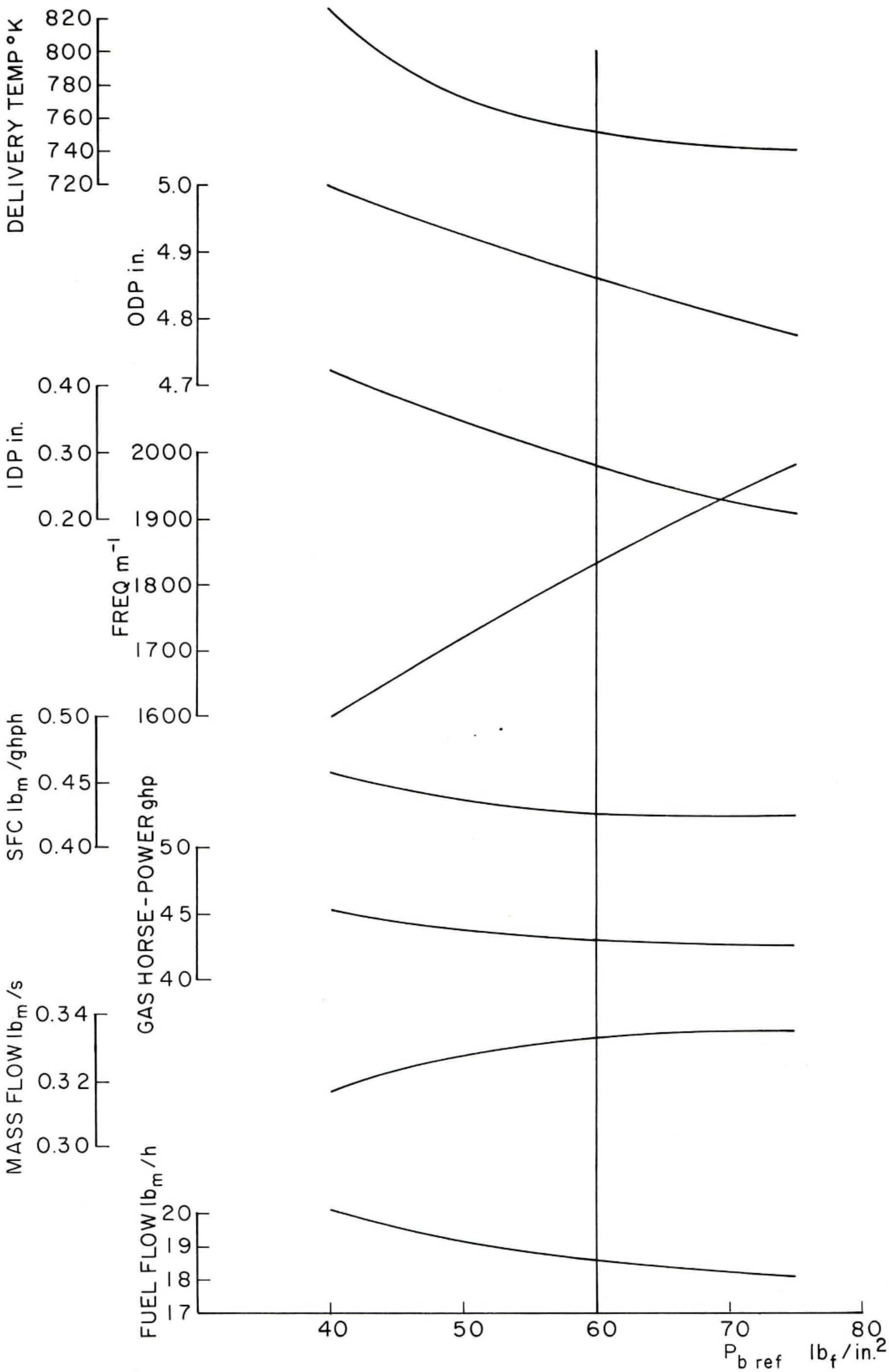

FIG.16: EFFECT OF SMALL CHANGES: BOUNCE REFERENCE PRESSURE

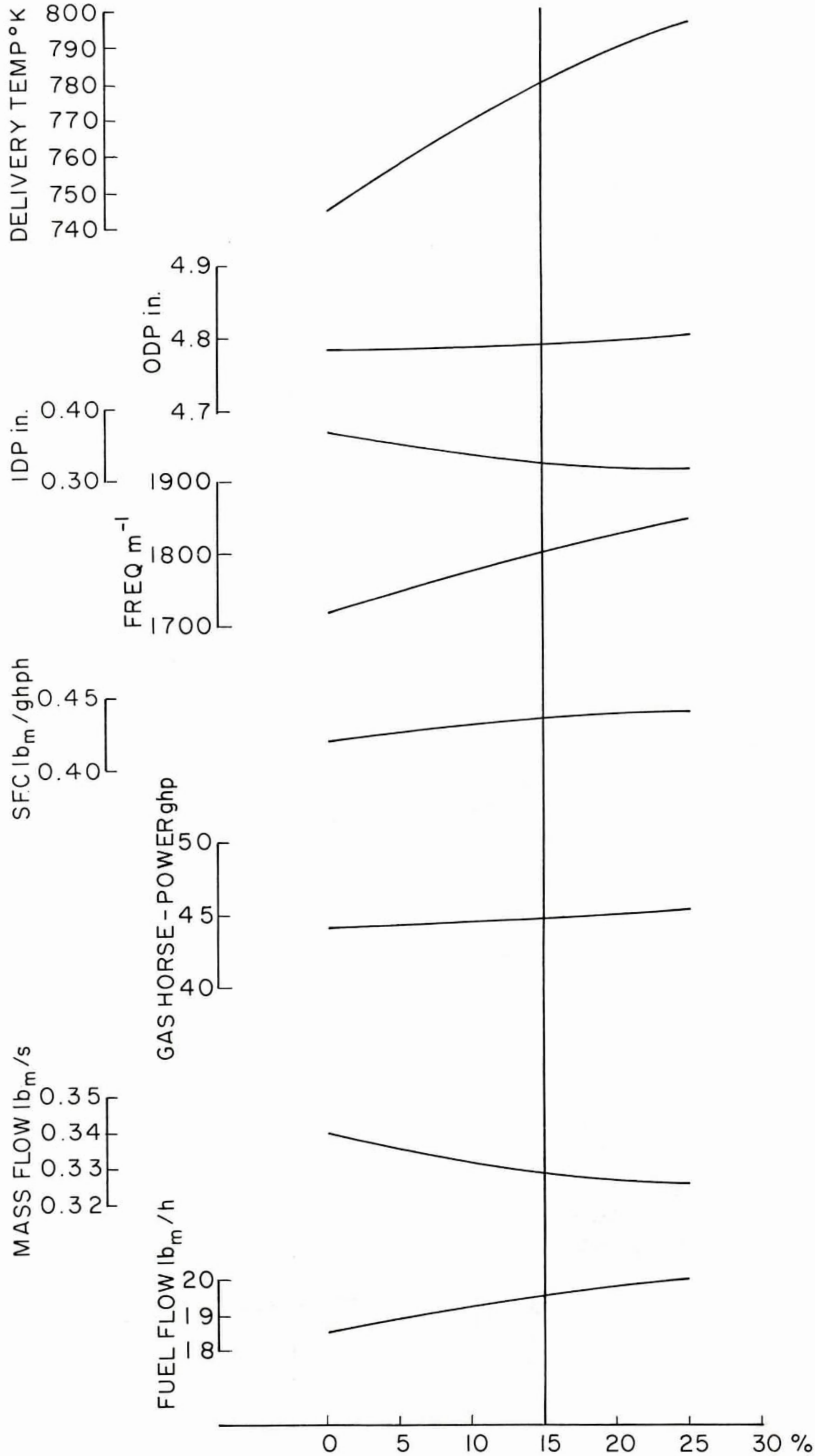

FIG.17: EFFECT OF SMALL CHANGES: COMPRESSOR DELIVERY VALVE LOSS

