REPORT No. 850

A VISUAL AND PHOTOGRAPHIC STUDY OF CYLINDER LUBRICATION

By Milton C. Shaw and Theodore Nussdorfer

SUMMARY

A V-type engine provided with a glass cylinder was used to study visually the lubrication characteristics of an aircraft-type piston. Photographs and data were obtained with the engine motored at engine speeds up to 1000 rpm and constant cylinder-head pressures of 0 and 50 pounds per square inch.

A study was made of the orientation of the piston under various operating conditions, which indicated that the piston was inclined with the crown nearest the major-thrust cylinder face throughout the greater part of the cycle. The piston moved laterally in the cylinder under the influence of piston side thrust.

The data and the photographs indicate that the lubrication of the piston assembly, under the conditions of these experiments, is hydrodynamic in nature for at least part of the stroke. Definitely wedge-shaped oil films were photographed and correlated with piston-orientation data.

The amount of lubricant present on the piston skirt varied with the relative angular positions of piston rings and with the lateral motion of the piston because of load. Rate and direction of piston-ring rotation varied with cylinder-head pressure and engine speed. Piston rings were observed to rotate as rapidly as 1 rpm at an engine speed of 1000 rpm.

Little oil was present on the piston skirt when the cylinder was operated under load, and the oil film on the face of a piston ring was estimated to be 0.0001 inch thick or less. In general, much less oil was present on the upper side of the inclined piston than on the lower side. In this case, the upper side was the major-thrust face.

INTRODUCTION

The physics of cylinder lubrication has progressed less rapidly than that of journal-bearing lubrication. This lack of development may be attributed to the fact that cylinder lubrication is achieved under transient conditions, whereas nearly all journal-bearing investigations have been conducted under steady conditions of load, speed, and temperature. In contrast to the many theoretical papers concerning journal-bearing lubrication, only two hydrodynamic treatments of cylinder lubrication have been found in the literature (references 1 and 2); these papers treat the lubrication of piston rings only under equilibrium conditions of load, speed, and temperature.

Theoretical investigations of cylinder lubrication must be continuously supported by experimentally proved facts in order that reasonable simplifying assumptions may be made. Although many engine tests have been made in cylinder-lubrication studies, relatively few experiments have been performed in which pertinent data were collected under carefully controlled operating conditions. References 3 to 11 describe such controlled experiments.

A visual study of any problem is generally gratifying and often reveals information that is available by no other means. If a piston can be observed in operation and if the manner in which the oil and the piston move along the cylinder can be studied, much useful information may be obtained. Attempts to study journal and slider lubrication using glass bearings are reported in references 12 to 15 but no publication that describes a similar attempt to study cylinder lubrication seems to be available.

A visual study of the cylinder-lubrication process was conducted during 1944 at the NACA Cleveland laboratory with a specially designed engine equipped with a glass sleeve. The orientation of the piston and the extent and characteristics of the lubricating film were examined with the test engine motored over a limited range of operating variables. A description of the apparatus and the techniques with some preliminary results are presented herein.

APPARATUS

Engine.—A two-cylinder, V-type (90°) engine was used in this investigation. One of the cylinder blocks was replaced by a special unit that held the glass sleeve and allowed the piston to be observed without obstruction. The sleeve of pyrex glass was centrifugally cast to a diametrical tolerance of ±0.0015 inch and was 0.25 inch thick. The surface finish was smoother than any ground or lapped metal surface would be.

A large cylindrical pressure chamber having a volume of 2000 cubic inches was used in place of the standard head. The ratio of the volume of this chamber to the piston displacement (33.6 cu in.) was such that only a 1.7-percent variation in the pressure occurred within the chamber as the piston moved from one dead-center position to the other. Preliminary study of such a simple constant-pressure system was considered advisable in order that interpretation of the data would be straightforward. The pressure chamber was equipped with a pressure gage, a pressure-relief valve, a fitting for connection to a nitrogen cylinder, and an oil-collection tube. The engine is diagrammatically and photographically shown in figures 1 and 2, respectively. The principal engine dimensions are as follows:

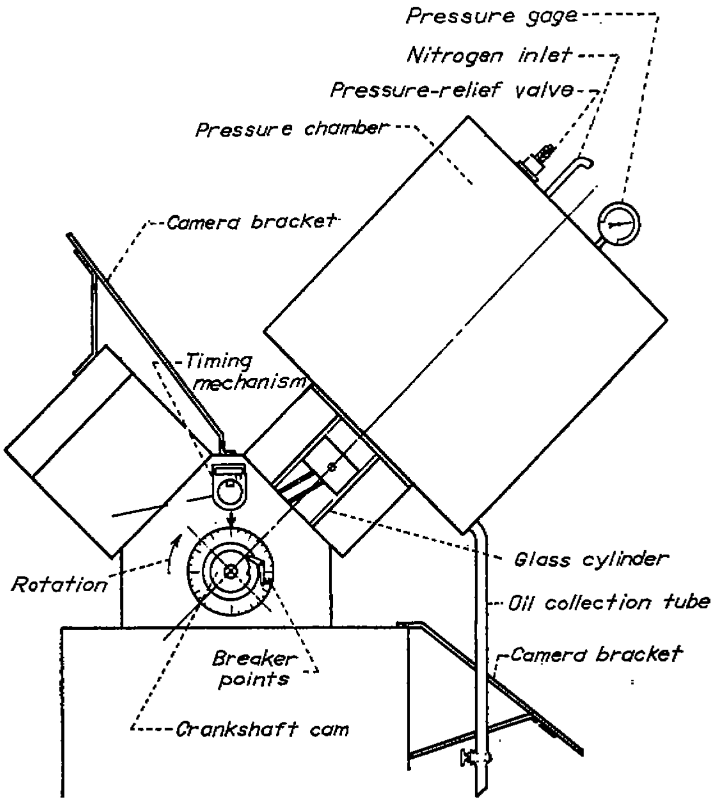
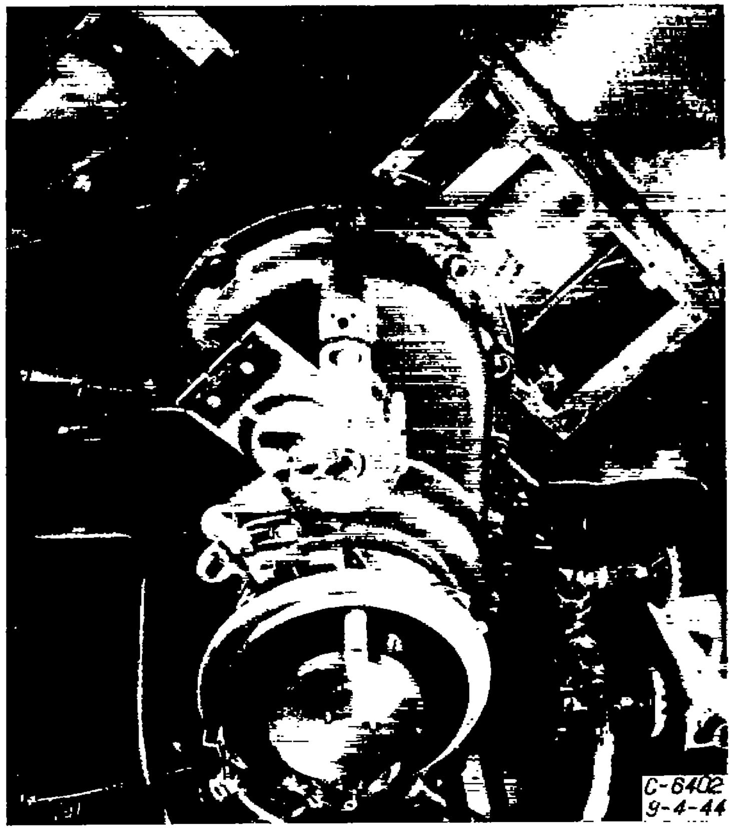
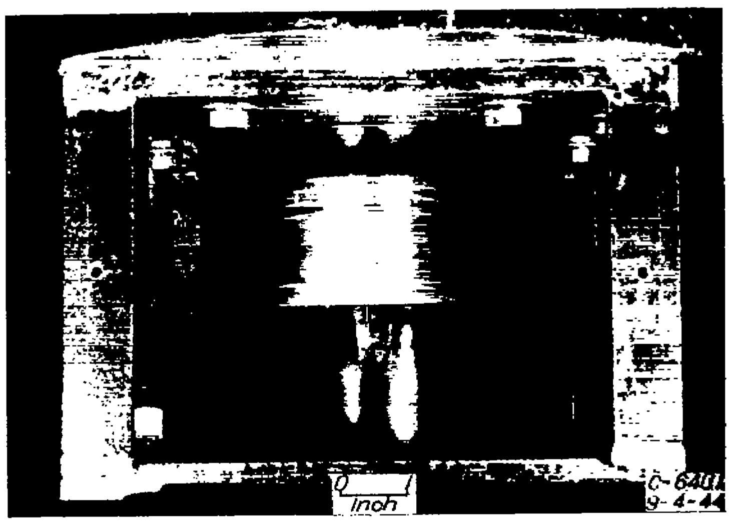


FIGURE 1.—Diagram of engine equipped with glass cylinder.



(a) View of apparatus.



(b) Close-up of glass cylinder and piston.
FIGURE 2.—Photograph of engine equipped with glass cylinder.

Bore, (in.)	3.125
Stroke, (in.)	4.375
Piston displacement, (cu in.)	33.6
Clearance volume, (cu in.)	2000
Diametral skirt clearance, (in.)	0.006
Diametral ring-belt clearance, (in.)	0.008
Weight of piston assembly, (lb)	0.946
Connecting rod:	
Length, (in.)	8.015
Distance from center of wrist pin to center of gravity, (in.)	5.61
Weight, (lb)	2.07
Moment of inertia about center of gravity, (lb) (ft) (sec^2)	0.00457
Ratio of connecting-rod length to crank throw	3.66

The pistons for this investigation were turned from aluminum castings to dimensions approximately proportional to those of a piston from an engine of 1710-cubic-inch displacement. The dimensions of both the model piston and the piston from the engine of 1710-cubic-inch displacement are given in figure 3.

