EFFECT OF RING TENSION, FACE WIDTH, AND NUMBER OF RINGS ON PISTON RING FRICTION

by

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(1951)

Signature of authors ______________________

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Department of Naval Architecture and Marine Engineering,
May 18, 1951.

Certified by

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ABSTRACT

A. EFFECT OF RING TENSION, FACE WIDTH, AND NUMBER OF RINGS ON PISTON RING FRICTION.

B. (1) William Comrie Gibson, Lt. U. S. Navy
   (2) Francis Avery Packer, Jr., Lt. U. S. Navy

C. Submitted for the degree of NAVAL ENGINEER in the Department of Naval Architecture and Marine Engineering on May 18, 1951.

D. Salient features of the report.

(1) Further tests were conducted with the M. I. T. Friction Engine. Isolation and measurement of Piston Ring Friction work during a crankshaft revolution at 1000 RPM while firing was satisfactorily accomplished. The titulary objective was achieved for the given set of conditions.

(2) Maximum feasible tensions caused a barely detectable increase in total piston ring friction work.

(3) Wide rings showed only slightly greater friction than the narrow rings.

(4) Reducing the number of piston rings from three (3) to one (1) caused a reduction in FMEP by about 25% in the case of the 3/16" rings, but in the case of the 1/16" rings, friction remained nearly constant.

(5) The 3/16" interrupted surface rings indicated values of FMEP, after run-in, comparable to that of the similar lapped surface rings.

(6) Further work with the MIT friction engine should include:

(a) Effect of ring size, profile, and numbers.
(b) Effect of interrupted surfaces.
(c) Effect of increasing piston speed so that ring variable effects might be more pronounced.

May 18, 1951.

Professor Joseph S. Newell
Secretary of the Faculty
Massachusetts Institute of Technology

Dear Sir:

In partial fulfillment of the requirement for the degree of Naval Engineer, from the Massachusetts Institute of Technology, we hereby submit our thesis entitled: EFFECT OF RING TENSION, FACE WIDTH, AND NUMBER OF RINGS ON PISTON RING FRICTION.

Respectfully,

W. C. Gibson, Lt., USN

F. A. Packer, Jr., Lt., USN

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ABBREVIATIONS AND SYMBOLS

S.A.	Spark Advance (°BTC)
BTC	Before Top Center
q_b	Blow-By (Ft^3/Hr)
R	Rotometer Scale Reading (Cm)
h	Δp across air inlet orifice (in.H_2O)
B.L.	Scale reading of dynamometer (in.Hg.)
T_1	Air Temperature before orifice (°F)
T_i	Air Temperature at engine inlet (°F)
T_j	Water jacket temperature (°F)
T_h	Cylinder head temperature above junk rings (°F)
T_L	Crankcase oil temperature (°F
p_L	Oil pressure (psig)
S_L	Junk ring oil supply pump stroke setting (in.)
q_L	Junk rings oil supply rate (CC/Min)
p_i	Inlet manifold pressure (in.Hg.)
p_e	Exhaust manifold pressure (in.Hg.)
p_1	Pressure before orifice (in.Hg.Abs.)
F	Fuel consumption (lbs/sec)
A	Air consumption (lbs/sec)
F/A	Fuel/air ratio
p_o	Barometric pressure (in.Hg.)
θ	Crank angle degrees
s	Piston speed ft/min
S	Piston stroke ft

FMEP	Total Piston Ring Friction Mean Effective Pressure	psi
CP-FMEP	Compression-Power Stroke piston ring fmep.	
EI-FMEP	Exhaust-Intake Stroke piston ring fmep.	
BMEP	Brake Mean Effective Pressure psi observed with dynamometer	

ACKNOWLEDGEMENT

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The cooperation of Perfect Circle Cooperation in furnishing necessary piston rings, and the interest and bibliographical assistance of M. D. Hersey, U.S.N.E.E.S., Annapolis, Md., is also appreciated.

INTRODUCTION

This thesis is to be a continuation of the work begun by Forbes and Taylor(1), and later continued by Leary and Jovellanos(2), and Livengood and Wallour(3). Forbes and Taylor demonstrated that the piston and ring friction increased with increased oil viscosity and increased slightly with increasing indicated mean-effective pressure. Combined piston and ring friction was greater at higher engine speeds. The results were of a preliminary nature, however.

The work of Leary and Jovellanos extended the use of the original apparatus to show that piston and ring friction decreased quite rapidly during the first hour of running (starting with new rings and cylinder) and more slowly for an extended period, thereafter. An interesting result of this investigation was the observation that the friction work measured during the process of ring scuffing was not greater than that for normal operation.

The work of Livengood and Wallour confirmed the results of the previous investigations. In addition, it was shown that piston-ring friction decreased with increased cylinder jacket temperatures, and that lowering the manifold pressure reduced the piston-ring friction. The cast iron piston rings operating in a SAE 4140 steel barrel had the lowest friction of the combinations tested. The cast-iron-rings in a porous chrome barrel had the greatest friction, and the SAE 4140 barrel with one chrome top ring had intermediate friction. These differences were small, however.

The results of the previous investigations appeared to show that the techniques developed are sound and could yield useful information. The electromagnetic pick-up used by Livengood and Wallour (3) and modified in February 1951 gave satisfactory results and adequate sensitivity. The much stiffer diaphragm for the combustion-cylinder suspension increased the natural frequency of the cylinder and thus improved the detail of the friction records obtained.

