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Connecting Rod Forces In the Horizontal Engine

VICTOR W. BOLIE¹

Abstract. Four equations define the effects of gravity, linkage and piston geometry, engine speed, inertia and mass of the rod and piston, thickness and viscosity of the oil film, and pressure in the crankcase and the combustion chamber. The equations were programmed for automatic digital computation, with typical values for the 15 parameters. The results are illustrated by graphs of the piston-head pressure and the tensile and normal forces acting on each end of the rod as functions of crankshaft rotation.

In the optimum design of bearings it is essential to determine accurately the forces acting on the bearing surfaces. In the case of the connecting rod in the reciprocating engine the bearing forces arise from a combination of gravity, linear acceleration, rotational acceleration, and piston-head pressure. These forces are related to the engine speed and crank geometry by equations which are ordinarily considered too complex to yield efficient results. However, the advent of the modern high-speed automatic digital computer² indicates that a reappraisal of the problems within the scope of the design engineer is in order.

The purpose of this paper is to present some interesting results of evaluating numerically the complete set of connecting-rod force equations with the aid of the automatic digital computer. It is assumed that the connecting rod is rigid, the engine speed is constant, and the axes of the crankshaft and piston are both in the horizontal plane. The results are plotted graphically to illustrate the variation of the forces with crankshaft rotation at selected engine speeds.

THE FORCE EQUATIONS

The equations for the tensile and normal forces acting at each end of the connecting rod must take into account the geometry of the rod linkage, the rotational speed of the engine, the mass and moment-of-inertia of the rod, the mass and viscous drag of the attached piston, and the instantaneous pressure acting on the face of the piston. The pertinent geometry and forces are illustrated schematically in Figure 1. The crank arm of length "a" rotates through the angle θ , causing the rod to rotate through the angle ϕ . The rod has the mass m and the length b , and has the moment-of-inertia I about its center of mass, which is located by the distance kb from the crank end. The

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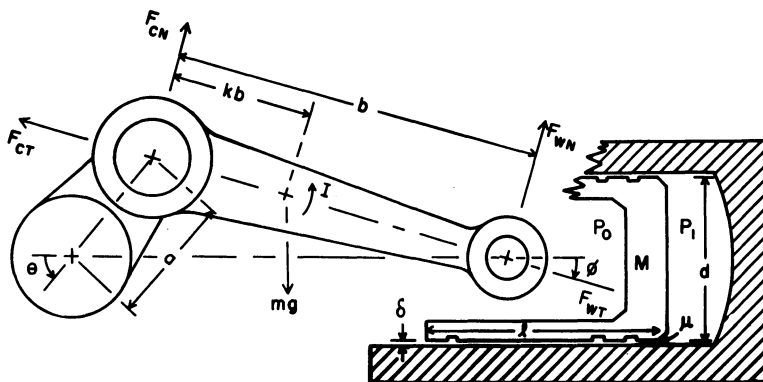


Figure 1. Connecting rod geometry and forces.

piston, a mass M , diameter d , and length l , is separated from the cylinder wall by an oil film of thickness δ and viscosity μ , and is acted upon by the combustion-chamber pressure p_1 as well as the crankcase pressure p_0 . The forces acting on the rod are the tensile and normal forces F_{CT} and F_{CN} acting on the crank end, and the tensile and normal forces F_{WT} and F_{WN} acting on the wrist pin end, as shown.

By resolution of the connecting-rod forces parallel and perpendicular to the piston axis, it is possible to obtain four simultaneous equations relating the four unknown forces F_{CT} , F_{CN} , F_{WT} , and F_{WN} . The first may deal with acceleration of the rod parallel to the piston axis, the second with acceleration of the rod normal to the piston axis, the third with the angular acceleration associated with rod rotation, and the fourth with the wrist pin force exerted on the rod by the piston. Certain characteristics of the matrix of coefficients in the four simultaneous equations permit a straightforward inversion without resorting to Cramer's rule. The resulting solutions for the desired force equations are found to be