The engine was driven by a 6-horsepower, direct-current motor; by means of a system of pulleys, any speed from 50 to 5000 rpm could be obtained. The glass sleeve was operated in still air with no special arrangement for cooling. The cylinder was lubricated by a cone of oil directed against the bottom of the sleeve by an auxiliary oil system. In all runs, the oil-inlet temperature was maintained approximately constant at 80° F and the pressure to the auxiliary nozzle

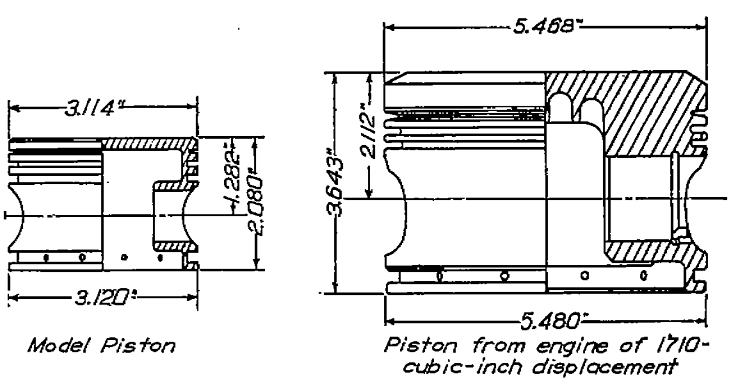
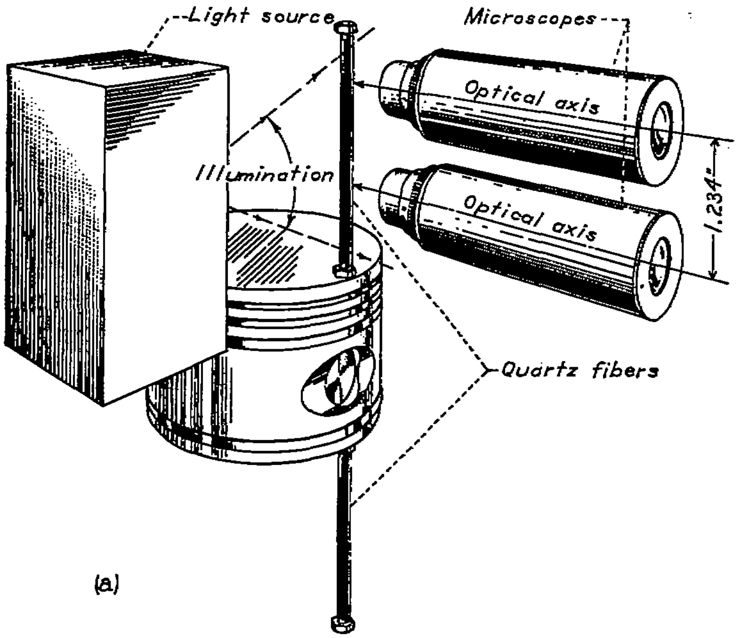
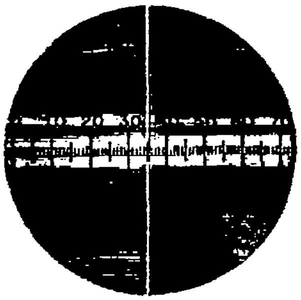


FIGURE 3.—Diagram of model piston and piston from an engine of 1710-cubic-inch displacement showing proportional dimensions.



(a) Arrangement of apparatus.



(b) Quartz fiber photographed through microscope. Small divisions represent 0.001 inch.

FIGURE 4.—Auxiliary apparatus for investigating piston orientation.

was maintained approximately constant at 40 pounds per square inch. The rate of oil flow to the bottom of the cylinder was 0.6 pound per minute; SAE 10 oil having a viscosity of 9.16×10^{-6} reyns (1 reyn is 1 (lb)(sec)/sq in.) at 80° F was used. The natural fluorescence of this oil was increased in this investigation by the addition of a fluorescent dye.

Apparatus for investigating piston orientation.—A knowledge of the variation of lateral position and inclination of the piston with crank angle was required to interpret photographs of the oil film between the piston and the cylinder wall. Quartz fibers 0.001 inch in diameter were mounted above and below the piston as shown in figure 4 (a) to provide a reference line on the piston. By means of a cathetometer, the central element of the major-thrust face of the cylinder was found to be parallel to the two co-linear fibers. Figure 4 (b), a photograph taken through one of the microscopes, shows that the fibers provide a distinct index. A very sharp high light was obtained on each edge of the fiber by side lighting, and the line along the left edge was used in all observations.

The relatively large distance of the reference line from the wrist pin magnified the linear displacement at the top or bottom of the piston because of a small change in its inclination. Unlike various index lines ruled directly upon the piston, the quartz fibers remained very distinct when viewed by stroboscopic light at engine speeds as high as 1500 rpm.

Two measuring microscopes with scales divided into 0.001-inch units were mounted one above the other on the cylinder block and were focused upon the quartz fibers (fig. 4 (a)). The microscopes were adjusted to give the same reading when moved to successive positions along a straight edge mounted parallel to the cylinder axis.

PHOTOGRAPHIC TECHNIQUES

The oil film between a piston and a cylinder is generally invisible when directly illuminated. A relatively thick oil film, such as obtains in a journal bearing under light load, may be rendered visible by means of a substance added to the oil. In lubrication studies with glass journal bearings, Barnard (reference 14) used a dye and Vogelpohl (reference 15) used graphite and sawdust. Because the oil films encountered in cylinder lubrication even under light loads are extremely thin, a concentration of dye or other material sufficient to make the extent of the film visible would be difficult, if not impossible, to obtain. In these runs two new methods of making very thin oil films readily visible were used.

Scattered-light method.—One method of rendering oil films visible utilizes a scattered-light technique. The relative positions of flash tubes, baffles, mask, and camera required in the scattered-light method are diagrammatically shown in figure 5. Four General Electric flash tubes

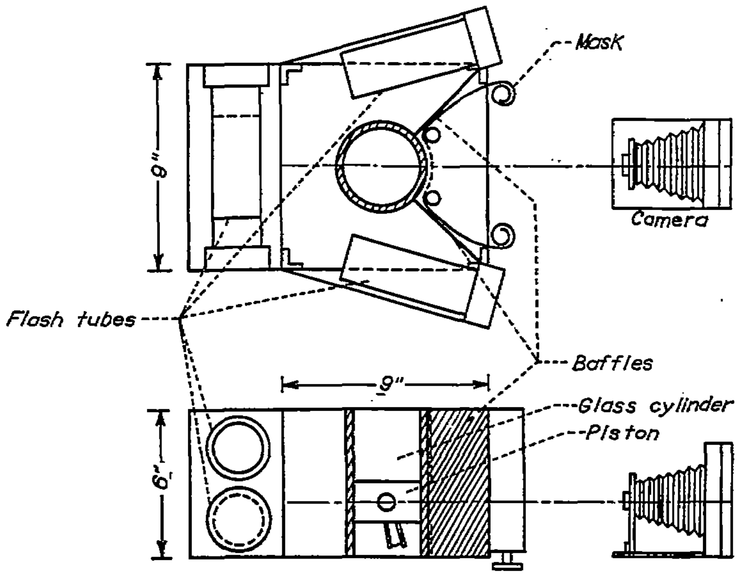


FIGURE 5.—Arrangement of apparatus for photographing cylinder lubrication by scattered-light method.

(FT–19 type) were operated stroboscopically. The circuit was "triggered" by a set of automotive breaker points that was operated by a cam on the crankshaft. (See figs. 1 and 2.) The mask was necessary to keep light from streaming directly across the cylinder at points above and below the piston and reducing contrast by fogging the film. The mask was arranged in the form of a roll with 36 openings, each slightly smaller than the size of the piston. Each opening was cut lower than the preceding one, providing a mask for use at any value of crank angle.

Light entered the space between the piston and the cylinder wall at a grazing angle, and the fluorescent dye in the oil caused the light to be scattered at points where an oil film was present. Some of the scattered light left the cylinder, which resulted in a sensation of light at all points where there was an oil film and one of darkness where no oil film existed. This phenomenon of light scattering is the basis of the ultramicroscope devised by Siedentopf and Zsigmondy in 1903.

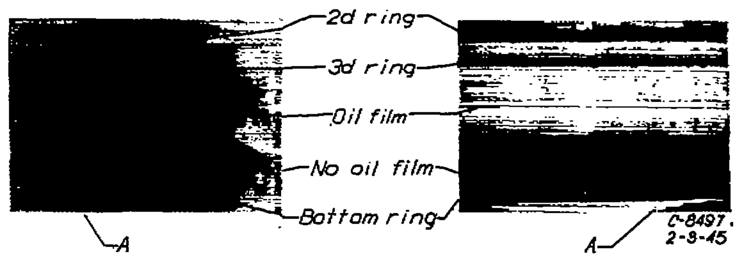
A few experiments were made in order to study the characteristics of the light scattered by a thin film of liquid between the piston and the cylinder wall. Nonfluorescent white mineral oil to which colloidal copper (nonfluorescent material) was added did not give as good results as an oil containing fluorescent dye even though the copper sol showed a strong Tyndall effect. Contrast between a complete oil film and a region having no oil film was greatly sacrificed when a light source with a relatively strong ultraviolet component was replaced by a light source with much more of its radiation in the visible region of the spectrum. Maximum contrast was obtained by the combined scattering of white light and the emitting of visible fluorescent light produced when scattered ultraviolet light was absorbed by the dye in the oil. No scattered light was observed in the absence of an oil film.

Satisfactory photographs were obtained from a single flash of the four lamps using a high-speed panchromatic film and a camera that had an f/1.9 lens. The scattered-light method was used to obtain single-flash photographs of the oil film under different combinations of engine speed and cylinder-head pressure. The oil film was reproduced with sufficient exactness from stroke to stroke so that a motion picture could be obtained from a sequence of single-flash photographs taken at successive values of crank angle. The slight discontinuity appearing from frame to frame in this type of motion picture indicates deviation in the extent of the oil film from stroke to stroke and is more enlightening than annoying.

Fluorescent-light method.—In the Fluorescent-light method, the glass cylinder was directly illuminated by long-wavelength ultraviolet radiation from which all the visible light had been filtered—a so-called beam of "black light." Inasmuch as all motor oils are fluorescent, a visible light was emitted from the oil film and only invisible utlraviolet rays were reflected from points at which there was no oil film. The intensity of the visible fluorescent light varied with film thickness and concentration of the fluorescent dye in the oil.

The fluorescent-light method rendered an oil film very distinct to the eye but the intensity of the fluorescent light was insufficient for obtainment of a photograph with a single stroboscopic flash of light. Good multiflash photographs were obtained by using a brilliant stroboscopic source of ultraviolet light and a camera with a fast lens. General Electric FT–19 flash tubes in conjunction with Hanovia Sc 5022 filters were the most satisfactory stroboscopic light source. Photographs were obtained with 25 flashes from two tubes using a camera equipped with an f/1.9 lens, a Wratten No. 2A filter, and a fast panchromatic film. The filter was used to prevent reflected ultraviolet light from entering the camera lens, an important consideration because of the extreme sensitivity of photographic film to ultraviolet rays of long wavelength.

Comparison of scattered-light and fluorescent-light methods.—Photographs of a stationary piston illustrating the difference in appearance of pictures taken by the scattered-light and the fluorescent-light methods are presented in figure 6. These photographs were taken in a succession



(a) Scattered-light photograph taken with single flash of two tubes.

(b) Fluorescent-light photograph taken with 40 flashes of two tubes.

FIGURE 6.—Photographs of stationary piston taken by scattered-light and fluorescent-light methods.

without disturbing the piston. The scattered-light photograph (fig. 6 (a)) was taken using a single flash from two tubes; the fluorescent-light photograph (fig. 6 (b)) was taken with 40 flashes from the two tubes. The scattered-light photograph, shows only the oil films that touch both the cylinder wall and the piston; whereas the fluorescent-light photograph shows the presence of all oil on the piston above a certain thickness whether or not the film touches the cylinder wall. For example, the oil film at point A (fig. 6) did not touch the cylinder wall; the fluorescent-light photograph shows the presence of oil, but the scattered-light photograph shows nothing at this point. Figure 6 also shows that the fluorescent-light method will not record the presence of as thin an oil film as the scattered-light method. (cf. the appearance of the ring faces in the two photographs.) The two methods are thus seen to be supplementary.

Although the multiflash-fluorescent method failed to give sharp photographs in all instances, it did indicate the average extent of the oil film during several successive strokes. Furthermore, the fluorescent method proved convenient for studies of the oil film by direct observation and could also be used to estimate the thickness of the oil film by comparing the intensity of the fluorescent light from the cylinder with that from a wedge of oil of known thickness.

RESULTS AND DISCUSSION

In this investigation, a crank angle of 0° corresponds to the top-center position of the piston. The major-thrust face is considered to be the one upon which the piston rides on the down stroke.

Piston orientation.—The results of the quartz-fiber investigations are given in figure 7 where the observed upper-microscope and lower-microscope readings are plotted against crank angle for three engine speeds at cylinder-head pressures of 0 and 50 pounds per square inch. Curves of total side thrust, velocity, and acceleration of the piston are given for each run for use in discussing the piston-orientation data. The method of computing the total side thrust of the piston is described in the appendix.

