Paper offer # 95P-9

SAE paper no 950988

Title Sheet

COVER INFORMATION FOR SAE TECHNICAL PAPERS

In order to ensure that the correct title, author names, and company affiliations appear on the cover and title page of your paper, please provide the information requested below.

US

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

PAPER TITLE

(upper and lower case):

Modeling the Effect of Engine Assembly Mass
on Engine Friction and Vehicle Fuel Economy

AUTHORS

(upper and lower case)

1. Feng An

2. Frank Stodolsky

3. _____

4. _____

5. _____

6. _____

7. _____

8. _____

9. _____

10. _____

COMPANY

(upper and lower case)

1. University of California, Riverside

2. Argonne National Laboratory

3. _____

4. _____

5. _____

6. _____

7. _____

8. _____

9. _____

10. _____

The submitted manuscript has been authored
by a contractor of the U. S. Government
under contract No. W-31-109-ENG-38.
Accordingly, the U. S. Government retains a
nonexclusive, royalty-free license to publish
or reproduce the published form of this
contribution, or allow others to do so, for
U. S. Government purposes.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

ANL/ES/CP--85239

CONF-950256--10

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

ABSTRACT

In this paper, an analytical model is developed to estimate the impact of reducing engine assembly mass (the term engine assembly refers to the moving components of the engine system, including crankshafts, valve train, pistons, and connecting rods) on engine friction and vehicle fuel economy. The relative changes in frictional mean effective pressure and fuel economy are proportional to the relative change in assembly mass. These changes increase rapidly as engine speed increases. Based on the model, a 25% reduction in engine assembly mass results in a 2% fuel economy improvement for a typical mid-size passenger car over the EPA Urban and Highway Driving Cycles.

INTRODUCTION

Automakers are striving to reduce vehicle weight and improve engine efficiency by developing lightweight engine materials and improved lubricants. The trend toward smaller, higher-revving engines requires lighter materials and improved designs to reduce engine friction and improve response. Reductions in engine assembly mass could be achieved by using intermetallics, metal-matrix composites, and ceramics instead of steel, cast iron, and aluminum. (We use the term engine assembly mass to refer to the moving components of the engine system, including crankshafts, valve train, pistons, and connecting rods. The mass of a typical engine assembly is about one third of the total engine mass [1].) In addition, design approaches (like using hollow crankshafts) could also reduce assembly mass. While improvements in lightweight materials give the engine designers an opportunity to reduce engine assembly mass, lightweight materials are often priced at a premium over conventional materials. Therefore, it is important to understand the correlation between fuel economy and engine assembly mass to help quantify the fuel economy benefit from lighter engine assembly materials.

In our search of recent literature, we could not find any discussion on the relationship between engine assembly mass and vehicle fuel economy. Papers by Lin and Patterson [2] and Uras and Patterson [3] contain the only direct discussion of the correlation between engine friction and assembly mass available in recent literature. In the past, most research in the area of engine assembly friction concentrated on improving measurement techniques and on how changes in engine design affect friction.

The main reason that few experiments have been done to correlate engine friction and engine assembly mass is that it is difficult to differentiate the mass reduction effect from other "side effects." Different piston materials not only change the piston mass, but also change characteristics of the entire engine system, such as distortion, lubrication, vibration, temperature, and thermal expansion. In addition, fuel consumption and friction changes as a result of mass reduction of the engine assembly is not as dominating as other factors.

In this paper, an analytical model is developed to describe the correlation between engine assembly mass, engine friction, and vehicle fuel economy. The reduction in engine assembly mass results in: (1) a reduction in total vehicle weight and moment of inertia, which in turn reduces the brake power requirement; (2) an engine downsizing assuming power-to-weight ratio (and hence vehicle acceleration) is held constant; and (3) a reduction in rubbing friction, which will reduce the engine friction factor. These three consequences all lead to a reduction in fuel consumption.

FUEL CONSUMPTION AND ENGINE FRICTION

An and Ross have established a model that gives the relationship between fuel consumption and engine friction based on the following relationship [4]:

MASTER

$$P_{fuel} = k V N + P_b / \eta \quad (1)$$

where P_{fuel} is the rate of fuel use expressed in energy terms, P_b is the brake power output, and k is the engine friction factor, which represents the fuel energy used to overcome engine friction per unit of engine revolution and engine displacement [$\text{kJ}/(\text{rev} \cdot \text{liter})$]. N is engine speed in revolutions per second, V is engine displacement, and η is a measure of indicated efficiency.