$$\begin{aligned}
 F_{WT} &= -G_1 k \sin^2 \phi \sec \phi \\
 &\quad - G_2 k \sin \phi + G_3 \tan \phi + G_4 \sec \phi \\
 F_{CT} &= -G_1 [k \sec \phi + (1-k) \cos \phi] \\
 &\quad + G_2 (1-k) \sin \phi + G_3 \tan \phi + G_4 \sec \phi \\
 F_{WN} &= G_1 k \sin \phi + G_2 k \cos \phi - G_3 \\
 F_{CN} &= G_1 (1-k) \sin \phi \\
 &\quad + G_2 (1-k) \cos \phi + G_3
 \end{aligned}$$

where

$$\begin{aligned}
 \phi &= \sin^{-1} [(a/b) \sin \theta] \\
 G_1 &= m k \ddot{x} - m a(1-k) \dot{\theta}^2 \cos \theta
 \end{aligned}$$

$$G_2 = m g - m a(1 - k) \dot{\theta}^2 \sin \theta$$

$$G_3 = (I/b) \ddot{\phi}$$

$$G_4 = - M \ddot{x} - (\pi d l \mu / \delta) \dot{x} - (\pi d^2 / 4) (p_1 - p_0)$$

with

$$\dot{x} = - \left[1 + \frac{q \cos \theta}{\sqrt{1 - (q \sin \theta)^2}} \right] a \dot{\theta} \sin \theta$$

$$\ddot{x} = - a \dot{\theta}^2 \left[\cos \theta + \frac{q \cos^2 \theta}{(1 - q^2 \sin^2 \theta)^{1/2}} - \frac{q(1 - q^2) \sin^2 \theta}{(1 - q^2 \sin^2 \theta)^{3/2}} \right]$$

$$\ddot{\phi} = - \frac{q(1 - q^2) \dot{\theta}^2 \sin \theta}{[1 - q^2 \sin^2 \theta]^{3/2}}$$

where $\dot{\theta}$ is the angular velocity of the crankshaft, and $q = a/b$. These equations can readily be programmed for automatic calculation, using any desired engine speed and any specified functional variation of combustion chamber pressure with piston travel.

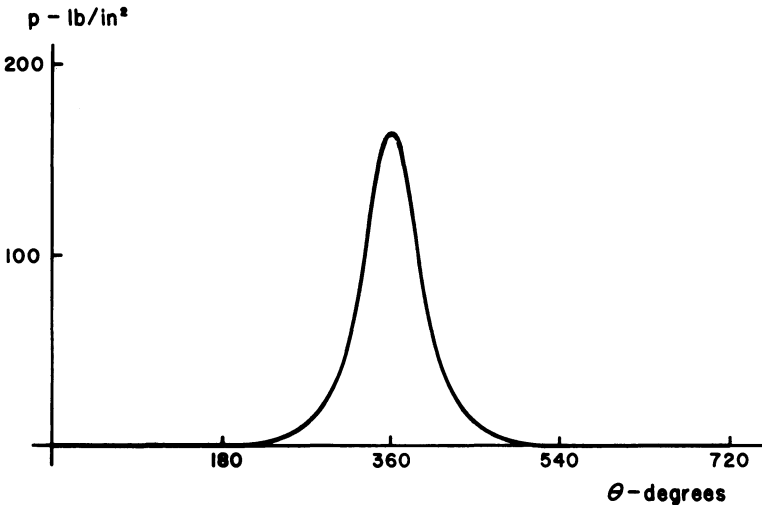


Figure 2. Gauge pressure vs. crank angle.

THE COMPUTED RESULTS

Data for defining the variation of combustion chamber pressure with crankshaft rotation can be obtained ideally from an indicator-card experiment. However, for purposes of illustration here, it is sufficient to assume the pressure follows a simple adiabatic law for a perfect gas without combustion. Thus, if p_{in} and p_{ex} represent the intake and exhaust pressures, and y denotes the piston travel

from top-dead-center, the following equations define the differential pressure $p = p_1 - p_0$ in terms of the crankshaft rotation θ , the compression ratio r , and the ratio γ of the specific heats for the gas medium.

$$\begin{aligned}
 y &= a + b - a \cos \theta - b \cos [\sin^{-1} (q \sin \theta)] \\
 p &= p_{in} - p_0 && \text{for } 0^\circ \leq \theta \leq 180^\circ \\
 &= p_{in} \left[\frac{r}{1 + \frac{r-1}{2a} y} \right]^\gamma - p_0 && \text{for } 180^\circ < \theta < 540^\circ \\
 &= p_{ex} - p_0 && \text{for } 540^\circ \leq \theta \leq 720^\circ
 \end{aligned}$$

A plot of the differential pressure p as a function of the crank angle θ , computed from the above equations with the typical values of $a = 3.375$ inches, $b = 13.0$ inches, $p_0 = p_{in} = p_{ex} = 14.7$ lb/in², $r = 6.0$, and $\gamma = 1.4$, is shown in Figure 2.