The vertical distance between the curves for the upper and the lower microscopes, at any particular crank angle, is a measure of the inclination of the piston. When the curve obtained with the upper microscope lies above that obtained with the lower microscope, the piston was inclined with its upper end toward the major-thrust face of the cylinder; when the reverse is true, it was inclined toward the minor-thrust face. The position of the mean of the two microscope readings relative to the 0.020-inch line determines whether the center of the piston was on the major- or the minor-thrust side of the cylinder center line. For example, when the engine was run at 50 rpm with a cylinder-head pressure of 50 pounds per square inch (fig. 7 (b)), the piston was inclined with its crown toward the major-thrust face of the cylinder and was also closer to the major-thrust face than the minor-thrust face at a crank angle of 100°.

The curves of figure 7 give the true inclination of the piston and a good approximation of its lateral position. These curves represent the loci of the lateral displacements of points on the piston having a vertical spacing of 1.234 inches (the distance between the microscopes). Because the positions of the microscopes relative to the top of the piston change continuously with crank angle, these curves give only an indication of the relative locations of fixed points on the piston. The exact location of the upper and lower ends of the piston could have been determined but this complicated refinement is unwarranted by the precision of the data.

Observations were made at an engine speed of 50 rpm in order to simulate operation under very high load at a high engine speed without putting too much stress on the glass cylinder. A total piston side thrust of 100 pounds at an engine speed of 50 rpm produces film thicknesses equivalent to a much higher load at a practical engine speed. The total piston side thrust reached a maximum at approximately the same crank angle in figures 7 (b), 7 (d), and 7 (f), but the lateral motion of the piston was considerably greater at 50 rpm than at the higher engine speeds. The piston ran approximately concentric with the cylinder when the resultant piston side thrust was small (figs. 7 (a), 7 (c), and 7 (e)), but considerable lateral motion was noted under load (figs. 7 (b), 7 (d), and 7 (f)).

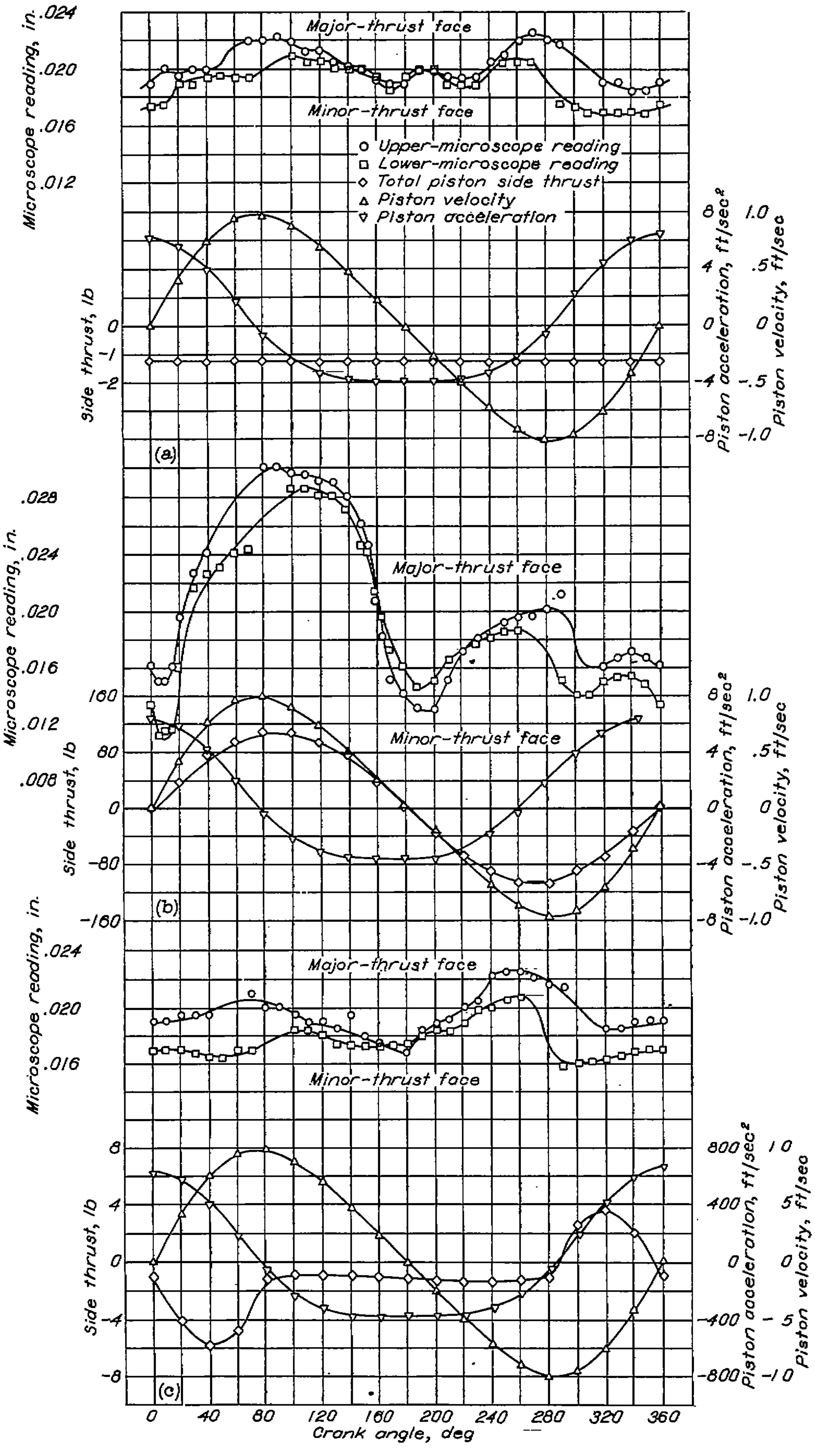
The piston was inclined with its upper end nearer the major-thrust face of the cylinder during the greater portion of both the up and down strokes. (See fig. 7.) Such an inclination is essential if the piston is to operate hydrodynamically as a curved slider bearing. The inclination was in all cases equal to or less than 16′ of arc and was generally much less.

In the region of bottom center an inverted piston slope was evident, particularly at low speed. As the piston reached bottom center (fig. 7 (b)), it moved toward the minor-thrust face of the cylinder, became vertical, and assumed a position with the top of the piston closer to the minor-thrust face of the cylinder than to the major-thrust face. As the velocity increased and an oil film was established, the piston returned to its normal inclination.

The piston did not move violently from one cylinder face to the other as the total piston side thrust changed direction. The lateral movement of the piston occupied a considerable portion of the stroke. The start of this movement at the top-center position lagged the reversal of piston side thrust, particularly at high engine speeds. Conversely, the movement at the bottom-center position anticipated the reversal of side thrust, particularly at low engine speeds. (See figs. 7 (b), 7 (d), and 7 (f).)

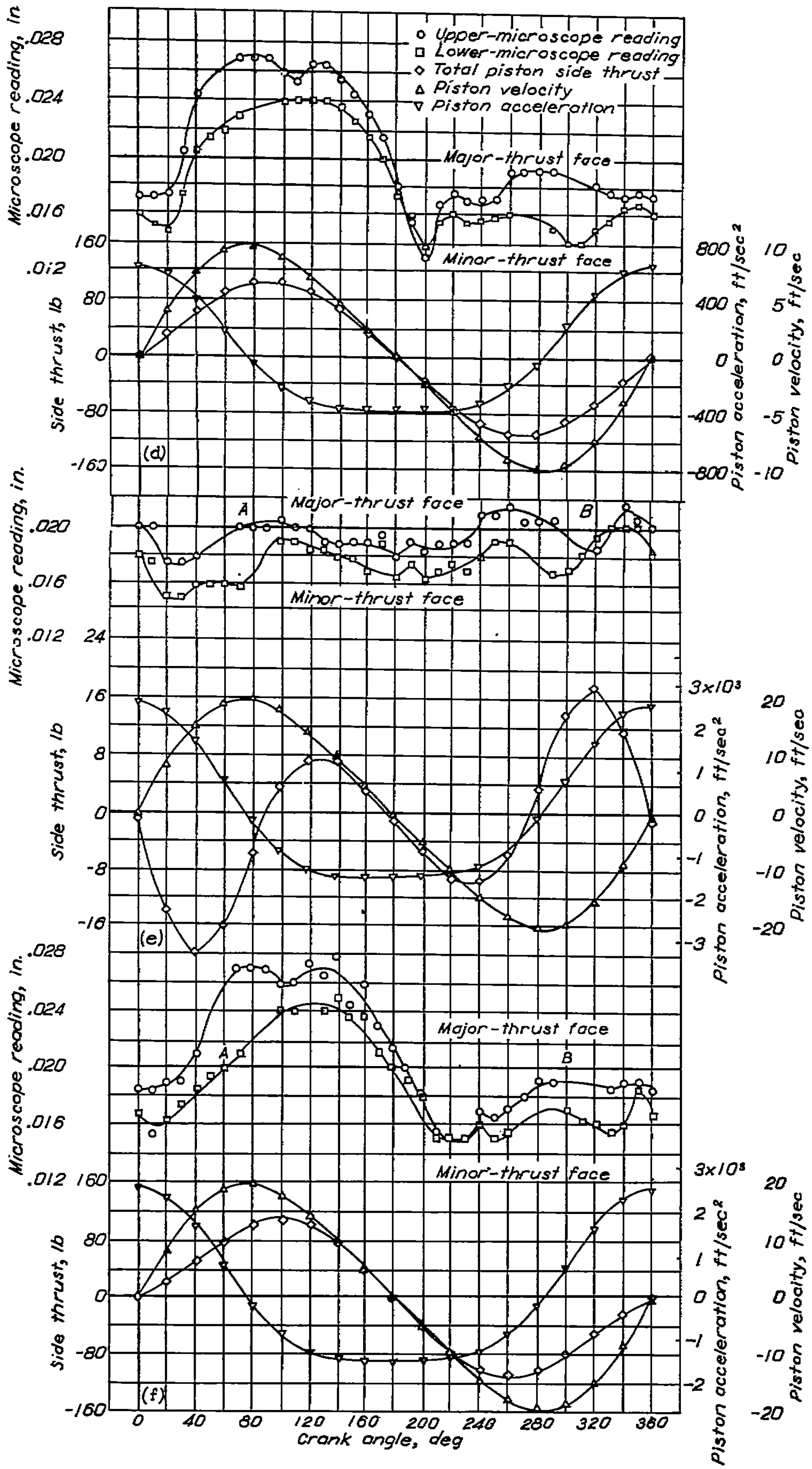
Inasmuch as the piston side thrust is symmetrical with respect to the zero axis in figure 7, piston displacement should be expected to be symmetrical with respect to the 0.020-inch line if all conditions were identical as the piston moved in either direction. More oil was always present, however, on the minor-thrust cylinder wall than on the major-thrust cylinder wall owing principally to the particular direction of crankshaft rotation used. The oil thrown from the big-end bearing was for the most part deposited on opposite cylinder walls with opposite directions of crankshaft rotation. The gravitational influence associated with the inclination of the cylinder at 45° to the vertical further contributes to the unequal distribution of oil. This difference in the quantity of oil present on opposite cylinder walls together with the influence of the inertia of the piston could at least partly account for the observed lack of symmetry of the piston-displacement curves.

Scattered-light photographs.—The photographs shown in figure 8 were obtained by the scattered-light method at an engine speed of 1000 rpm and at cylinder-head pressures of 0 and 50 pounds per square inch. Photographs of the piston operating under very light load (fig. 8 (a)) show that oil was present on both sides of the piston at nearly all times. The oil film that was present is shown to be striated and broken into relatively small patches. The piston was operating close to the center of the cylinder but somewhat closer to the minor-thrust cylinder face. The microscope data confirm this interpretation. (See points A and B, fig. 7(e).)