The engine frictional work is determined by pumping work and the work to overcome the rubbing friction and operate accessories. The total frictional energy is approximately proportional to the summation of engine friction mean effective pressure (f_{mep}), pumping mean effective pressure (p_{mep}), and an accessory term (k_{acce}). That is:

$$\frac{\delta k}{k} = \alpha_{rub} \frac{\delta f_{mep}}{f_{mep}} + \alpha_{pump} \frac{\delta p_{mep}}{p_{mep}} + \alpha_{acce} \frac{\delta k_{acce}}{k_{acce}} \quad (2)$$

where the contribution of rubbing frictional loss to total frictional loss (α_{rub}) usually ranges from 40 to 90%, depending on load and engine speed (see note a). A general assumption is that the accessory loss always contributes 10% to the total loss. In this paper, we are only interested in the rubbing friction term. We discuss how to estimate rubbing losses in the following sections.

ENGINE FRICTION AND MASS

On the basis of motored engine tests, where the engine is disassembled or broken down in stages, an approximate breakdown of rubbing friction is piston assembly, 60%; valve train, 25%; and crankshaft bearings, 15% [1]. Their relative importance, however, varies over the speed range. We assume this relationship is the same when the engine is firing. The major impacts on friction of reducing piston assembly mass are that (1) the inertial forces are lower and (2) the side thrust is reduced. The latter only affects piston assembly friction, but the inertial force influences all three items and has a much larger impact.

We assume the total engine friction mean effective pressure ($t_{f_{mep}}$) varies with speed according to Equation 13.5 of Heywood [1]:

$$t_{f_{mep}} = C_1 + C_2 N + C_3 N^2 \quad (3)$$

This semi-empirical equation is very useful. In this paper, we will verify the above equation based on simple physical laws, and show that the N^2 term is predominantly a direct

contribution from inertial force. The coefficients in the above equation will be determined accordingly.

PISTON-RING ASSEMBLY FRICTION - Bishop has developed correlations for piston and ring friction for boundary (mixed-lubrication) and viscous (hydrodynamic) friction conditions [5]. He found that under boundary conditions, the frictional mean effective pressure of the piston-ring assembly (f_{mep_p}) is proportional to normal load (W_{pr}), while in the viscous case, f_{mep_p} is simply proportional to engine speed. In actual engines, a combination of boundary and viscous friction occurs. Thus:

$$f_{mep_p} = \alpha' (f_{mep_p})_{boundary} + (1 - \alpha') (f_{mep_p})_{hydro} \quad (4)$$

and

$$f_{mep_p} \sim b_1 N + b_2 W_{pr} \quad (5)$$

This is a simplified but classical expression of the relationship between f_{mep_p} and N and mass. A more sophisticated formula is presented by Lin and Patterson [2] (see note b).

W_{pr} can be expressed as:

$$W_{pr} = W_p' + M_s a_1, \quad a_1 = a_p + g_c \quad (6)$$

where M_s is the assembly mass, and a_p and g_c are the piston and gravity acceleration-rate vectors, respectively. The term $M_s g_c$ represents the side thrust effect due to gravity force and the term $M_s a_p$ represents the inertia force. Evidently, g_c (about 10 m/s^2) is much smaller than a_p ($> 100 \text{ m/s}^2$). Thus, the side thrust effect is much smaller and we consider it to be negligible. W_p' is the nonmass-related normal load from ring tension, cylinder pressure, and thermal expansion.

One fact that needs to be stressed is that all rotational accelerations are proportional to the square of rotational speed. Thus the piston acceleration rate (a_p) is proportional to the square of engine speed.

Thus:

$$f_{mep_p} \sim b_3 + b_4 N + b_5 M_s N^2 \quad (7)$$

Based on Equation 7, we obtain:

$$\frac{\delta f_{mep_p}}{f_{mep_p}} = \left(\frac{\delta M_s}{M_s} \right) \cdot \gamma_{Ms} \quad (8)$$

$$\gamma_{Ms} = \frac{b_7 (N/1000)^2}{1 + b_6 (N/1000) + b_7 (N/1000)^2}$$

Here, γ_{Ms} is the fraction of normal load from the engine assembly mass, and b_6 and b_7 are adjustable coefficients.