Using the apparatus of Livengood and Wallour, it was decided to extend the studies to include other variables: (1) face width of the rings, (2) ring tension, and (3) number of rings. The rings are to have a face width of 1/16" and 3/16", each to be obtained in both maximum and minimum feasible diametral tensions. It was desirable to check the friction characteristics of an interrupted surface piston ring as compared with a lapped surface ring. The Perfect Circle Corporation supplied the necessary piston rings to make the comparisons.

An oil scraper ring was not used for the following reasons:

(1) Crosshead seals eliminate problem of oil seeping into combustion chamber from below.

(2) Oil scraper ring would introduce an unnecessary component of friction which would further complicate the analysis.

The apparatus used in this investigation was basically the same as that used by the previous investigators. It consisted of an elastically mounted combustion-cylinder sleeve that would have a small motion along the axis of the sleeve due to the friction forces between the sleeve and the piston rings. This motion was recorded photographically during the test runs and was the means of determining the average friction forces.

Figure I shows the cylinder and crosshead assembly. The light cylinder was clamped on the inner circumference of the two annular steel diaphragm springs. The outer edges of these diaphragms were clamped to a heavy cast-iron barrel. The space between the sleeve and the cylinder barrel formed the water jacket. The clamping was accomplished by the cylinder head at one end and a steel plate at the other.

The cylinder head was provided with unsplit junk rings which, together with oil supplied under pressure, formed a seal which effectively closed the combustion chamber against leakage of the gases. These rings had approximately 0.002 inch diameteral clearance within the sleeve. The junk-ring grooves were deep enough to allow the rings to center themselves properly with the sleeve. The lands between the junk rings had a diameter small enough to ensure that there was no contact with the sleeve.

Vent holes were provided in the cylinder head to allow oil and gases which had leaked past the junk rings to escape and thus avoid a rise in pressure above the upper diaphragm. In order to reduce such gas leakage to minimum, a metered supply of oil was introduced under pressure into a passage leading to the junk-ring grooves. Most of this oil escaped through the leak-off passages above the diaphragm, but some found its way into the combustion chamber and onto the cylinder wall, where it provided piston ring lubrication. Piston rings received all their lubrication in this manner, except for a negligible amount traveling from the crankcase up past the crosshead seals. This "top-cylinder"lubrication is the primary point of difference between the friction engine and an internal combustion engine in service. However this should cause negligible difference when optimum oil flow to junk rings is maintained, and only comparative results are desired. The cylinder head was cooled by cold water flowing through its jacket passages. Low temperature of the head was desired in order to keep the oil viscosity at the junk rings as high as possible.

Two spark-plug wells, which are shown in figure I, were sealed off from the jacket coolant by rubber seals which exerted no appreciable constraint on the axial motion of the sleeve.

A water-jacketed crosshead cylinder was installed between a CFR crankcase and the cylinder assembly described above. An aluminum-alloy crosshead, operating in this cylinder, carried on its upper end a special piston. In order to reduce leakage of oil by the crosshead and scuffing by the seals, the crosshead was chrome plated. Since no wrist pin was required for this piston, the central portion was reduced in diameter to decrease the weight. The piston to take the 1/16" rings was on hand. (See Figure III). However, for the 3/16" rings, a new piston had to be made. This is shown in figure II. The crosshead and piston assembly was designed so that during engine operation only the piston rings touched the upper, or combustion, cylinder sleeve. Two oil seals were installed below the combustion cylinder. The upper seal prevented the oil and gases which leaked by the piston rings from escaping into the crankcase. These products were led out through passages above this seal and thus could be measured. The lower seal helped to prevent the crankcase oil which lubricated the crosshead from contaminating the lubricant supplied through the junk rings to the combustion cylinder. Oil caught by lower seal was returned to crankcase automatically so that crankcase level was held nearly constant.

PROCEDURE

It was noted by Livengood and Wallour (3) that the greatest change in piston ring friction mean-effective pressure occurred during the first hour of run-in, and that after the third hour there was no significant change in the friction. For this reason it was decided to limit the run-in tests to three[3] hours as opposed to ten[10] hours for the test runs of Livengood and Wallour.

The piston rings used were supplied by the Perfect Circle Corporation of Hagerstown, Indiana, to the same specifications as those of reference 3. Piston ring data is found in Table I. The diametral tensions and gaps of the rings were measured before and after each run by means of a device described by Leary and Jovellanos[2].

Before each run the SAE 4140 liner was lapped fifty strokes with number 600 emery using an old piston and cast iron rings as a tool. This insured that each run would commence with the liner in a similar surface condition.

Before the combustion-cylinder sleeve assembly was installed in the engine, the sleeve motion measuring apparatus was calibrated by loading it with test weights. These curves are shown in figures VII and VIII.

Prior to each run-in test the apparatus was flushed with clean oil and refilled to the same crankcase level. Runs were made under the following conditions:

Engine speed, rpm	1000
Fuel-air ratio	Best Power
Spark Advance	Best Power
Manifold pressure, in. Hg. abs.	29.2 $\pm$0.2
Crankcase oil temp. °F	150 $\pm$ 2

Cylinder Head Temperature, °F			65 ± 2
Combustion and Crosshead Cylinder Jacket Temperature, °F			180 ± 1
Inlet mixture temperature, °F			150 ± 1
Oil pressure, psi			50
Lubricating oil	Texas Co.	URSA P-20	
Specific gravity at 60°F			0.88
SSU at 130°F			160.3
SSU at 210°F			52.8

(Oil for all runs taken from same barrel. Oil is wholly parafine base and distilled mineral oil with no additives; used for both junk rings and crankcase)

Fuel: Marine white, unleaded octane rating	78

The engine was run for about 3 hours under these conditions, and friction records were taken photographically approximately every half hour. After the test run with each set of rings the engine was stopped and dismantled to remove the two bottom rings from the piston. The engine was then reassembled and the test run continued for about two hours with one piston ring, during which time friction records were taken. It was not felt necessary to recalibrate the sleeve-motion measuring apparatus after the change of rings because of the rapidity of the change. Subsequent calibrations proved the electro-magnetic pick-up to be operating satisfactorily.