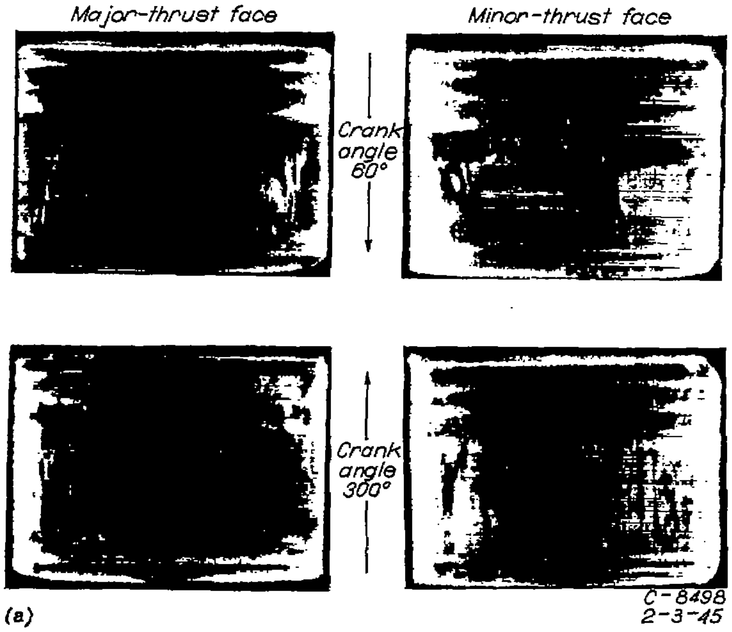
(a) Engine speed, 50 rpm; cylinder-head pressure, 0 pound per square inch.
(b) Engine speed, 50 rpm; cylinder-head pressure, 50 pounds per square inch.
(c) Engine speed, 500 rpm; cylinder-head pressure, 0 pound per square inch.

FIGURE 7.—Variation of piston orientation with crank angle.

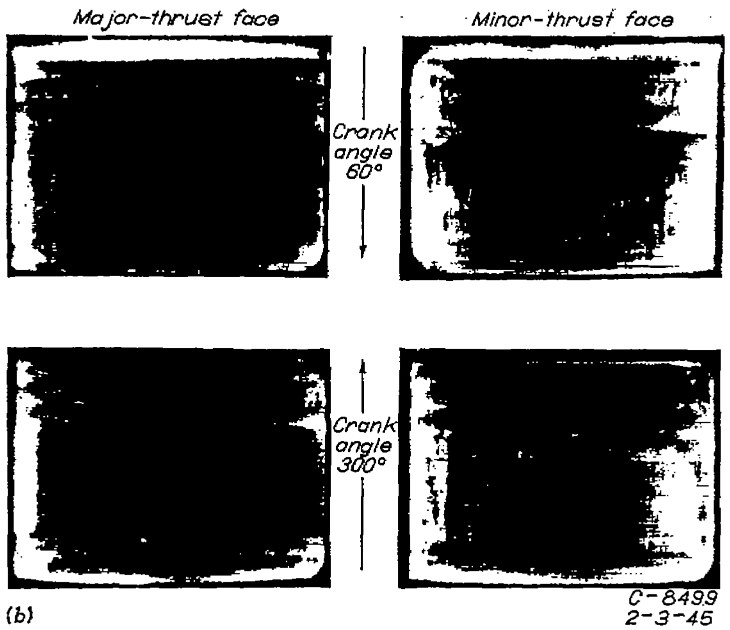


(d) Engine speed, 500 rpm; cylinder-head pressure, 50 pounds per square inch.
(e) Engine speed, 1000 rpm; cylinder-head pressure, 0 pound per square inch.
(f) Engine speed, 1000 rpm; cylinder-head pressure, 50 pounds per square inch.

FIGURE 7.—Concluded. Variation of piston orientation with crank angle.



(a) Engine speed, 1000 rpm; cylinder-head pressure, 0 pound per square inch.



(b) Engine speed, 1000 rpm; cylinder-head pressure, 50 pounds per square inch.

FIGURE 8.—Scattered-light photographs of running piston.

The photographs in figure 8(b) at a crank angle of 60° and a cylinder-head pressure of 50 pounds per square inch have the general appearance of a piston operating without load except that the quantity of oil present is less. The instantaneous total piston side thrust was 78 pounds in the major-thrust direction. The microscope data at point A in figure 7(f) indicate that at a crank angle of 60° the piston was in the act of moving from the minor-thrust face toward the major-thrust face, which explains the absence of a load-supporting oil film on the major-thrust face of the cylinder.

The photographs in figure 8 (b) at a crank angle of 300° and a cylinder-head pressure of 50 pounds per square inch show representative views of a loaded piston taken from the major-thrust and minor-thrust sides of the cylinder. The total piston side thrust was 80 pounds in the direction of the minor-thrust face and it is evident from the photographs that the load was being carried on this face. The load-carrying oil film shown in the lower-right photograph of figure 8 (b) is seen to have an approximately parabolic upper boundary. The corresponding piston-orientation data, given at point B in figure 7 (f), indicate that the piston was nearer the minor-thrust than the major-thrust cylinder face, substantiating the interpretation of the photographs. The upper-microscope reading is greater than the lower-microscope reading, which indicates that the piston was inclined with its crown nearer the major-thrust cylinder face. Inasmuch as a limited quantity of oil was available on the piston skirt, it is reasonable to assume that the upper film boundary would conform to a locus of constant film thickness. In order for the locus of constant film thickness to be an approximately parabolic curve, as shown in the lower-right photograph of figure 8 (b), the piston would have to be inclined as indicated by figure 7 (f).

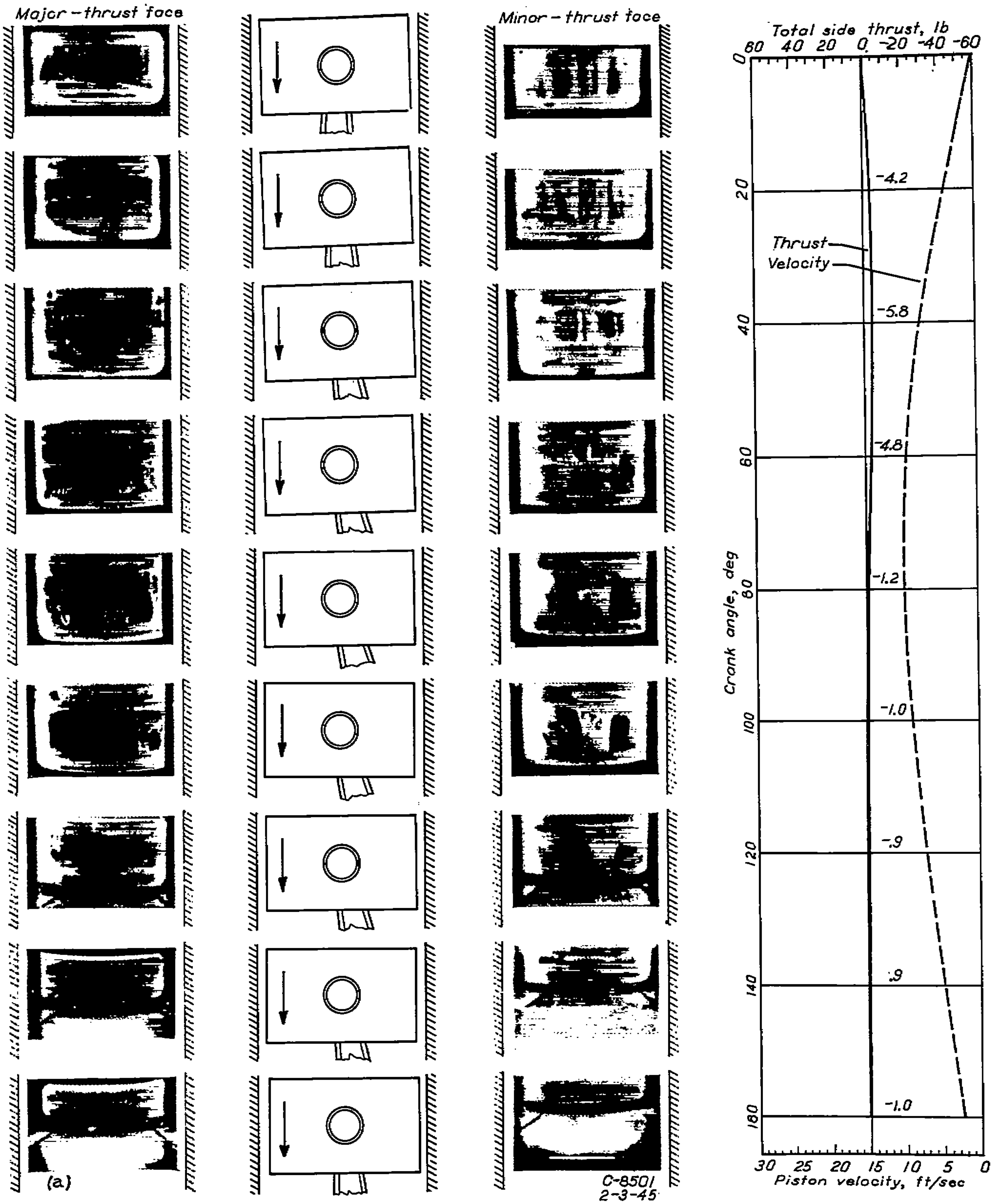
The series of photographs shown in figure 9 were also obtained by the scattered-light method. The photographs in the first column were taken from the major-thrust face of the cylinder; those in the third column were taken from the minor-thrust face. The entire piston was not visible in the region of bottom center because the piston recedes into an invisible portion of the test cylinder. The drawings in the second column show the orientation of the piston at crank-angle intervals of 20° as determined by the quartz-fiber investigation. The total piston side thrust and the piston velocity are plotted against crank angle in the fourth column.

The space between the piston skirt and the cylinder face was usually not filled with oil. When the piston was operating without side thrust, a greater portion of this space was filled with oil than when the piston was operating under load. (See figs. 8 and 9.) Under comparable conditions of operation, less oil is shown on the major-thrust face than on the minor-thrust face. This unequal oil distribution is probably due to the gravitation of oil to the minor-thrust face of the cylinder.

The area covered by the solid oil film when the cylinder was operated under load is an indication of the thickness of the film. Figure 9 (d) shows more area of oil in the loaded side of the piston at 500 rpm than is shown in figure 9 (h) at 1000 rpm with the same load. At 500 rpm the piston has time to squeeze the oil more thinly over the skirt. Additional information concerning oil-film distribution is presented in the discussion of fluorescent-light photographs.