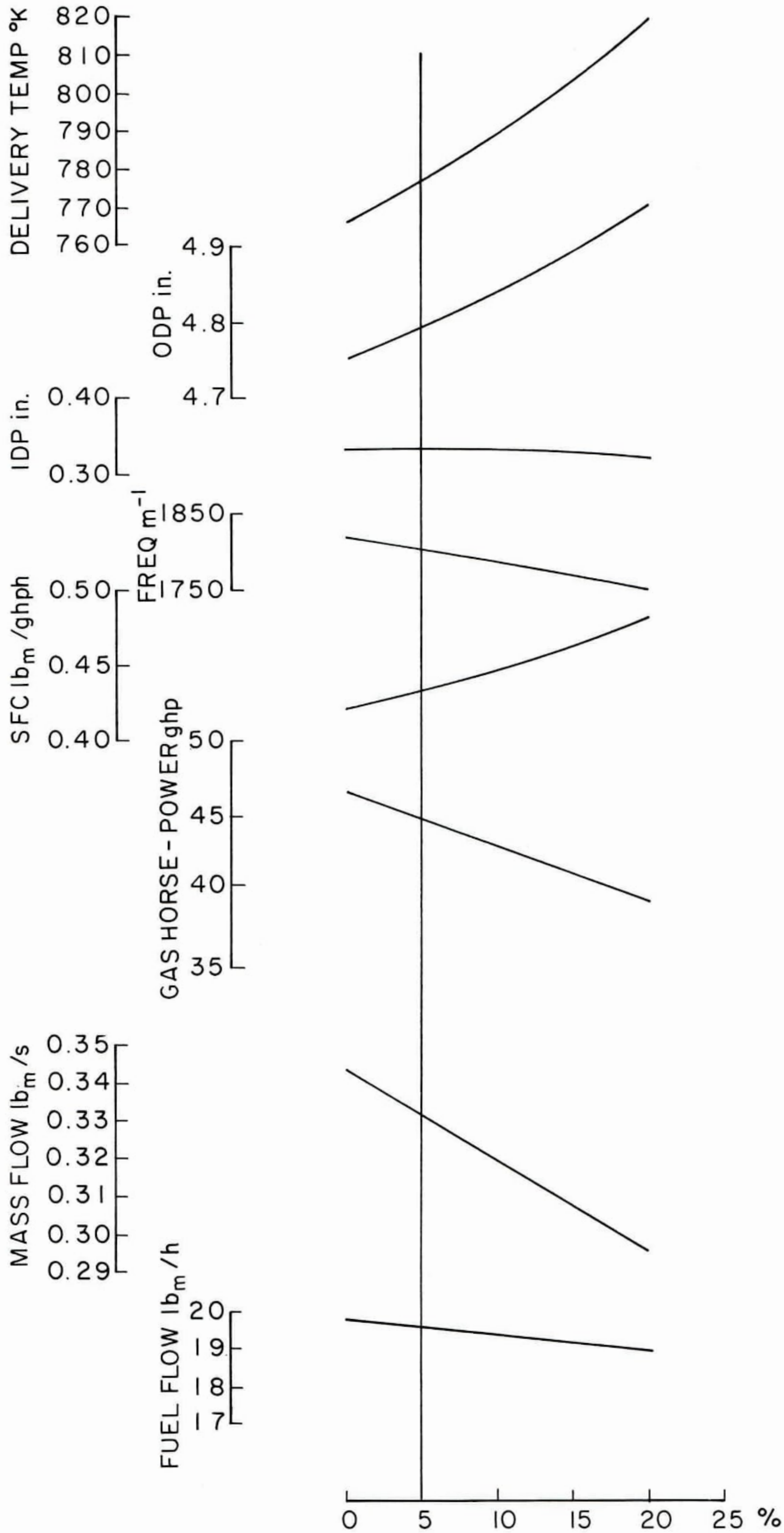

FIG.18: EFFECT OF SMALL CHANGES: COMPRESSOR INLET VALVE LOSS

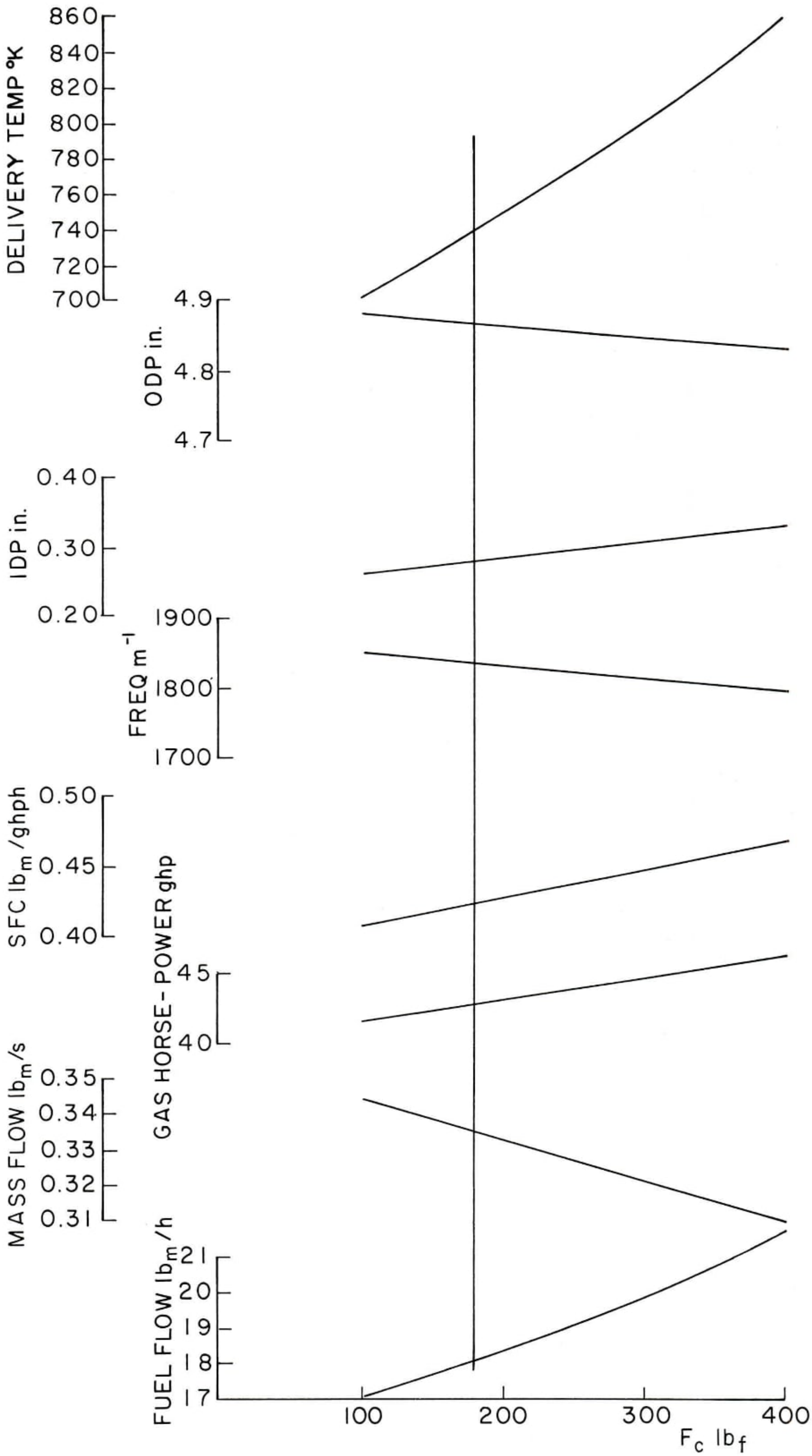

FIG.19: EFFECT OF SMALL CHANGES: COULOMB FRICTION

