Table 1 Unbalanced mechanical shaking forces and moments

	Forces, lb			
	Primary		Secondary	
	Vertical	Horizontal	Vertical	Horizontal
Case 1	4470	800	0	2840
Case 2 articulated	0	0	0	0
Case 2 nonarticulated	0	0	0	0

	Moments, ft-lb			
	Primary		Secondary	
	Vertical	Horizontal	Vertical	Horizontal
Case 1	132,170	12,260	28,710	102,040
Case 2 articulated	108,040	108,040	74,670	9470
Case 2 nonarticulated	46,820	46,820	122,290	9040

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DISCUSSION

J. W. Holmes[1]

The authors are to be complimented on an analysis that treats the articulated rod mechanism with complete generality. The paper represents an original contribution, inasmuch as work done in this area in the early days of the multibanked internal combustion engine usually involved one or more of these simplifying assumptions:

1 Cylinder centerlines were not offset.

2 The articulating pin angle to the master rod centerline was equal to the bank angle (common in aircraft radial engines).

3 The articulating pin was so located that it lay on the line through the crankpin and wristpin at outer dead center of the slave cylinder.

The paper is very well organized; however, one wonders if it would not have been somewhat less forbidding in Figs. 1 and 2 and in the analysis if two deletions had been made. As the authors state, the compression rod is merely a special case of the master power rod and therefore does not merit separate analysis. Since angular acceleration of the crankthrow is neglected, the main bearing reactions arrived at are simply the combination of the crankpin applied forces and the centrifugal force of the crankthrow. It is felt that omitting the analysis of both the compressor rod and the crankthrow would have focused attention more sharply on the analysis of the rod mechanism, where the paper makes its most significant contribution.

The paper points out that the harmonics of the torque at the individual crank throws are useful for determining the excitation of torsional vibration. The harmonics of bearing reactions at each throw are also valuable for evaluating the internal bending moments in the engine frame. In a unit whose mounting gives a stiff coupling to earth and whose frame has a typical degree of flexibility, these moments are apt to be of greater concern than are the overall shaking forces and couples.

It is hoped that in further extensions of this effort the authors will consider the development of a direct formulation of the harmonics of force and torque at the individual crankthrow as has been done for the more restricted analyses mentioned earlier. Not only are such expressions convenient for many computations, but they also give better insight into the significance of the many parameters involved.

Again, the authors are to be praised for an ambitious and stimulating treatment, which encompasses the total scope of variation in the articulated rod mechanism.

[1]Mount Vernon, Ohio.

Authors' Closure

The authors wish to thank Mr. Holmes for his interesting discussion. He has brought additional historical insight into this problem, particularly in regard to the assumptions made in previous analyses. He has also pointed out the further application of this type of analysis to the engine frame forces.

In regard to the bearing force analysis, the angular acceleration of the crank throw has been neglected in the present computer implementation. The angular acceleration is clearly included in the analysis (equation (57)), and can be readily included if desired.

In the matter of a direct harmonic order analysis, the authors believe that the present approach is preferable. As given in the paper, the instantaneous forces are calculated. From this it is a relatively simple process to search out the maxima, and these are often quite important. If the anaylsis is approached in terms of individual harmonic orders, the entire series must be summed to obtain maxima. If harmonic components are required from the instantaneous force calculation, only those components that are required need to be computed.