The electromagnetic pickup of reference 3 was used for measuring sleeve motion. This allowed greater detail in the friction records when the diaphragm system was made 30 times as stiff as that of reference 1 and 2. The measured natural frequency of this system was calculated to be about 1100 cycles per second, while the spring constant was about 815,000 pounds per inch.

The electromagnetic pickup (See Fig. IV) was connected to an impedance bridge circuit which was supplied with a carrier voltage of 5,000 cycles per second. The output from the bridge, which was adjusted in both amplitude and phase to an almost perfect balance, was amplified and applied to the Y-axis deflection of a DuMont type 208 oscilloscope. During the engine operation, the motion of the cylinder sleeve changed the position of the armature with respect to the pickup and thus changed the balance of the bridge circuit. The edge of the resulting modulated carrier wave was centered on the oscilloscope screen and was photographed for a permanent record of the sleeve motion. Such photographs were made by turning off the X-axis sweep of the oscilloscope and projecting the image of the trace onto a film which moved at right angles to the trace motion at a known speed (25 inches per second). Figure V shows a photograph of the electrical measuring equipment.

Adequate sensitivity was provided by the electromagnetic pickup with a modified laminated armature. The system was calibrated by loading the sleeve diaphragm assembly with test weights and recording the trace deflection as observed on the oscilloscope. A celluloid grid was placed against the tube face; the horizontal lines which appeared on the photographic records established a convenient force scale which was helpful in interpreting the results. The top deadcenter position of the engine crankshaft was established by a neon light which flashed once per revolution of the engine crankshaft. Light from the lamp illuminated a slit, and an image of the slit was focused on the back of the film as it passed through the film gate in the camera.

Fuel-air mixture was supplied to the engine from a steam jacketed vaporizing tank, and the exhaust gases were cooled in a surge tank before passing into the laboratory exhaust system. Inlet and exhaust pressures were measured in these tanks. Temperature were measured with mercury-in-glass thermometers with the exception of the lower crankcase oil supply, which was measured by a vapor pressure thermometer. All accessory fuel, water, and oil pumps were electric driven. Engine torques were measured by means of a cradled electric dynamometer and hydraulic scale. Figure VI shows a photograph of the friction engine on the test stand.

Precision of Friction Measurement

Static calibration at 180°F jacket temperature of the friction measuring apparatus was made before each run with weights from 10 to 50 lbs. Deflections were observed on the oscilloscope grid to the closest 0.05", and are plotted in Figs. VII and VIII. The curve was assumed linear within the expected range of frictional forces. A straight line approximation of the relationship is shown in the calibration plot. See Figs. VII and VIII.

Evidently the new laminated armature which replaced the powdered iron piece used in previous years, provides a reliable magnetic pick-up, once it has been properly adjusted. The overall sensitivity has not been changed. In order to account for the anticipated friction of 3/16" rings, a lower gain setting on the oscilloscope was used so that the trace would not come too close to the edge of the screen.

Several early runs were discarded due to sticking of the junk rings after a short period of firing. This difficulty was rectified by insuring a steady flow of oil (2cc/min) and keeping the head circulating water temperature (T_h) at 65°F.

Theoretical Considerations

The friction of two surfaces sliding over each other, such as piston rings over a cylinder liner, is usually differentiated as dry, fluid, and mixed friction. In analyzing the friction of piston rings, it is considered that the friction is of all three natures because of the various conditions that exist during the cycle--that is, extreme variation is gas pressures from one to about forty atmospheres, variation in relative velocity, variation in distribution of oil over the bearing surfaces, presence of impurities and foreign matter in the lubricant, and roughness of the surfaces.

Examining the characteristics of the various types of friction, it is seen that with dry friction the friction is directly proportional to the pressure between the two surfaces and independent of the relative sliding velocity and area of contact. With fluid friction, in addition to the effect of viscosity and thickness of the lubricant and the form of the surfaces in contact, the frictional resistance varies as the area of contact and the relative sliding velocity, normal pressure remaining constant. At present, mixed friction defines the zone between fully fluid and dry friction. Hydrodynamic theory breaks down here because the extent of the oil film is not known.

In general, the variation of piston ring characteristics might have the following effect on piston ring friction:

1. Increase of ring tension would increase friction of the dry type and decrease the fluid type of friction.

2. Increase of face width (tension remaining the same) would decrease the dry friction and increase the fluid friction.

3. Decreasing the number of piston rings used would decrease the friction of both types.

4. An interrupted surface ring would have more friction than a similar lapped surface ring because it would break up the lubricant film and produce the higher frictional coefficients of dry friction. Reference 7 indicates that an interrupted surface would produce a better film in the long run because of the entrained oil in the grooved surface, and thus have comparable friction characteristics to a lapped surface ring.