For demonstration purposes, the coefficients are calibrated by using the engine and measurement data from Lin and Patterson [2]. They measured f_{mep} vs. piston-ring assembly mass for a firing engine under part-load (see Appendix for engine design parameters). They showed that a 25% increase in mass increased the assembly f_{mep} by 0.4% at 1000 rpm and 2% at 2000 rpm. Using this information, we obtain $b_6 = -0.2$ and $b_7 = 0.013$. Table 1 shows how γ_{Ms} varies as N , based on these data and on Equation 8.

For comparison, using Lin and Patterson's formula [2] we obtain:

$$\gamma_{Ms} = 0.55 \cdot 10^{-2} \cdot \left(\frac{N}{1000} \right)^2 \left(1.9 \sqrt{\frac{N}{1000}} + 1 \right) \quad (9)$$

Table 1 shows that Equation 9 gives very similar results, especially in the low-engine-speed range.

TABLE 1 Fraction of Normal Load from Engine Assembly Mass vs. Engine Speed

γ_{Ms} from	1000 rpm	2000 rpm	3000 rpm	4000 rpm
Eq. (8)	1.6%	8.0%	22.6%	51.0%
Eq. (9)	1.6%	8.0%	21.2%	42.4%

The conclusion derived from Table 1 is that Equation 7 gives a very good description of the correlation between piston assembly mass and friction mean effective pressure (f_{mep_p}). The N^2 term represents the contribution from the assembly mass. Table 1 also shows that the contribution from the mass term increases rapidly as engine speed increases. When $N = 1000$ rpm, the inertia term contributes less than 2% of the total friction. But when N is larger than 4000 rpm, the contribution from inertial force starts to dominate.

Table 2, generated by using Equation 8, shows that the assembly mass has much greater impact at higher engine speeds.

TABLE 2 Percentage Decreases of Piston-Ring f_{mep} due to Decreases of Assembly Mass at Different Engine Speeds

	1000 rpm	2000 rpm	3000 rpm	4000 rpm
10%	0.2	0.8	2.3	5.1
20%	0.3	1.6	4.1	7.8
30%	0.5	2.4	6.1	11.7
40%	0.6	3.2	8.1	15.6
50%	0.8	4.0	10.2	19.5

CRANKSHAFT BEARING FRICTION - Crankshaft friction contributions come from journal bearings (connecting rod, main and accessory or balance shaft bearings) and their associated seals. Equation 13.21 from Heywood shows that the friction in a hydrodynamically lubricated bearing yields a relation in the same form as Equation 5 [1]:

$$f_{mep_c} = c_1 N + c_2 W_{cb} \quad (10)$$

where W_{cb} can be expressed as $W_c + M_s a_c$. Here, a_c is the mean crankshaft acceleration rate (a rotational acceleration rate) and is also proportional to N^2 . W_c is the nonmass-related normal load. Thus:

$$f_{mep_c} = c_3 + c_4 N + c_5 M_s N^2 \quad (11)$$

VALVE TRAIN FRICTION - Loads acting on the valve train (W_v) mainly result from the spring forces (kS_v) and the inertial forces of the component masses (M_v):

$$W_v = k S_v + M_v a_v \quad (12)$$

where kS_v is the spring force and a_v is the mean acceleration rate of the valve train and is also proportional to N^2 . Valve train friction is very difficult to determine. Bishop developed a formula for valve train friction, showing that it decreases linearly with engine speed for a fixed load [5]. On the other hand, the valve train mean effective pressure f_{mep_v} also increases linearly with the normal load W_v :

$$f_{mep_v} = c_5 + c_6 N + c_7 M_v N^2 \quad (13)$$

Unlike the piston-ring assembly friction, there are no published measurement data for crankshaft bearing and valve train frictions. Thus, a new methodology is needed to solve this problem.

TOTAL ASSEMBLY FRICTION - Because piston assembly, valve train, and crankshaft bearing contributions are assumed to be 60%, 25%, and 15% of the total friction, respectively, it follows that:

$$\frac{\delta f_{mep}}{f_{mep}} = 0.6 \frac{\delta f_{mep_p}}{f_{mep_p}} + 0.25 \frac{\delta f_{mep_v}}{f_{mep_v}} + 0.15 \frac{\delta f_{mep_c}}{f_{mep_c}} \quad (14)$$

Combining Equations 7, 11, and 13 yields Equation 3. From previous discussions, it can be determined that all N^2 terms are directly derived from the inertial force, thus representing the contribution from the assembly mass. More detailed analyses show that the N^2 term is also related to air and fluid drags or turbulent dissipation because of the moving parts in the piston-ring assembly, valve train, and crankshaft. Here, it is assumed that this impact is very small and negligible.