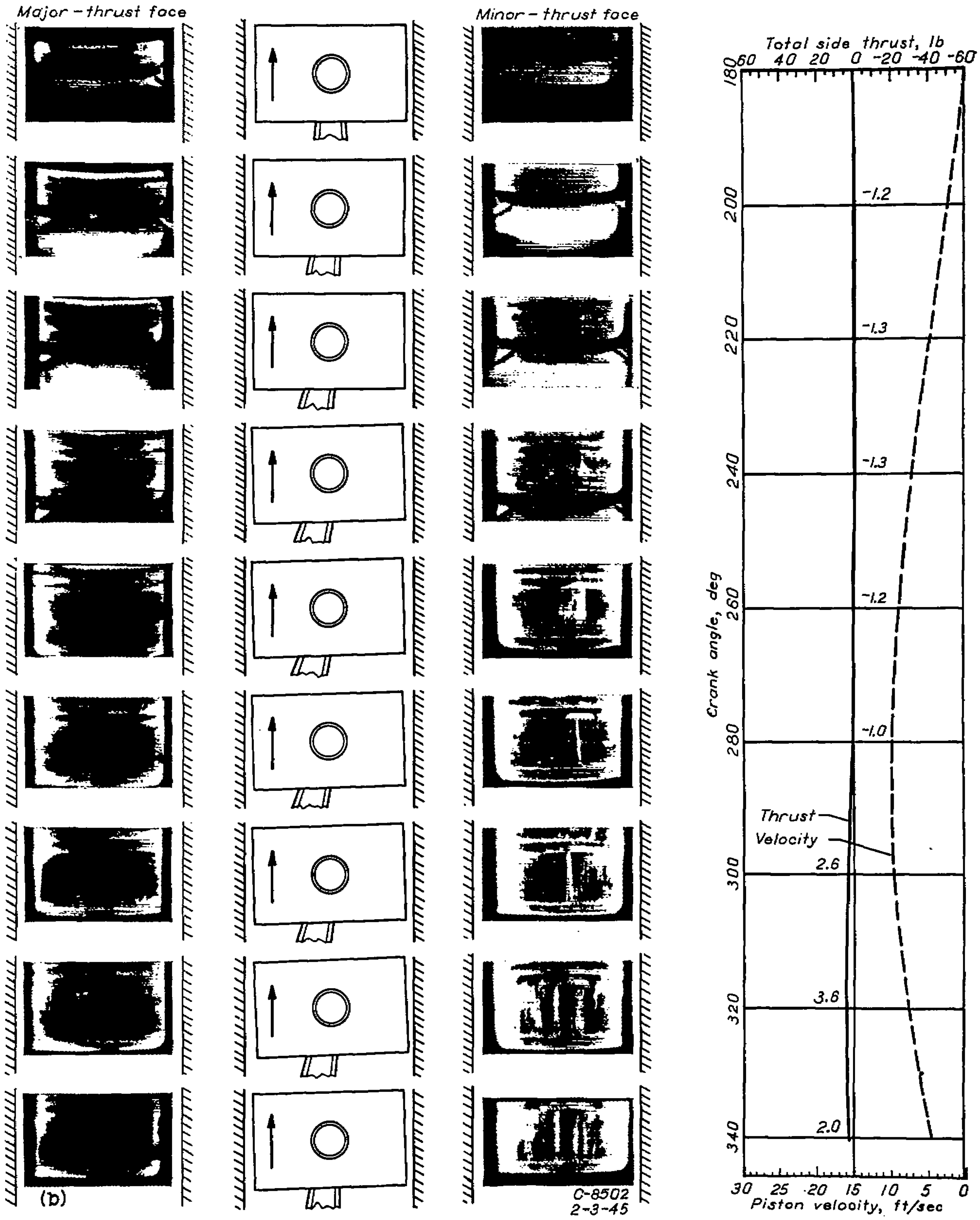
The piston may be considered analagous to a curved slider bearing. The oil supply to the piston skirt is limited by the rings on either side of the skirt, whereas the usual slider bearing operates with abundant lubrication. The orientation of the piston and the shape of the oil film are, consequently, dependent upon the amount of oil present in addition to the important variables in plane-slider theory (load, velocity, and viscosity).

A load-carrying oil film was never observed to cover an arc of more than 180°, nor was oil ever observed to flow circumferentially around the piston.



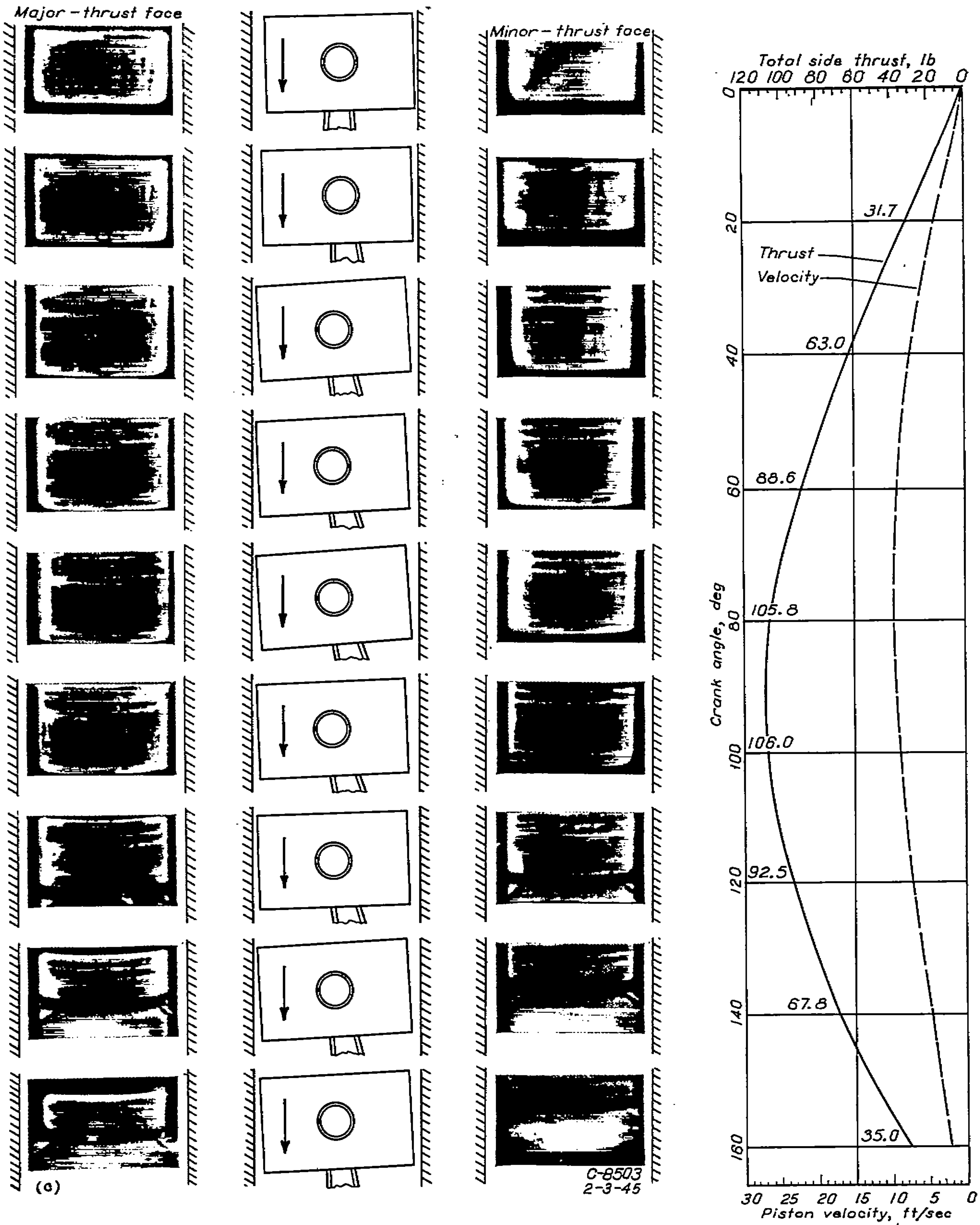
(a) Engine speed, 500 rpm; cylinder-head pressure, 0 pound per square inch; piston moving down.

FIGURE 9.—Photographs taken from major-thrust and minor-thrust faces of cylinder using scattered-light method



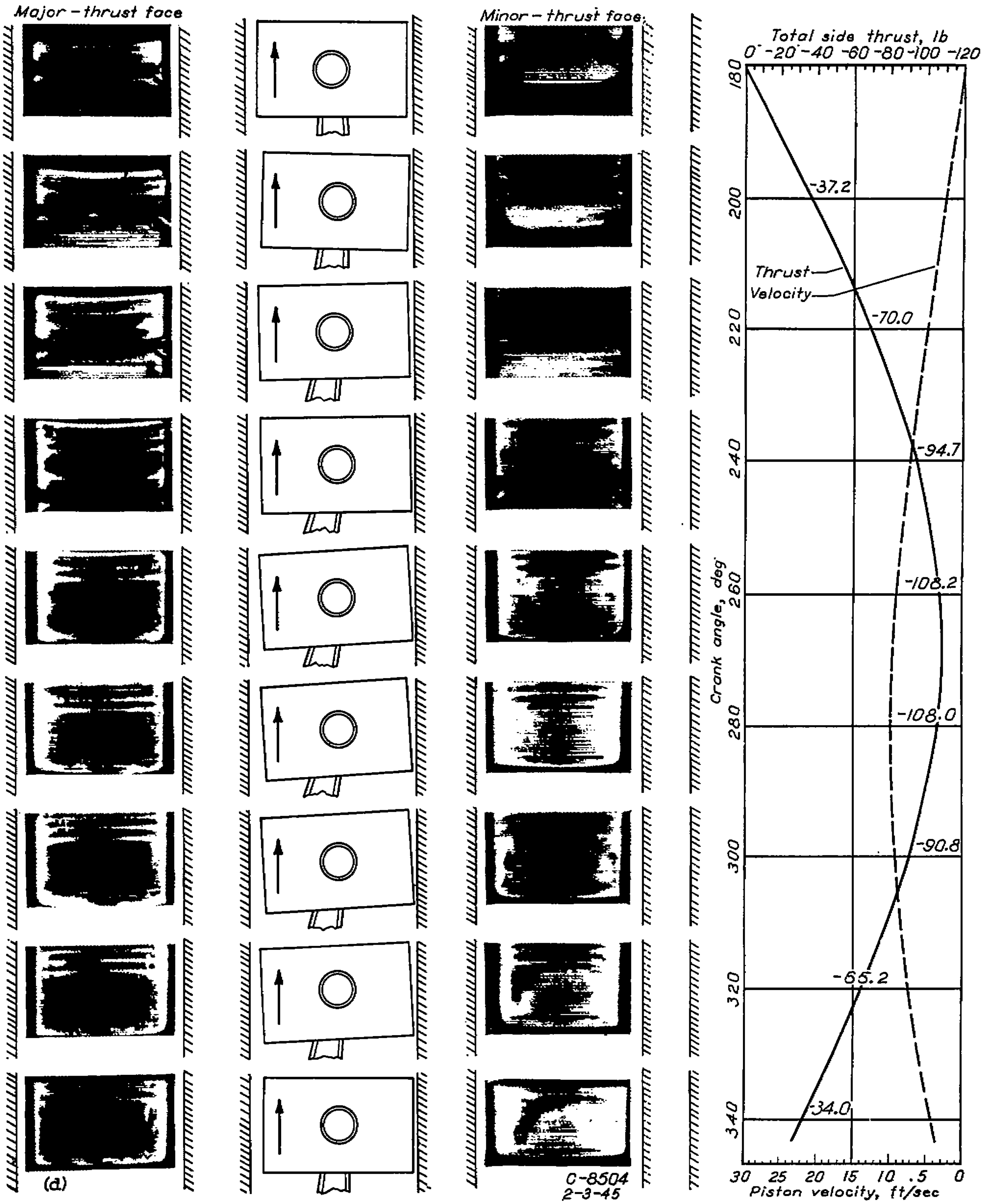
(b) Engine speed, 500 rpm; cylinder-head pressure, 0 pound per square inch; piston moving up.