The effect of mixed friction on the above is generally impossible to predict because of the vague knowledge of the mechanics of this type. The relative magnitude of the various types of friction under engine operating conditions is unknown.

RESULTS

1. Ring tension had little effect on total piston ring friction. Differences in piston ring mean-effective pressure due to ring tension indicates slightly higher ring friction for rings of maximum feasible tension.
2. Ring face width had but a small effect on piston ring friction. The two runs with three 3/16 inch rings indicated slightly increasing friction over the period of the test as opposed to a decrease in friction with the three 1/16 inch rings.
3. Reducing the number of piston rings from three to one reduced the friction mean-effective pressure by about 25% in the case of the 3/16 inch rings, but in the case of the 1/16 inch rings the friction remained nearly the same.
4. The 3/16 inch interrupted surface rings indicated very high friction at the beginning of the run, but as the run-in time increased the friction continued to decrease to a level equivalent to that of the 3/16 inch lapped surface rings.
5. Blow-by for the single 1/16 inch rings was nearly twice as great as that for the runs with the three 1/16 inch rings. High tension reduced blow-by in the wide rings, but tension appeared to have no effect on blow-by past the narrow rings.

DISCUSSION OF THE RESULTS

Effects of Ring Tension.

In general it can be said that the changes in ring tension produced no significant changes in the total friction work. The band covering the results for the rings of the highest total tension was slightly above the corresponding band for the rings of lowest total tension. When divided into components, FMEP during the compression-power strokes (CP-FMEP) did not noticeably change with tension whereas FMEP during the exhaust-intake strokes (EI-FMEP) showed a slight increase with increased tension.

Tischbein noted in his experiments that the friction coefficient went up as the wall pressure increased, and he concluded that there was a smaller proportion of fully fluid friction involved since the fluid friction would theoretically decrease with increasing wall pressure. However, it should be noted that these results were obtained at low gas pressures, i.e. approximately atmospheric conditions.

Considering that the top ring bears the brunt of the gas pressure and contributes most of the friction as seen in the results, it is noted that removal of the two bottom, light tension, narrow rings hardly effects the piston ring friction while the removal of the two bottom, heavy tension, wide rings caused a noticeable drop in the friction work. This effect of total tension is further confirmed by the difference between the total FMEP for runs C and C' and the difference between the total FMEP for runs D and D'. It might be considered then that the two bottom rings show the effect of ring tension which fact would be in accordance with the results of Tischbein since they are under lower gas pressures.

Effect of Ring Face Width

The runs with three 1/16 inch runs appeared to have a decreasing trend noted when three 3/16 inch piston rings were used. The magnitude of the friction work in each case was nearly the same. When the friction work was broken down into compression-power and exhaust-intake components no significant trends were noted within the band of the plotted datum. This is especially noticeable in the exhaust-intake component where the band width of the plotted points is often fifty percent of the maximum values observed. When dealing with such small magnitudes of friction such dispersion of results is unavoidable. There is hardly any noticeable effect of ring face width on piston ring friction.

Effect of Change in Number of Rings

As mentioned under "Effects of Ring Tension", the top ring apparently contributes the major portion of the friction work during the test runs. Further by breaking the friction into its components, it is easily seen that most of the friction occurs during the compression-power strokes. This is attributable to the high gas pressures to which the top ring is subjected. During the exhaust-intake strokes where the gas pressures were near atmospheric, friction was always light, and EI-FMEP was observed in most cases to be less than one pound per square inch. Here the friction was relatively independent of the number of rings.

The greatest change in friction work due to change in number of rings was noted after run D when the change of total ring tension was from about 30 pounds to 10 pounds. The least change in friction work due to a change in number of rings was noted after Run A, where the change of total ring tension was from about 3 pounds to one pound. The change in tension after Runs B, and C was about seven pounds, and here the drop in FMEP was perceptible but not as great as after Run D. Thus it might be said that the change in total wall pressure is the predominant factor causing the change in FMEP.

Performance of the Interrupted Surface Rings

In order to see if there had been any perceptible wear, photomicrographs were taken of a 3/16" Interrupted Surface Ring before and after approximately five hours of running under firing conditions. Wear was not perceptible except so far as polishing of the lands which was pronounced. (See Figs. X and XI). As expected, FMEP in Run E was very high for the first hour of run-in probably due to ring surface roughness breaking the oil film, but after several hours the FMEP dropped steadily and approached the FMEP observed with the other 3/16" high tension rings (Run D). The striking difference, however, was the slightly increasing trend of FMEP with time in Run D, whereas in Run E the FMEP seemed to decrease with time.

Although the three Interrupted Surface rings show equivalent or possibly better performance after run-in as compared to the three (3) lapped surface rings (same tension) there is an inconsistency when the condition of only one ring is compared.

Run D' indicates lower FMEP than Run E'. This would seem to show greater contact pressure, with more dry and mixed friction on the lands of the Interrupted Surface ring. However, the step-functions photographed near T.D.C. were not so pronounced in Run E' as they were in Run D' (see Figs. XXIII and XXIV). This phenomenon may have been caused by either the measured difference in BMEP (Run D' averaged 10 psi higher than Run E') or by the entrained oil effect mentioned previously. Variation in vertical clearances between groove and ring, and accumulation of carbon deposits may have also been contributing factors. The fact that the run-in Interrupted Surface top ring, however, appears to have had <u>less dry friction</u> than the run-in Lapped Surface top ring is significant, since piston ring wear is related to magnitude of dry friction and time in contact.