Unfortunately, no good data exist to determine the coefficients of crankshaft bearing and valve train terms. Heywood presented a semi-empirical formula to describe the engine total motoring friction [1]:

$$tf_{mep}(\text{bar}) = 0.97 + 0.15 \left(\frac{N}{1000} \right) + 0.05 \left(\frac{N}{1000} \right)^2 \quad (15)$$

This equation is based on several four-stroke-cycle, four-cylinder spark ignition engines between 845 and 2000 cm^3 displacement, at wide-open throttle. Because the stated equation is based on motoring tests at wide-open throttle, it does not include much throttling pumping loss. It may include some loss from valve flow, but this contribution is small and negligible during motoring testing. Assuming that the above equation can be used to represent the rubbing friction under the firing case and the N^2 term is the predominant contribution from inertial force (assembly mass), it can be determined that the fraction of tf_{mep} from the assembly mass γ_T is:

$$\gamma_T = \frac{0.05 \left(\frac{N}{1000} \right)^2}{0.97 + 0.15 \left(\frac{N}{1000} \right) + 0.05 \left(\frac{N}{1000} \right)^2} \quad (16)$$

Table 3 shows how γ_T varies as N , based on the above equation.

TABLE 3 Fraction of Total Engine Friction from Assembly Mass vs. Engine Speed

	1000 rpm	2000 rpm	3000 rpm	4000 rpm
γ_T	4.3%	13.6%	24.1%	33.8%

On the basis of the following equation, Table 4 can be generated.

$$\frac{\delta tf_{mep}}{tf_{mep}} = \gamma_T \left(\frac{\delta M_s}{M_s} \right) \quad (17)$$

TABLE 4 Percentage Decreases of Total Frictional Mean Effective Pressure (tf_{mep}) due to Decreases of Assembly Mass at Different Engine Speeds

Assembly mass decrease	Engine Speed			
	1000 rpm	2000 rpm	3000 rpm	4000 rpm
10%	0.4	1.4	2.4	3.4
20%	0.9	2.7	4.8	6.8
30%	1.3	4.1	7.2	10.1
40%	1.7	5.4	9.6	13.5
50%	2.1	6.8	12.0	16.9

Note that the results in Table 2 and Table 4 are consistent, even though they come from totally different assumptions and sources. This supports the use of Equations 15 and 16 to describe the correlation between engine friction and assembly mass.

Tables 3 and 4 show that the frictional contribution from assembly mass at low engine speed is relatively small, only about 4% at 1000 rpm. But γ_T increases rapidly with engine speed, more than tripling at 2000 rpm, and roughly doubling again at 3000 rpm. When the engine speed is at or beyond 3000 rpm, the contribution becomes significant for both γ_T and tf_{mep} .

Because no measurement data are available to calibrate the coefficients in crankshaft bearing and valve train terms, their contributions can only be roughly estimated on the basis of the results presented in Tables 1 and 3. Thus, the

contribution of engine friction from these two terms, $\gamma_{cb,vt}$, can be derived as:

$$\gamma_{cb,vt} = \gamma_T - 60\% \cdot \gamma_{M_s} \quad (18)$$

Table 5 shows how $\gamma_{cb,vt}$ changes with engine speed.

TABLE 5 $\gamma_{cb,vt}$ and Engine Speeds

	1000 rpm	2000 rpm	3000 rpm	4000 rpm
$\gamma_{cb,vt}$	3.3%	8.8%	11.5%	8.4%
% of γ_T	77%	65%	48%	25%

The contribution from valve train and crankshaft bearing to total friction $tfmep$, $\gamma_{cb,vt}$, may increase with engine speed in the low engine-speed range and decrease in the high engine-speed range. It peaks around 3000 rpm. Table 5 also shows that $\gamma_{cb,vt}$ as a percentage of γ_T , is very large at low engine speed, nearly 60 to 80% of the total. As the engine speed increases, this contribution declines rapidly.

VEHICLE FUEL ECONOMY AND ASSEMBLY MASS

On the basis of Equation 1, the relative change in fuel economy is proportional to the relative change in the engine friction factor k . Thus based on Equation 2 we have:

$$\frac{\delta MPG}{MPG} = -\alpha_{mech} \frac{\delta k}{k} = -\alpha_{mech} \alpha_{rub} \frac{\delta tfmep}{tfmep} \quad (19)$$

where α_{mech} is the engine mechanical loss, including rubbing, pumping, and accessory loss, over total fuel energy use. α_{rub} is the contribution of rubbing friction to the total mechanical loss, which increases linearly as the engine percentage load or part-load p_{load} (see note a) [1].