For illustrative purposes the connecting-rod force equations were numerically evaluated with an automatic digital computer, using the following typical values for the parameters in the computing program.

- | | |
|---------------------------------|------------------------------------|
| $a = 3.375$ inches | $\delta = 0.003$ inch |
| $b = 13.0$ inches | $\mu = 27$ centipoise |
| $m = 8.55$ lb | $r = 6.0$ |
| $g = 32.16$ ft/sec ² | $p_0 = 14.7$ lb/in ² |
| $I = 65.0$ lb ft ² | $p_{in} = 14.7$ lb/in ² |
| $l = 7.75$ inches | $p_{ex} = 14.7$ lb/in ² |
| $d = 5.5$ inches | $\gamma = 1.4$ |
| $M = 17.05$ lb | |

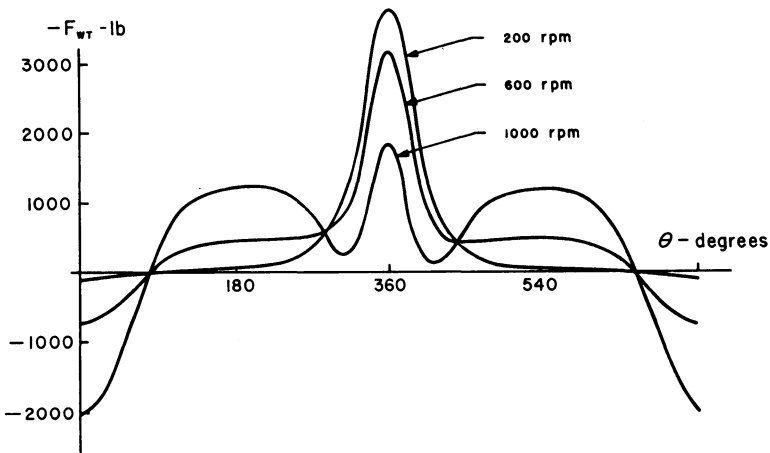


Figure 3. Tensile force at wrist pin.

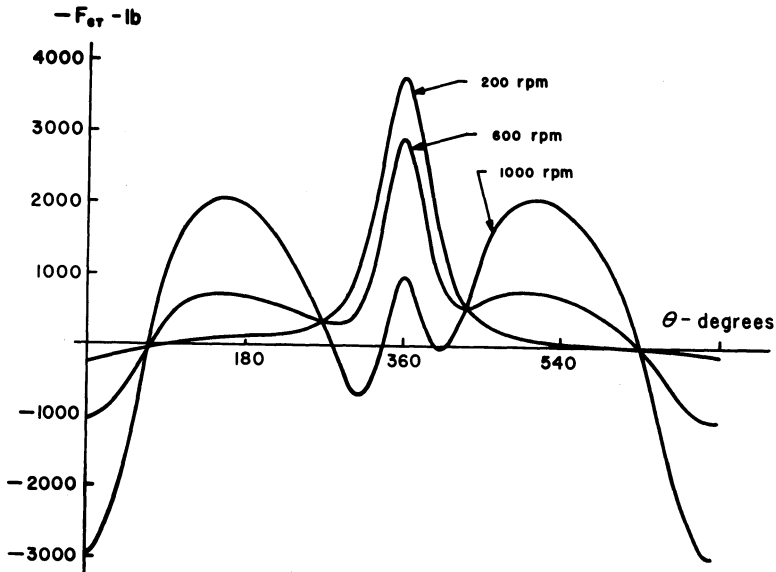


Figure 4. Tensile force at crank pin.