DISCUSSION

Variation in Piston Ring Blow-by.

With the narrow faced rings, tension did not seem to effect the blow-by, while reducing the number of rings increased the blow-by appreciably. This might be seen as an increased restriction to gas flow due to the number of rings. The ring gap through which part of the blow-by takes place was practically the same for all runs. Another factor which might possibly effect blow-by is the angular position of the various rings in their respective grooves, thus if all the ring gaps were lined up vertically on the piston, one might expect more blow-by than if the ring gaps were widely separated. The angular position of the ring was not observed. (See Figs. XVII).

With the wide-faced rings the effect of tension was not as consistent. Runs C, C', and D' were in the same range while run D was markedly lower. One logically might say this was due to the extreme change in wall pressure due to the ring tension since the change of number of rings did not appreciably effect blow-by in Run C and C'. (See Fig. XVIII).

The three interrupted surface rings had a blow-by slightly above the corresponding three lapped surface rings of similar tension. Here again the unknown variable of ring gap position as well as the nature of the surface might have played a part. (See Fig. XIX).

Remarks as to the Effects of Circularity

Diametral bore gauge readings at TDC before the test runs were started in March 1951 measured 3.251 inches on the electromagnetic pick-up diameter and 3.253 inches at 90° to the above. This could mean 0.001 inch wear and 0.002 inch distortion from the true 3.250 inch diameter. Bore taper was less than 0.0005 inch. After completing Runs A, B, C, and D and with four lapping operations, there was no perceptible departure from the initial measurements. The surface hardness of the liner, of course, is much higher than that of the rings used.

The 0.002 inch deviation from circularity might show up as an effect on FMEP. It would probably be more noticeable in the case of the wide stiff rings than in the case of the narrow flexible rings since the former would experience more difficulty in conforming to the true shape of the liner. Furthermore, it supposedly would be more pronounced at TDC where gas pressures were highest. Rotation of the rings was not observed; however, if it occurred, variation in local wall pressures might conceivably result in dispersion of the observed FMEP values.

CONCLUSIONS

(1) Maximum feasible tensions caused a barely detectable increase in total piston ring friction work.

(2) Wide rings showed only slightly greater friction than did the narrow rings.

(3) Reducing the number of piston rings from three (3) to one (1) caused a reduction in FMEP by about 25% in the case of the 3/16" rings, but in the case of the 1/16" rings, friction remained nearly constant.

(4) The 3/16" interrupted surface ring indicated values of FMEP after run-in comparable to that of the similar lapped surface ring.

RECOMMENDATIONS

(1) Further study of ring size, profile and number of rings is recommended.

(2) Results indicate that run-in time for wide rings should be greater than three (3) hours used to run-in the narrow 1/16" rings.

(3) Further study of Ring Tension is not considered to be warranted.

(4) Further study of friction work with interrupted surfaces is recommended.

(5) Consider the possibility of increasing piston speed so that ring variable effects would be more pronounced.

(6) Consider the possibility of building a similar friction engine of large size, where the absolute value of FMEP would be greater, and possibly the error and dispersion would be relatively smaller.

(7) Study the effect on the photographic record of large variation in vertical clearance between top ring and groove.