In addition to its impact on engine friction factor k , the total assembly mass reduction reduces the overall vehicle weight as well. For the average mid-size passenger car, the engine contributes 10 to 15% of total vehicle weight. The assembly mass contributes about one-third of total engine mass. Here, we assume that the engine contributes 12.5% of the total vehicle weight. Thus an engine assembly mass contributes about 4.2% of the total vehicle weight.

An and Ross show that, without downsizing the engine, every 10% reduction in vehicle mass results in an improvement of about 2.7% in fuel economy for an average model-year 1989 mid-size car [4]. If the reduction

in assembly mass is combined with engine downsizing to maintain the power-to-weight ratio, the impact is much greater. An and Ross also show that every 10% reduction in engine displacement alone results in a 6.6% improvement in fuel economy. Thus, a 10% reduction in both vehicle mass and engine displacement results in a fuel economy improvement of as much as $2.7\% + 6.6\% = 9\%$. (Note that this estimate does not include the influence of decreasing the engine assembly mass.) Of course, this estimate varies for different vehicles and driving cycles. We use α_{v+M} to represent this quantity.

Note that this process is iterative because the reduction in engine displacement results in a further reduction in engine weight, which in turn results in further engine downsizing and fuel economy improvement. This cumulative factor can be determined in the following manner:

$$\sum_{n=1}^{\infty} (12.5\%)^n \cdot \frac{12.5\%}{1-12.5\%} = 14.3\% \quad (20)$$

Thus, the general relationship between fuel economy improvement and assembly mass reduction is:

$$\begin{aligned} \frac{\delta MPG}{MPG} = & -\alpha_{mech} \alpha_{rub} \frac{\delta tfmep}{tfmep} - \alpha_{v+M} \frac{14.3\%}{3} \frac{\delta M_s}{M_s} \\ = & -(\alpha_{mech} \alpha_{rub} \gamma_T + \alpha_{v+M} 4.8\%) \left(\frac{\delta M_s}{M_s} \right) \end{aligned} \quad (21)$$

Here, γ_T is defined by Equation 16.

Table 6 shows how the fuel economy of an average vehicle (for example, a 1993 Honda Accord with a 104 kW [140 hp] engine and automatic 4-speed transmission) responds to changes in assembly mass in the EPA Urban, Highway, and Composite Driving Cycles, according to this relationship.

The average engine speed (including engine idling speed) of the 1993 Accord is about 1700 rpm for the EPA Urban Driving Cycle and 2300 rpm for the EPA Highway Driving Cycle. The part-loads, which are calculated as the average torques required in the driving cycles over the maximum torque, are about 21% for urban cycle and 31% for highway cycle. The rubbing friction loss contributes about 55 to 60% of total friction loss for both driving cycles. The mechanical losses over total energy use are 53% for urban and 42% for highway. Thus, the overall improvement in fuel economy resulting from a 10% reduction in engine assembly mass is about 0.7% in both urban and highway driving. Note that about half of the improvement is from engine friction reduction and half is from engine

downsizing. If the engine assembly mass is reduced by 25%, the vehicle fuel economy is improved by about 1.8% for both driving cycles.

TABLE 6 Increase in MPG due to 10% Decrease in Assembly Mass in EPA Driving Cycles for 1993 Honda Accord

	Urban	Highway	Composite
N	1688 rpm	2273 rpm	
Part-load	20.6%	30.6%	25.1%
α_{rub}	56.2%	59.2%	57.6%
α_{mech}	52.6%	41.6%	47.7%
γ_T	10.9%	16.5%	13.4%
MPG impro. due to friction red.	0.32%	0.41%	0.36%
MPG impro. due to downsizing	0.40%	0.29%	0.35%
Total MPG impro.	0.72%	0.70%	0.71%

As a comparison, the impact on three other 1993 vehicles is shown in Table 7. These three vehicles are the 86 kW (115 hp) VW Golf, 104 kW (140 hp) Ford Taurus, and 66 kW (88 hp) Ford Escort. Each example has a 4-speed automatic transmission. Table 7 shows that fuel economy improvements range from 1.5 to 2.4% for these vehicles. The larger improvement for the VW Golf results from its higher average engine speeds in both driving cycles. For future smaller and higher-revving engines, the impact should be even greater.