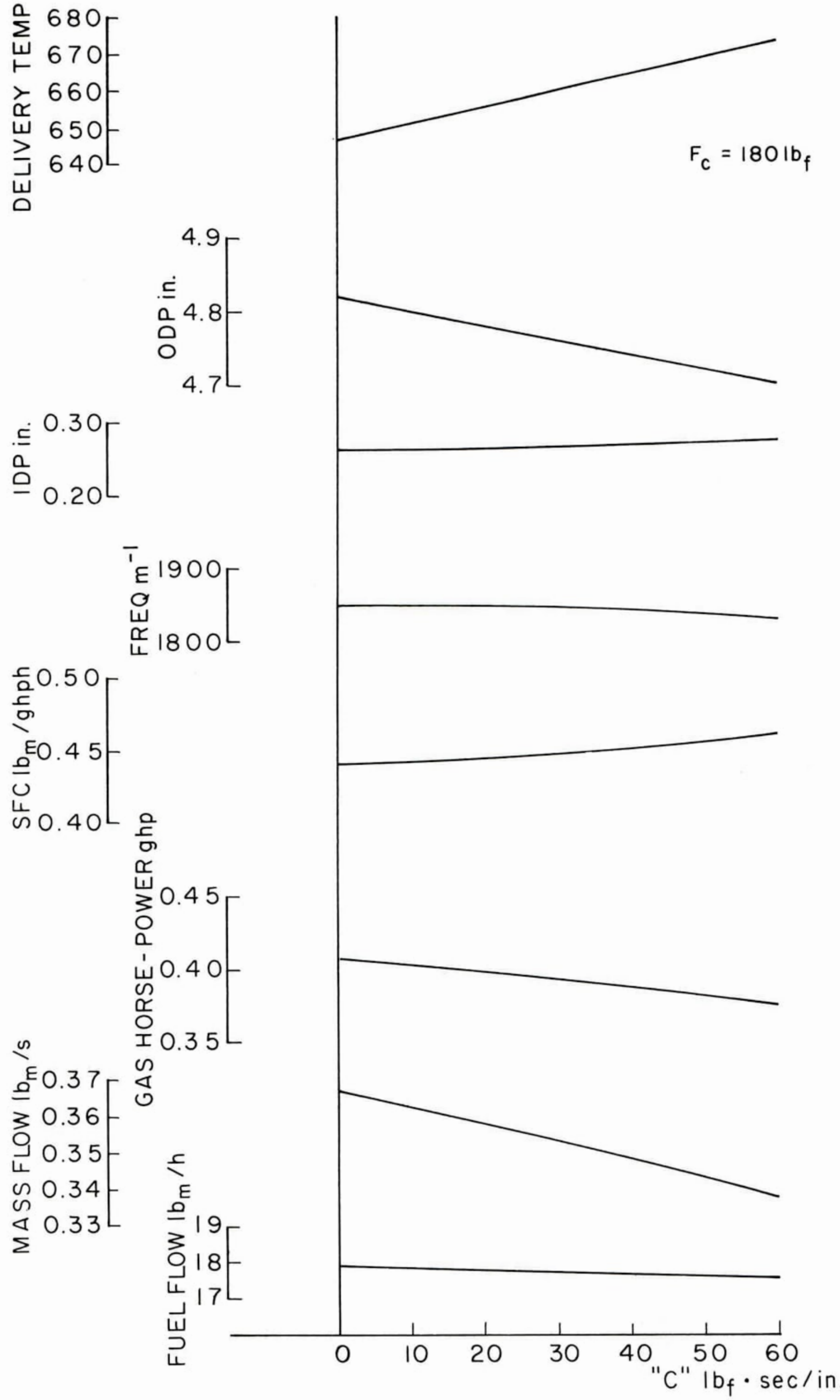

FIG.20: EFFECT OF SMALL CHANGES: VISCOUS FRICTION COEFFICIENT

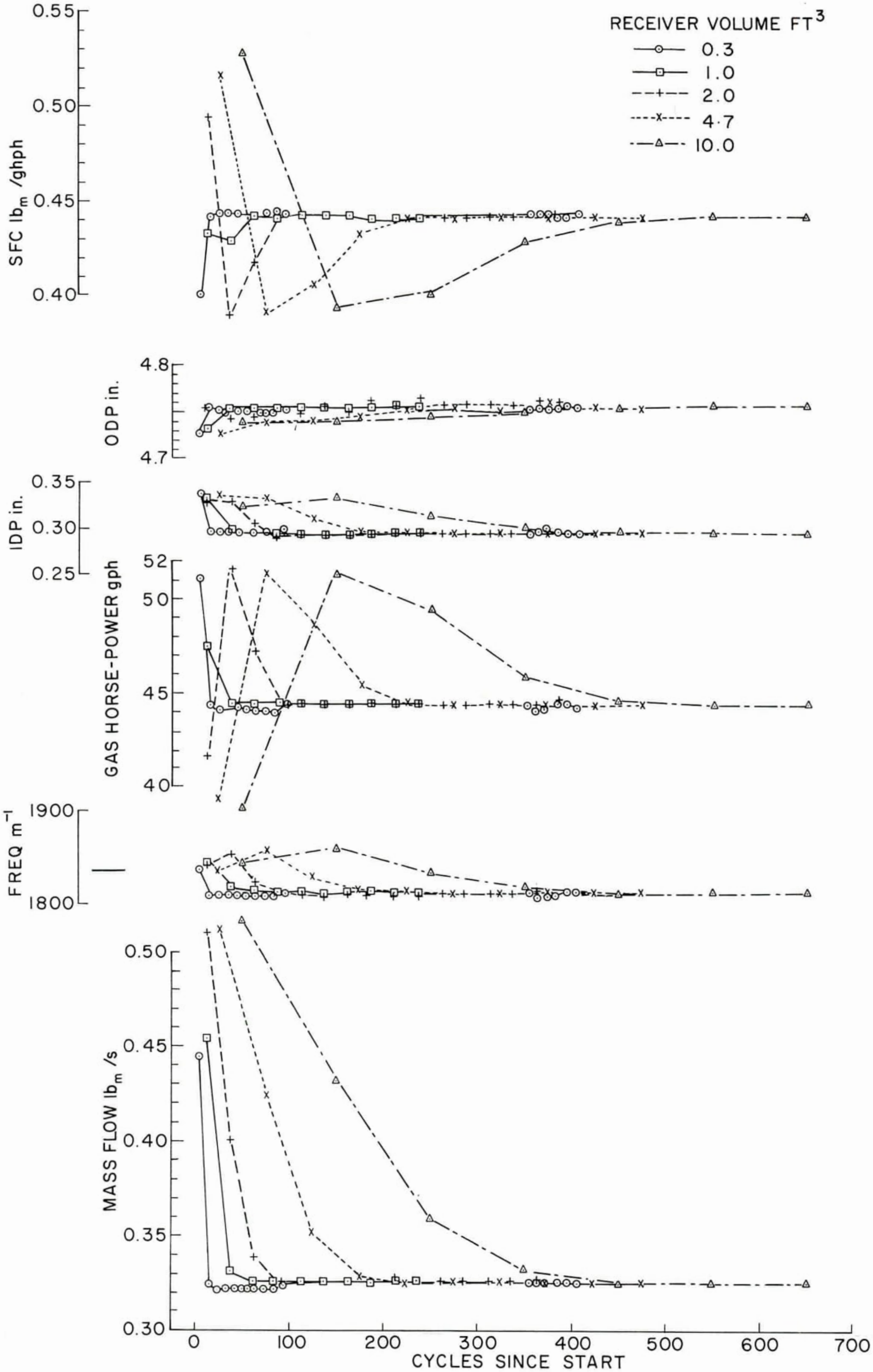

FIG.21: STABILITY

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FREE PISTON GASIFIER STUDIES: FURTHER DEVELOPMENT AND USE OF THE HYBRID SIMULATION
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Previous work on the modelling of the free piston gasifier has been extended to include refined models of the combustion process and of engine friction. The model was used to study the sensitivity of the engine performance to environmental and design parameters.

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2. Analogue computers (Hybrid)
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