APPENDIX A

Summary of Data and Calculations

Appendix A

Summary of Data and Calculations

Sheet A-1

Date	Photo No.	Rel. Time	CP Loop Area	EI Loop Area	Total Area	Conv. Factor	CP-FMEP	EI-FMEP	Total FMEP	BMEP	CP-FMEP / BMEP	EI-FMEP / Total FMEP	
Units	—	Min.	IN^2	IN^2	IN^2	—	PSI	PSI	PSI	PSI	—	—	
4-24-51	A,1	30	*7.47	*1.05	*8.52	.284	4.24	0.60	4.84	76.0	.056	.124	3 Narrow Rings Low Tension
	A,2	60	14.39	0.98	15.37		4.08	0.28	4.36	78.6	.052	.064	
	A,3	105	13.10	0.88	13.98		3.72	0.25	3.97	79.5	.047	.063	
	A,4	125	12.62	0.88	13.50		3.59	0.25	3.84	79.0	.045	.065	
	A,5	140	12.12	0.83	12.95		3.44	0.24	3.68	80.3	.043	.065	
	A,6	155	12.59	0.52	13.11		3.57	0.15	3.73	80.3	.044	.040	
	A,7	170	3.31	1.78	5.09		0.94	0.50	1.45	M	—	.345	
	A',1	225	10.78	2.21	12.99	.284	3.06	0.63	3.69	80.4	.038	.171	1 Narrow Ring Low Tension
	A',2	235	10.14	2.51	12.65		2.88	0.71	3.59	80.8	.036	.198	
	A',3	245	8.86	2.49	11.35		2.52	0.71	3.22	81.6	.031	.220	
	A',4	255		Record	Spoiled		—	—	—	82.0	—	—	
	A',5	263	10.85	2.03	12.88		3.08	0.58	3.66	83.7	.037	.158	
	A',6	273	2.40	0.37	2.77		0.68	0.10	0.78	M	—	.128	
4-27-51	B,1	30	14.03	2.81	16.84	.284	4.02	0.79	4.79	68.0	.057	.165	3 Narrow Rings High Tension
	B,2	50	6.47	1.20	7.67	.570	3.69	0.68	4.37	70.2	.053	.155	
	B,3	70	6.66	1.04	7.70		3.80	0.59	4.39	66.1	.057	.135	
	B,4	90	7.67	0.87	8.54		4.37	0.50	4.86	66.6	.065	.103	
	B,5	120	7.01	0.99	8.00		4.00	0.56	4.56	67.6	.059	.123	
	B,6	140	6.14	0.77	6.91		3.50	0.44	3.94	69.3	.050	.111	
	B,7	180	5.10	0.67	5.77		2.91	0.38	3.29	70.5	.041	.115	
	B,8	195	6.02	1.74	7.76		3.43	0.99	4.42	79.1	.043	.224	
	B,9	202	5.99	4.37	10.36	.284	1.70	1.24	2.94	M	—	.422	
	B',1	275	10.08	0.64	10.72	.284	2.86	0.18	3.05	81.6	.035	.059	1. Narrow Ring High Tension
	B',2	295	10.46	1.73	12.19		2.97	0.49	3.46	81.6	.036	.141	
	B',3	305	10.18	1.33	11.51		2.89	0.38	3.27	81.1	.035	.116	
	B',4	316	9.88	1.24	11.12		2.81	0.35	3.16	81.6	.034	.111	
	B',5	335	8.56	3.52	12.08		2.43	1.00	3.43	82.5	.029	.292	
	B',6	345	1.90	0.49	2.39		0.54	0.14	0.68	M	—	.206	
4-17-51	C1	40	5.28	0.61	5.89	.570	3.01	0.35	3.36	71.8	.042	.114	3 Wide Rings Low Tension
	C2	70	5.43	0.97	6.40		3.10	0.55	3.65	75.5	.041	.151	
	C3	92	5.71	1.38	7.09		3.26	0.79	4.05	78.2	.042	.196	
	C4	130	6.18	1.15	7.33		3.52	0.65	4.17	80.3	.044	.155	
	C5	160	7.28	0.61	7.89		4.15	0.35	4.50	81.6	.050	.078	
	C',1	250	8.77	1.78	10.55	.284	2.49	0.51	3.00	81.2	.031	.170	1 Wide Ring Low Tension
	C',2	270	7.51	1.43	8.94		2.13	0.41	2.54	82.8	.026	.161	
	C',3	280	9.25	2.48	11.73		2.63	0.70	3.33	81.6	.032	.210	
	C',4	293	10.10	2.15	12.25		2.87	0.61	3.48	83.7	.034	.175	
	C',5	305	10.04	3.03	13.07		2.85	0.68	3.71	84.6	.034	.230	

* Preamp. gain setting low; mult. by factor of 2.

W.C.G. F.A.P 5-9-51

Appendix A

Summary of Data and Calculations

Sheet A-2

Date	Photo. No.	Rel. Time	CP Loop Area	EI-Loop Area	Total Area	Conv. Factor	CP-FMEP	EI-FMEP	Total FMEP	BMEP	CP-FMEP / BMEP	EI-FMEP / Total FMEP	Remarks
Units	—	MIN	IN^2	IN^2	IN^2	—	PSI	PSI	PSI	PSI	—	—	
4-20-51	D1	25	5.57	1.30	6.87	.570	3.18	0.74	3.92	75.2	.042	.189	3 Wide Rings High Tension
	D2	55	5.59	1.34	6.93	↓	3.19	0.76	3.95	76.8	.041	.193	
	D3	80	6.23	1.28	7.51	↓	3.55	0.73	4.28	76.8	.046	.170	
	D4	110	7.36	1.23	8.59	↓	4.20	0.70	4.89	77.2	.054	.140	
	D5	135	6.78	1.63	8.41	↓	3.86	0.93	4.80	77.8	.050	.194	
	D'1	210	8.60	0.81	9.41	.284	2.44	0.23	2.67	78.6	.031	.086	1 Wide Ring High tension
	D'2	225	7.70	1.04	8.74	↓	2.19	0.30	2.48	79.7	.027	.121	
	D'3	240	8.40	0.75	9.15	↓	2.38	0.21	2.60	79.7	.030	.081	
	D'4	250	7.97	0.68	8.65	↓	2.26	0.19	2.46	80.4	.028	.077	
	D'5	275	7.63	1.02	8.65	↓	2.17	0.29	2.46	81.6	.026	.118	
	D'6	285	7.57	0.88	8.45	↓	2.15	0.25	2.40	82.0	.026	.104	
4-28-51	E1	-5	7.09	4.91	12.00	.570	4.04	2.80	6.84	M	—	.410	3 Wide Interrupted Surface Rings High tension
	E2	+20	13.70	3.73	17.43	↓	7.81	2.13	9.95	60.7	.129	.214	
	E3	75	10.10	1.58	11.68	↓	5.76	0.90	6.66	64.6	.089	.135	
	E4	120	8.99	0.97	9.96	↓	5.12	0.55	5.67	70.2	.073	.097	
	E5	150	8.12	0.85	8.97	↓	4.63	0.48	5.12	63.8	.072	.094	
	E6	185	6.26	1.27	7.53	↓	3.57	0.72	4.29	67.1	.055	.168	
	E7	205	6.35	0.86	7.21	↓	3.62	0.49	4.11	68.9	.053	.119	
	E8	215	2.52	1.73	4.25	↓	1.44	0.99	2.42	M		.410	
	E'1	275	6.04	0.68	6.72	.570	3.44	0.39	3.83	67.1	.051	.102	1 Wide Interrupted Surface Ring High Tension
	E'2	290	6.73	0.68	7.41	↓	3.84	0.39	4.23	69.2	.055	.092	
	E'3	305	5.22	0.74	5.96	↓	2.97	0.42	3.40	69.7	.043	.123	
	E'4	335	6.61	1.25	7.86	↓	3.77	0.72	4.48	70.2	.054	.158	
	E'5	350	0.90	0.57	1.47	↓	0.51	0.32	0.84	M	—	.382	
			IN.lbs	IN.lbs	IN.lbs								
3-30-51	B'1	250	91.5	33.3	124.8	—	2.13	1.21	3.34	78.7	.027	.362	Early run with 1 narrow ring High Tension
	B'2	315		SPOILED			—	—	—	80.3	—		
	B'3	335	92.8	9.0	101.8	.268	2.50	0.24	2.74	80.3	.031	.088	
	B'4	345	95.0	9.3	104.3	.268	2.53	0.27	2.80	80.7	.031	.096	
	B'5	380	107.9	10.3	118.2	.268	2.87	0.30	3.17	81.6	.035	.095	