FIGURE 9.—Continued



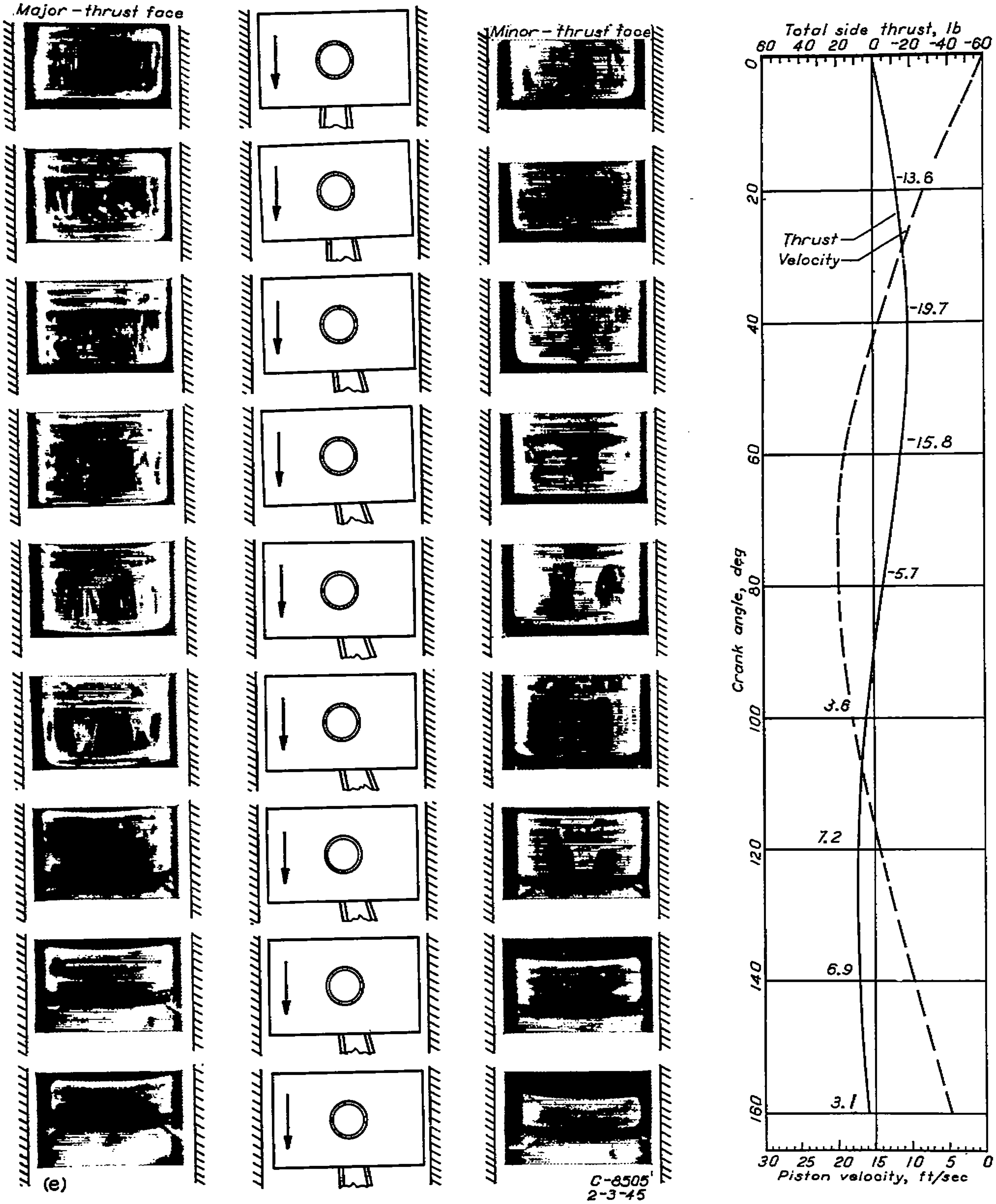
(c) Engine speed, 500 rpm; cylinder-head pressure, 50 pounds per square inch; piston moving down.

FIGURE 9.—Continued.



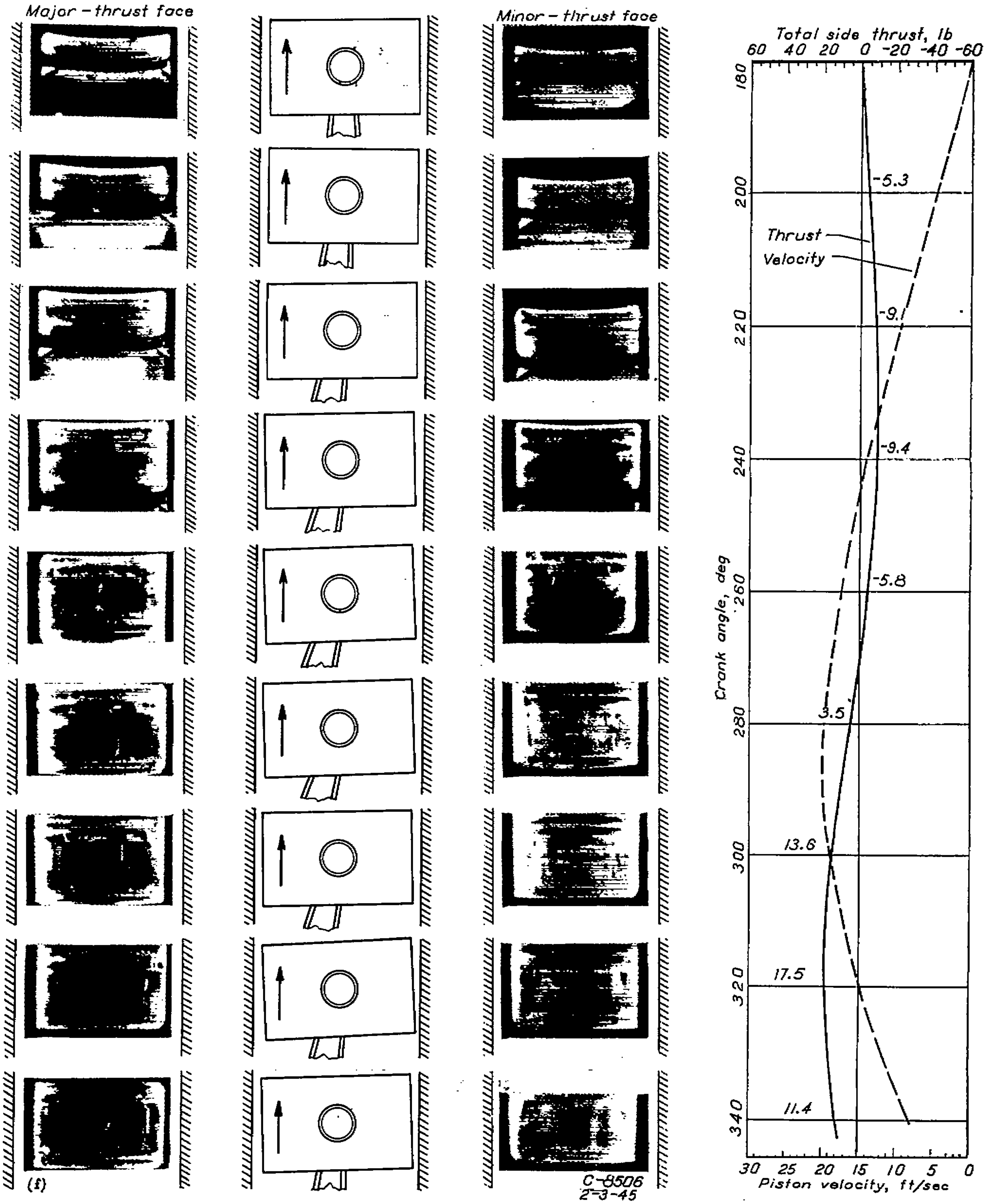
(d) Engine speed, 500 rpm; cylinder-head pressure 50 pounds per square inch; piston moving up.

FIGURE 9.—Continued.



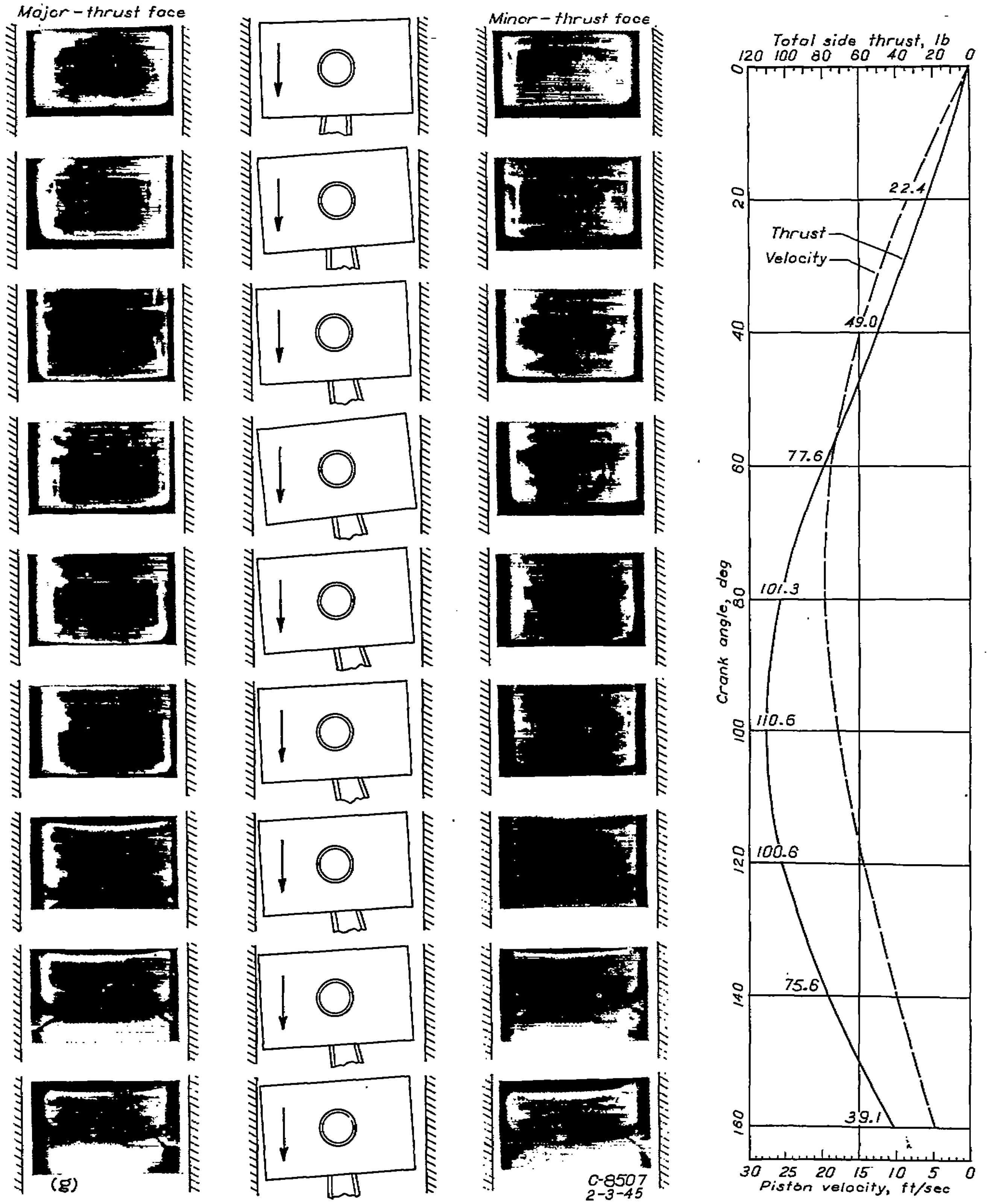
(e) Engine speed, 1000 rpm; cylinder-head pressure, 0 pound per square inch; piston moving down.

FIGURE 9.—Continued.