TABLE 7 Percentage Increase of MPG due to 25% Decreases of Assembly Mass in EPA Driving Cycles

	Urban	Highway	Composite
Accord	1.8	1.8	1.8
Taurus	1.6	1.4	1.5
Escort	1.7	1.5	1.6
VW Golf	2.5	2.2	2.4

CONCLUSIONS

On the basis of the An-Ross model (Equation 1) and a semi-empirical formula (Equation 15), a correlation

between vehicle fuel economy, engine frictional mean effective pressure, and engine assembly mass is established. The relative changes in engine frictional mean effective pressure, friction factor, and fuel economy are proportional to the relative change in assembly mass, and they increase rapidly as engine speed increases. If a 25% reduction in engine assembly mass becomes possible in an average mid-size passenger car, about 2% improvement in fuel economy will be achieved. For future smaller and higher-revving engines, this improvement could be greater.

ACKNOWLEDGEMENT

This work was supported by the U.S. Department of Energy, Office of Transportation Technologies, Deputy Under Secretary for Energy Efficiency and Renewable Energy, under contract W-31-109-Eng-38. The authors thank Jim Eberhardt, Director, Office of Transportation Materials at the Department of Energy for support and guidance. The authors would also like to thank Marc Ross and Don Patterson of the University of Michigan for their guidance. We would also like to thank Dan Santini, Bill Marr, and Roy Cuenca of Argonne National Laboratory for their advice and comments on the earlier draft. We also thank Mary Harris and Amit Bando of RCG/Hagler Bailly for their support of this work. The authors are solely responsible for the contents of this paper.

REFERENCES

1. Heywood, J.B., 1988, *Internal Combustion Engine Fundamentals*, McGraw-Hill, Inc., New York.
2. Lin, S.-S., and D.J. Patterson, 1993, *Piston-Ring Assembly Friction Modeling by Similarity Analysis*, SAE paper no. 93079, Society of Automotive Engineers, Inc., Warrendale, Pa.
3. Uras, H.M., and D.J. Patterson, 1987, *Effect of Some Piston Variables on Piston and Ring Assembly Friction*, SAE paper no. 870088, Society of Automotive Engineers, Inc., Warrendale, Pa.
4. An, F., and M. Ross, 1993, *A Model of Fuel Economy and Driving Patterns*, SAE paper no. 930328, Society of Automotive Engineers, Inc., Warrendale, Pa.
5. Bishop, I.N., 1965, "Effect of Design Variables on Friction and Economy," SAE Transactions, Vol. 73, pp. 334-358, published by Society of Automotive Engineers, Inc., Warrendale, Pa.

APPENDIX

Engine Design Parameters in Lin and Patterson's Paper (ref. 2):

assembly mass:	709.5 g
bore:	80 mm
ring tensions:	24 N, 25 N, 102 N
stroke:	84 mm
ring width:	3.6 mm, 3.6 mm, 1.975 mm
connecting rod length:	134 mm
($C_{sk}-C_{cyl}$):	$12 \cdot 10^{-6}$ m/m°C
piston pin off-set:	1.525 mm
K_s :	$8.7559 \cdot 10^6$ N/m
skirt width, length:	39.37 mm, 57.15 mm

NOTES

- ^a From Figure 13-6 of Heywood [1], an empirical relationship between α_{rub} and p_{load} is established as:

$$\alpha_{rub} = 0.6 + 0.3 \cdot p_{load}$$

- ^b Lin and Patterson [2] created a more sophisticated semi-empirical engine piston-ring assembly friction model based on the concept of the Stribeck diagram and similarity analysis. The piston-ring assembly friction mean effective pressures (f_{mep_p}) were calculated from the estimated friction coefficient (F_c) of the assembly as:

$$F_f(\theta) = F_c(\theta) \cdot W_{pr}(\theta)$$

$$f_{mep_p} = \frac{1}{V_d} \int_0^{4\pi} F_f(\theta) \cdot dS(\theta)$$

Assuming that the assembly was generally operating back and forth between mixed-lubrication regions, they derived that the coefficient of friction F_c is a linear function of the square root of the duty parameter, which is proportional to the engine speed N over normal load W_{pr} :

$$f_{mep_p} \propto (1 + \alpha_1 \sqrt{\frac{N}{W_{pr}}}) \cdot W_{pr}$$