The computer program was designed to evaluate the four forces F_{WT} , F_{CT} , F_{WN} , and F_{CN} as functions of the crank angle θ for the three different engine speeds of 200 rpm, 600 rpm, and 1200 rpm. The results of these calculations are plotted in Figures 3, 4, 5, and 6. In Figure 3, the 200 rpm curve for the tensile force F_{WT} at the wrist-pin is seen to resemble the waveform of the differential piston pressure. However, at higher engine speeds the resemblance is lost on account of the increased force required to decelerate the piston in its travel-limit regions. In Figure 4, the curves for the tensile

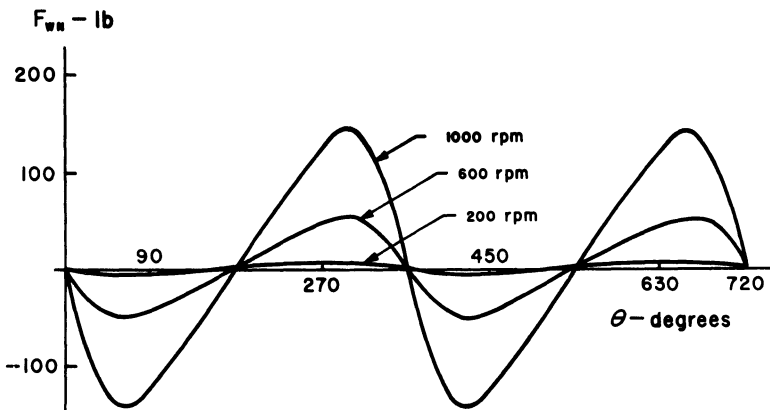


Figure 5. Normal force at wrist pin.

force F_{CT} at the crank-pin are similar in form to those in Figure 3, the differences being attributable to the extra force required to decelerate the connecting rod in its travel-limit regions. In Figure 5, the curves for the normal force F_{WN} at the wrist-pin are found to be dominated by the force required to accelerate the connecting rod in a direction normal to the piston axis. In Figure 6, the curves for the normal force F_{CN} at the crank-pin are similar in form to those in Figure 5, the differences being attributable to the fact that the center of mass of the connecting rod is closer to the crank end.

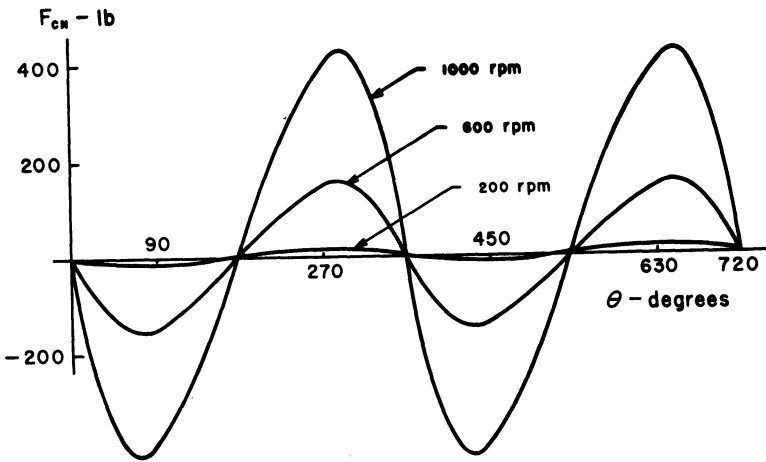


Figure 6. Normal force at crank pin.

CONCLUSIONS

The equations representing the tensile and normal forces acting on each end of the connecting rod are expressed in forms which allow considerable flexibility in application. For example, the effects of viscous drag may be observed by repeating the calculations with increased values of μ . More precise results for the case of combustible gases can be obtained simply by substituting the pressure wave-form from an indicator card experiment. The lubricant-surface non-uniformity introduced by piston rings can be compensated at least in part through the substitution of an appropriate "effective length" of the piston. By transferring the mg term from the expression for G_2 to the expression for G_1 , the equations may also be used for engines having the piston axis and crankshaft centerline both in the vertical plane. Explorations for minimizing the bearing loads with optimum geometry could easily be made by incorporating the force equations into an appropriate digital computer program. Extensions of this work could also accommodate non-rigid connecting rods and torsional vibrations in the crankshaft.