F.A.P. W.C.G.
5-10-51

APPENDIX B

Sample of Calculation Procedure

APPENDIX B

Sample of Calculation Procedure

Step 1. The developed film was placed in an enlarger such that the enlarged distance between the TDC marks was exactly 9 inches or twice the stroke of the engine. A tracing of the wave envelope and grid was made on a piece of 8 1/2 x 11 paper--one of the compression-power strokes and one of the exhaust-intake strokes. After the tracing was made, the result was compared with the balance of the cycles on the film. This was to insure that the envelope traced was representative of the particular run.

Step 2. The tracings were placed in the MIT transfer machine (Reference 2) where the coordinates were changed from Force versus crank angle to Force versus volume and the work loops were formed.

Step 3. Each work loop was then integrated by use of a planimeter. $W = \int F ds$. For example, take the compression-power work loop of run B_1' --planimeter readings start 7948

start	7948	
		-- 1020
	8966	
		-- 1016
	9982	
		-- 1018
finish	10002	
Average		1018

planimeter constant: 100 units = 1 sq. in.

work loop area = 10.18 sq. in.

From enlarger, 1.35 inches was equivalent to one inch on the oscilloscope.

From the transfer machine, stroke was 4.5 inches.

From the oscilloscope calibration plot (see Fig. VII), slope of graph was 15.9 pounds per inch of deflection.

Thus the work of the loop is

$$W = 10.18 \times \frac{1}{1.35} \times \frac{4.5}{5} \times 15.9 = 108 \text{ in. lb.}$$

$$FMEP = \frac{W}{V_d}$$

V_d friction engine = 37.3 cu. in.

$$CP\text{-}FMEP = \frac{108}{37.3} = 2.89 \text{ psi}$$

Similarly for the EI-FMEP of run B13 is

EI-FMEP = 0.38 psi

total FMEP = 3.27 psi

$$\text{Conv. Factor} = \frac{FMEP}{AREA} = \frac{2.89}{10.18} = 0.284$$

AIR FLOW

Measured by ASME standard square edged orifice (Diameter .515")

Simplified formula ($T_1 = 75°F \pm 3°F$)

$$\dot{M}_a = A = .00079\sqrt{p_1 h}$$

Reference: M.I.T. Notes on Air Flow by W. A. Leary and D. Tsai.

BRAKE MEAN EFFECTIVE PRESSURE

$$BMEP = \frac{792000 \times B.L.}{KV}$$

K = Hydraulic Dyn. Constant = 5000
V_d = displ. Volume = 37.3 cu. in.

BMEP = 4.25 BMEP

$$BHP = \frac{N \times B.L.}{K} = \frac{1000}{5000} h = 0.20 \times B.L.$$

Reference: W.A. Leary Scale Data Computations 5/46
Dynamometer No. 12.
R = 12.6050 in.
d_c = 1.6135 in. (computed)
d_a = 1.6060 in. (actual)
x = 1 lb. per. in.

APPENDIX C

Original Data

SHEET C-1

EXPERIMENT NO. A-A'₁ **TITLE** 1/16" LOW TENSION RINGS **DATE** APRIL 24, 1951 **SLOAN LABORATORY**

ENGINE CFR-FRICTION **FUEL** MARINE WHITE **S.G.** ____ **WET BULB** ____ **DRY BULB** 77° F

BORE 3 1/4 **STROKE** 4 1/2 **COMPRESSION RATIO** 5.05 **BAROMETER (ACT.)** ____ **(CORR.)** 30.38 (A) 30.30 (A'₁)

CONSTANTS BMEP = B.L. X 4.25 BHP = $\frac{\text{B.L.} \times \text{RPM}}{5000}$ = 0.20 x B.L. DYN. CORR. ZERO