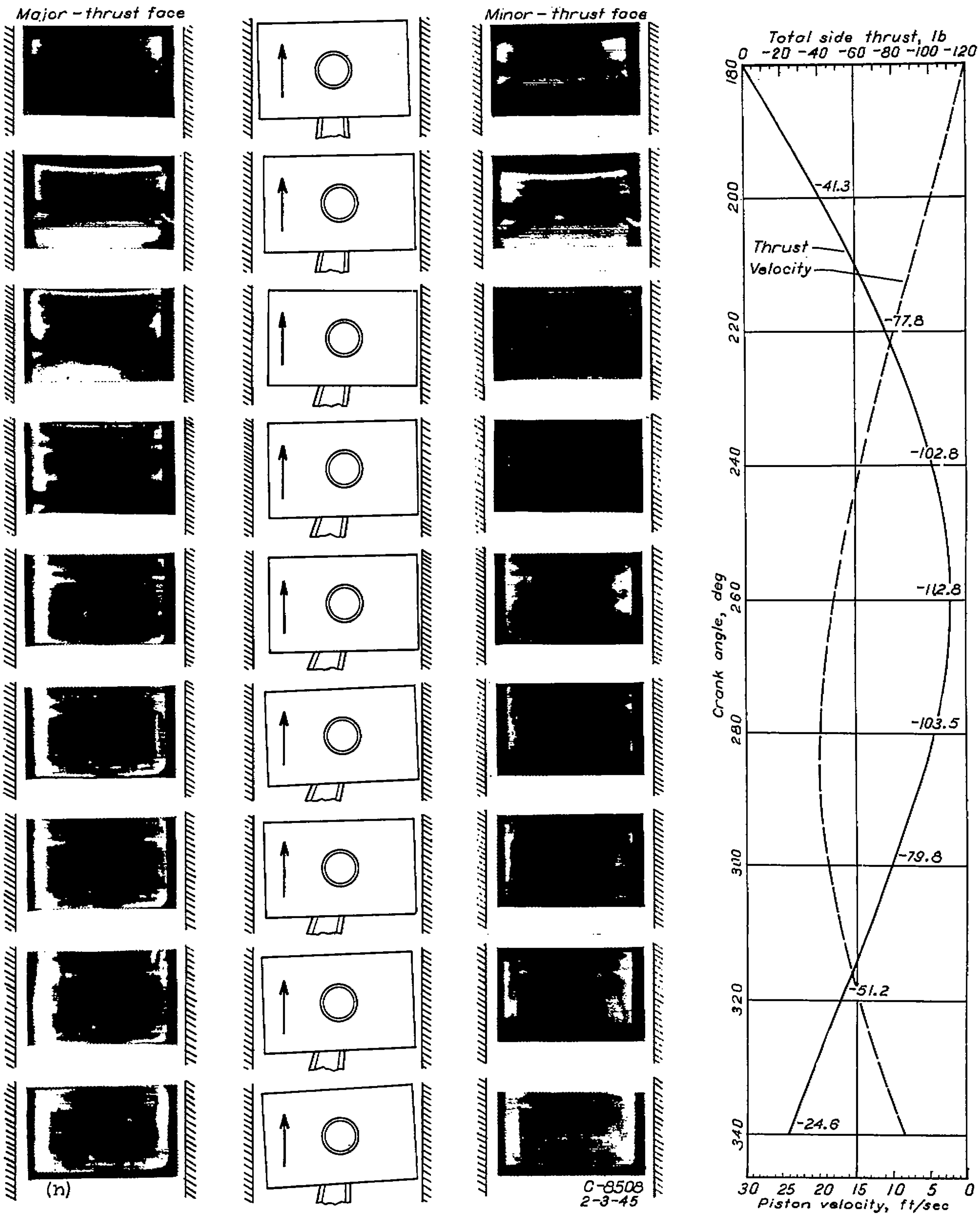
(f) Engine speed, 1000 rpm; cylinder-head pressure, 0 pound per square inch; piston moving up.

FIGURE 9.—Continued.



(g) Engine speed, 1000 rpm; cylinder-head pressure, 50 pounds per square inch; piston moving down

FIGURE 9.—Continued.



(h) Engine speed, 1000 rpm; cylinder-head pressure, 50 pounds per square inch; piston moving up.

FIGURE 9.—Concluded. Photographs taken from major-thrust and minor-thrust faces of cylinder using scattered-light method.

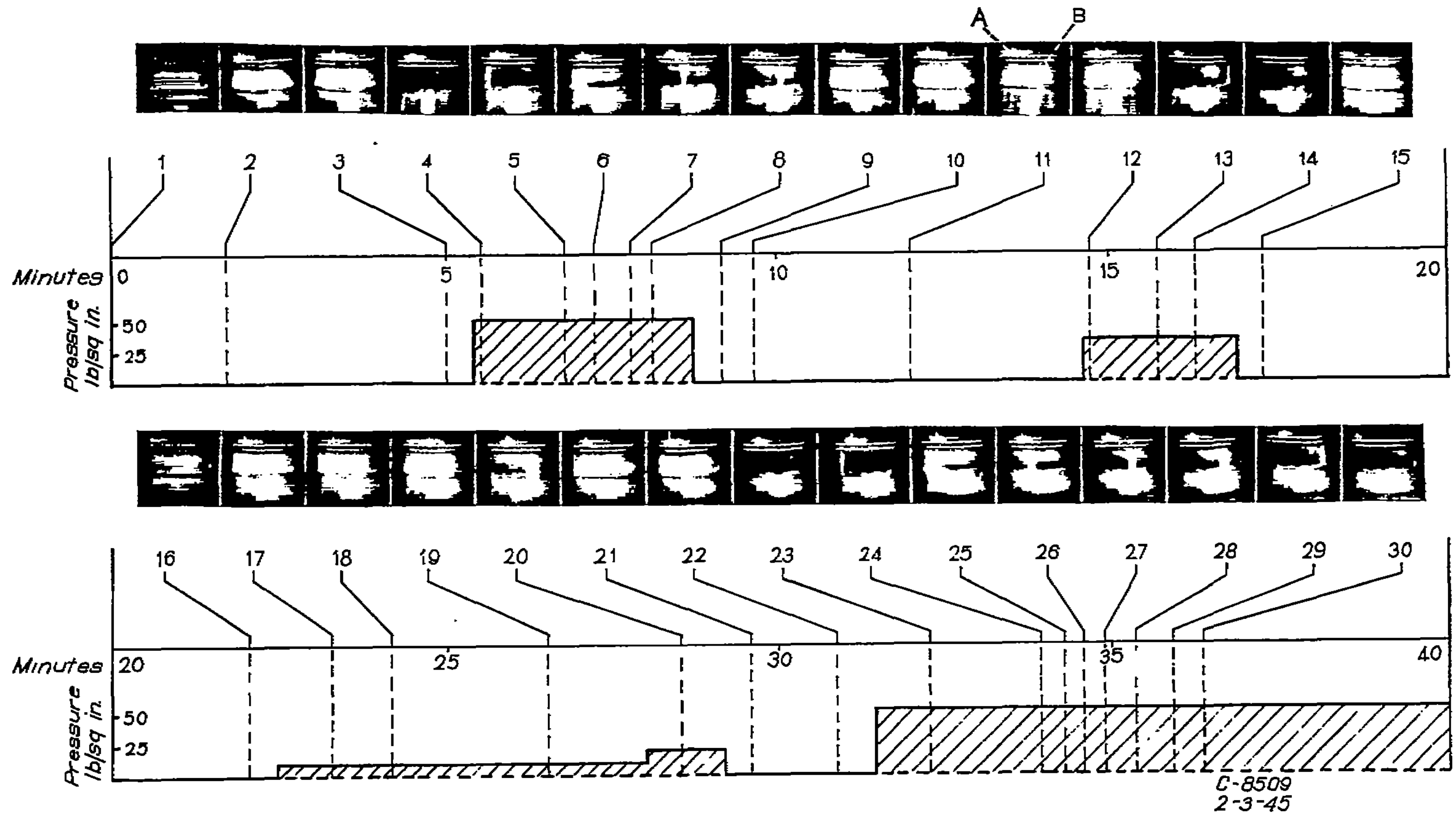


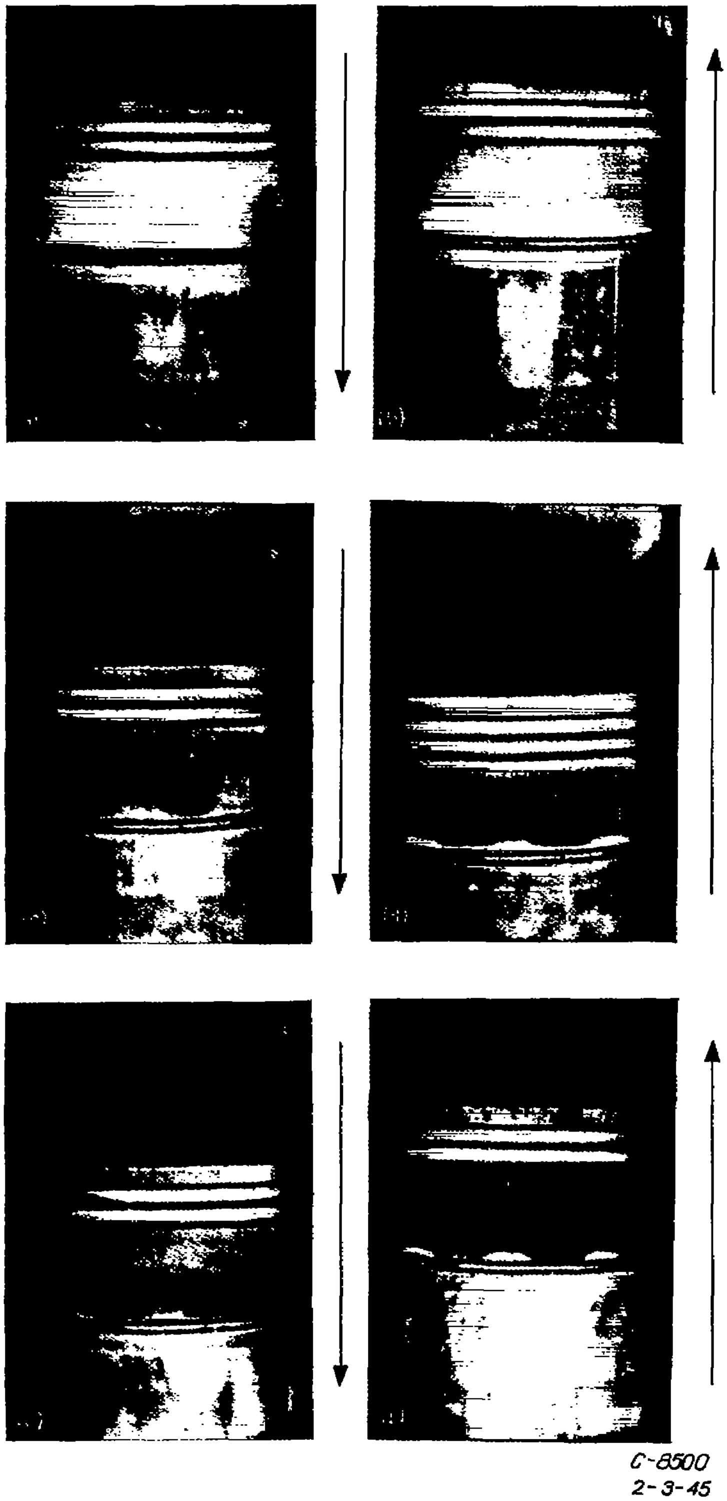
FIGURE 10.—Sequence of fluorescent-light photographs showing rotation of piston rings as observed from major-thrust face of cylinder. Engine speed, 1000 rpm; crank angle, 270°.

Fluorescent-light photographs.—The photographs in figure 10 are a representative sequence and illustrate the influence of the cylinder-head pressure upon the relative angular positions of the top and third compression-ring gaps, designated A and B, respectively, in the figure. These photographs were taken using 25 successive flashes for each picture. The time at which each picture was taken is indicated on the figure together with the prevailing pressure (cross-hatched areas). All rings in the test piston were observed to rotate at different rates. Rings were observed to rotate at a rate as high as 1 rpm at an engine speed of 1000 rpm. The direction and the rate of rotation may be varied by changes in either engine speed or cylinder-head pressure. The response of the piston rings to changes in engine speed or cylinder-head pressure is approximately reproducible.

The oil-metering action of the third compression ring is evident in figure 10. The sequences of photographs from 4 to 8 and from 23 to 30 were taken at the times indicated but under otherwise similar conditions. These two series of photographs cover periods in which the third compression-ring gap (B) moved from left to right. A gradual increase in the quantity of oil on the skirt is evident as the ring gap comes into view (photographs 5 and 24). The similarity of corresponding photographs in the two sequences indicates that the observed variation in skirt lubrication is associated with the orientation of the piston ring. Because nearly all the piston rings have noncircular pressure diagrams, such a fluctuation is to be expected. The relative orientation of the several piston rings may account for observed fluctuations in oil consumption, blow-by, and friction horsepower of a test engine.

A number of fluorescent-light photographs taken from the major-thrust face are shown in figure 11. These photographs offer additional evidence of the lag in lateral movement of the piston. A residual oil pattern, found at the top of the cylinder, was evident only when there was sufficient side thrust to cause appreciable lateral movement of the piston. Figures 11 (c) to 11 (f) show this pattern, which was obtained only with cylinder-head pressure. (No such pattern is evident in figs. 11 (a) and 11 (b).) The top horizontal band of the pattern coincides with the top of the piston at top center and is probably produced by oil thrown from the crown of the piston upon its reversal. The lower V-shaped portion of the pattern begins and ends at crank-angle values approximately at the beginning and the end of the lateral piston movement, as shown in figure 7. The oil was evidently squeezed from the top ring groove and the space above the ring as the piston moved toward the major-thrust face and was metered past the top edge of the piston. This metering action is attested by the fact that all striations visible in the pattern have counterparts on the top of the piston.

The fluorescent-light photographs in figure 11 show no oil film between the rings and the cylinder face. The comparable scattered-light photographs in figure 8, however, indicate



(a) Cylinder-head pressure, 0 pound per square inch; crank angle, 60°.
(b) Cylinder-head pressure, 0 pound per square inch; crank angle, 300°.
(c) Cylinder-head pressure, 30 pounds per square inch; crank angle, 80°.
(d) Cylinder-head pressure, 30 pounds per square inch; crank angle, 270°.
(e) Cylinder-head pressure, 50 pounds per square inch; crank angle, 80°.
(f) Cylinder-head pressure, 50 pounds per square inch; crank angle, 290°.

FIGURE 11.—Photographs taken from major-thrust face of cylinder using fluorescent-light method. Engine speed, 500 rpm.

that oil was present. When the intensity of the fluorescent light from the oil film beneath the rings was compared with that from films of known thickness, the piston-ring oil films shown in figure 11 were estimated to be 0.0001 inch or less.

The great difference in the amount of lubricant present on the piston skirt operating with and without piston side thrust is shown in figure 11. This difference might be explained by the lateral motion of the piston under load. As the piston moved back and forth in the cylinder, considerable pressure was developed, which caused the oil to flow out past the piston rings. The upward flow of oil past the oil-control rings was relatively small because there was little driving pressure from below. Consequently, little oil collected on the skirt when the piston was operated under high load. Under low-load conditions, the piston operated approximately concentric with the cylinder and little pressure was developed to force the oil from the skirt. Figure 11 shows that there is real danger of making the oil-control rings too efficient. If the oil-control rings hold back too much oil, the lubrication of the compression rings, as well as of the piston skirt, will be endangered. The ideal condition, as far as piston-skirt lubrication is concerned, is to operate without skirt rings.

SUMMARY OF RESULTS

A study of piston-skirt lubrication was made by direct observation of the oil film, utilizing two photographic techniques, in investigations with a glass-cylinder engine motored over a limited number of conditions (cylinder-head pressures of 0 and 50 lb/sq in. and engine speeds to 1000 rpm). The results of this investigation are summarized as follows:

1. In accordance with hydrodynamic theory, the piston was inclined in such a direction as to favor an oil wedge on the loaded side of the cylinder during the greater portion of the engine cycle.
2. The piston moved laterally from the major-thrust to the minor-thrust face of the cylinder under the influence of piston side thrust.
3. The piston rings, particularly the lower oil-control rings, had a considerable effect upon piston-skirt lubrication.
4. The amount of lubricant present on the piston skirt varied with the relative angular position of the piston rings and with cylinder pressure.
5. Rate and direction of piston-ring rotation varied with cylinder-head pressure and engine speed. Rates of rotation as high as 1 rpm at an engine speed of 1000 rpm were observed.
6. The estimated thickness of an oil film on the piston-ring face, when operating under load, was 0.0001 inch or less.
7. In general, much less oil was present on the upper side of the inclined piston than on the lower side. In the present case, the upper side was the major-thrust face.

AIRCRAFT ENGINE RESEARCH LABORATORY,
NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
CLEVELAND, OHIO, *August 1, 1945*.

APPENDIX

COMPUTATION OF TOTAL SIDE THRUST OF PISTON

The total piston side thrust is considered herein as the thrust exerted by the piston skirt upon the cylinder and is composed of four components: reciprocating inertia force, gas force, gravitational force, and friction force. The methods of computing each of these components are outlined and discussed in the following paragraphs. The friction component was neglected in making the computations of total side thrust because of its relatively insignificant magnitude, as subsequently shown in the discussion of friction force.

Reciprocating inertia force.—The reciprocating inertia force is produced by the acceleration of the mass of the piston assembly and the mass of the upper end of the connecting rod and may be computed by the method of reference 16. The total mass of the connecting rod may be replaced by the kinematically equivalent system of two masses (M_A and M_B) shown in the crank-mechanism diagram (fig. 12)

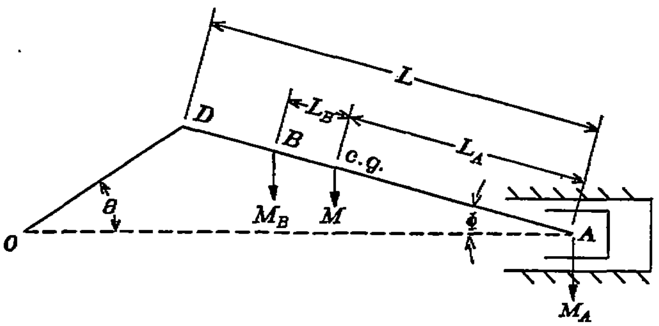


FIGURE 12.—Crank-mechanism diagram.

where

- D center of crank pin
- L connecting-rod length
- L_A distance from center of gravity to wrist pin
- L_B distance from center of gravity to point B
- M total mass of connecting rod
- M_A component of mass of connecting rod concentrated at point A
- M_B component of mass of connecting rod concentrated at point B
- O center of crank shaft

The masses M_A and M_B will cause the connecting rod to have the same total mass, moment of inertia, and center of gravity as the true rod if these masses and their locations are chosen according to the following equations:

$$L_A L_B = \frac{I}{M}$$

$$M_A = \frac{L_B}{L_A + L_B} M$$

$$M_B = \frac{L_A}{L_A + L_B} M$$

where I is the moment of inertia about the center of gravity.

The center of gravity of the connecting rod was determined by balancing it on a knife edge, and the moment of inertia was determined in the usual manner from the frequency of oscillation of the rod about the center line of the wrist pin. Acceleration of the point A was obtained from Smith's compilation of piston accelerations (reference 17), whereas the acceleration of point B was obtained by the simple construction shown in the acceleration diagram (fig. 13)

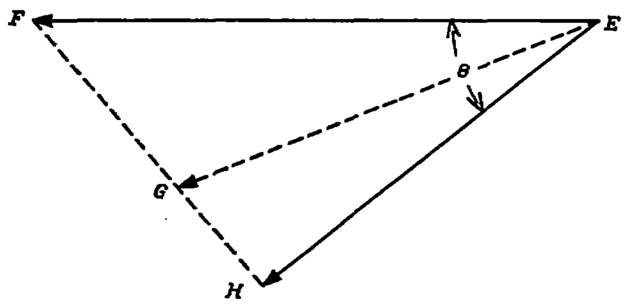


FIGURE 13.—Acceleration diagram.

where

- EF piston acceleration (from reference 17)
- EH centripetal acceleration of D in crank-mechanism diagram$=OD$ (angular velocity of OD)^2
- EG acceleration of B in crank-mechanism diagram

Point G is located by making $FG=\left(\frac{AB}{AD}\right)FH$.

The force diagram may be completed and the inertia component of piston side thrust determined from the following equation, which is derived in reference 16:

$$JK = F_B\left(\frac{AB}{AD}\right)$$

where

- JK force vector shown in force diagram.
- FB component of piston side thrust due to force exerted by B in crank-mechanism diagram.

The resulting force diagram is shown in figure 14

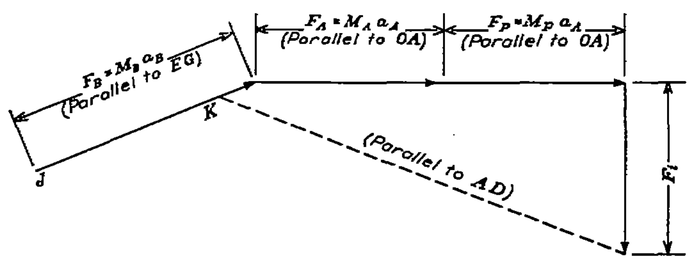


FIGURE 14.—Force diagram.

where the F's are various components of thrust and the a's are accelerations; subscripts B, A, p, and i refer to points B and A, gas pressure, and inertia, respectively.

Computed values of inertia side thrust for an engine speed of 1 rpm are as follows:

Crank angle (deg)	Inertia side thrust (lb)
0	0
20	1.25×10^{-5}
40	1.86
60	1.47
80	.46
100	−.47
120	−.83
140	−.80
160	−.42
180	0

The inertia side thrust at any engine speed is obtained by multiplying these values by the square of the speed in revolutions per minute. A positive value indicates a force directed against the major-thrust cylinder face.

Gas force.—The component of piston side thrust that is due to gas pressure is obtained by multiplying the gas force acting on the piston by tan ϕ. (See crank-mechanism diagram.) A positive gas pressure will cause a piston side thrust in the major-thrust direction for crank-angle values from 0° to 180° and in the minor-thrust direction from 180° to 360°.

Gravitational force.—The inclination of the cylinder center line at an angle of 45° gives rise to a gravitational component of piston side thrust. The gravitational force acting vertically at the wrist pin is equal to the weight of the piston assembly plus the weight of the upper end of the connecting rod (32.2 M_A). This force is approximately 1.5 pounds and may be resolved into two components: one perpendicular to the cylinder axis, and the other along the connecting-rod axis. The magnitudes of these components vary with the inclination of the connecting rod. The component perpendicular to the cylinder axis (gravitational side thrust) varies from approximately 0.8 to 1.4 pounds during each revolution of the crank. Inasmuch as the gravitational side thrust is small, a constant value of 1.1 pounds in the minor-thrust direction has been assumed at all crank angles.

Friction force.—The friction developed between the piston assembly and the cylinder face constitutes another component of piston side thrust. This component is equal to the product of the total normal force between the piston assembly and the cylinder, the coefficient of friction, and the tangent of the angle ϕ. The total normal force between the piston assembly and the cylinder is considered herein to be the nonfriction side thrust and the side thrust due to piston-ring tension and pressure behind the rings.

When relatively high pressure behind the rings and an over-all coefficient of friction of 0.05 were assumed, the frictional contribution to the piston side thrust was found to be less than 3.5 percent of the nonfrictional components at engine speeds up to 1000 rpm. The friction component of side thrust, like the gravitational component, acts in the minor-thrust direction at all crank angles.

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