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P_1	P_E	T_1	AIR CONS.	FUEL CONS.	F/A	S.A.	R	h	BMEP	Pi	Ti	Th	SL	δc	δL		Pi	Pc	Pi
COMMENCED MOTORING	1140	A₁	1000					50																				
" FIRING	1145			16.5							77				23°					146		.970	2.0			29.48	30.28	30.28
	1200			17.5		150	176		-0.10	-0.10	77	.01038	.00078	.0753		8.20	5.70	74.4	-0.90	146	65	.970	2.0	2.5				
(PREAMP. GAIN INCRE.) PHOTO. A₁1	1215			17.9		150	181				77	.01023	.00080	.0781		8.30	5.55	76.0		151		.970	1.7	3.0				
	1230			18.4		151	180				77	.01023	.00080	.0781			5.55	78.2		148		.968	2.3	2.7				
PHOTO. A, 2	1245			18.5		149	180				77	.01023		.0781			5.55	78.6		149		.968	3.5	2.5				
BAD INTERFERENCE ON SCOPE	1320			18.7		151	180				77	.01027		.0778			5.60	79.5		150		.970	2.0	2.9				
PHOTO. A, 3	1330			18.7		150	181				77	.01027		.0778			5.60	79.5		149		.970	1.9	3.4				
	1345			18.5		150	180				78	.01027		.0778			5.60	78.6		151		.970	1.8	2.9				
PHOTO A, 4	1350			18.6		151	179				78	.01027		.0778			5.60	79.0		151		.970	1.8	3.0				
PHOTO A, 5	1405			18.9		151	180				78	.01038		.0771			5.70	80.3		149		.970	1.8	3.0				
PHOTO. A, 6	1420			18.9		150	180				78	.01027		.0778			5.60	80.3		150	64	.970	1.7	2.9				
CEASED FIRING	1530																					.969						
MOTORING PHOTO A, 7	1535		1000	-		152	180		-0.1	-0.1	78	.01046	—	—			5.8	—	-0.90	150	66	.969	2.0					
SECURED ENG. REMOVED #2 AND #3 RINGS.	1537															OIL PAST JUNK RINGS TO BEAKER: 347 grams.												
TIME: 3ʰ 57ᵐ																FLUID IN BLOWBY FLASK							2 g.					
COMMENCED MOTORING	1505	A'₁	1000					50																				
COMMENCED FIRING	1510		1000	18.0					-0.1	-0.1	77				23°				-0.90			.969	2.0					
	1520			18.5		145	181				77	.01027	.00080	.0780		8.30	5.65	78.6		148	65	.969	2.0	6.3		29.40	30.20	30.20
[redacted] PHOTO A'₁ 1	1530			18.9		145	178				77	.01022	.00079	.0792		8.25	5.55	80.4		150		.969	2.0	6.2				
PHOTO A'₁ 2	1540			19.0		153	180				77	.01022	.00079	.0772		8.25	5.55	80.8		151		.969	1.5	6.7				
PHOTO A'₁ 3	1550			19.2		153	181				77	.01022	.00080	.0781		8.30	5.55	81.6		151		.967	2.2	5.5				
PHOTO A'₁ 4	1600			19.3		151	181				77	.01022	.00080	.0781		8.30	5.55	82.0		150		.967	2.1	5.5				
PHOTO A'₁ 5	1608			19.7		151	179				78	.01022	.00080	.0781		8.30	5.55	83.7		150		.967	2.0	—				
	1610																											
MOTORING PHOTO A'₁ 6	1618			—			~~179~~				78					—	5.9						1.8	2.3				
SECURED ENG. TIME: 1ʰ 25ᵐ	1625						179				78																	

RING DATA: TENSION GAP.
- #1 1.38 lbs .014"
- #2 1.49 .017"
- #3 1.43 .013"

OIL TO JUNK RINGS (BEAKER) 110 gms.
FLUID IN BLOWBY FLASK 15.8 gms (EXTREMELY BLACK)

SCOPE GAIN = 40 (BOTH RUNS)

F.A.P. W.C.G. 5-8-51 RUNS A - A'₁

SHEET C-2

EXPERIMENT NO. B_1, B_1' **TITLE** 1/16" HIGH TENSION RINGS **DATE** APRIL 27, 1951 **SLOAN LABORATORY**

ENGINE CFR - FRICTION **FUEL** MARINE WHITE **S.G.** ____ **WET BULB** ____ **DRY BULB** 75° F

BORE 3 1/4 **STROKE** 4 1/2 **COMPRESSION RATIO** 5.05 **BAROMETER (ACT.)** ____ **(CORR.)** 30.40

CONSTANTS BMEP = B.L. X 4.25 BHP = $\frac{B.L. \times RPM}{5000}$ = 0.20 x B.L. ABS. PRESSURES

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P_1	P_E	T_1	AIR CONS.	FUEL CONS.	F/A	S.A.	R	h	BMEP	P_i	T_i	T_h	S_L	q_c	q_b		P_i	P_e	P_1
COMMENCED MOTORING	1015	B_1		(SCOPE GAIN : 40)																								
" FIRING	1020			16.0													5.8											
PHOTO RECORD B_1-1	1050		1000	16.5		150	180	50	-0.1	-0.3	75	.01036	.0008	.0773	23°	8.3	5.7	68.0	-0.9	151	65	.970	2.0	3.0		29.5	30.1	30.3
REDUCED SCOPE GAIN TO: 35	1100																5.6											
PHOTO RECORD B_1-2	1110			15.6		152	180	50	-0.1	-0.3	74	.01020	.0008	.0784		8.3	5.5	70.2		152	66	.970	2.0	2.8			30.1	30.30
	1120			15.5		151	181		-0.1	-0.3	74	.01020	.0008	.0784		8.3	5.45	66.1		151	65	.970	2.0	3.0			30.1	30.30
PHOTO RECORD B_1-3	1130			15.7		149	181		-0.1	-0.3	73	.01014	.00078	.0769		8.2	5.55	66.6		150	65	.970	1.0	3.3			30.1	30.30
PHOTO RECORD B_1-4	1150			15.8		149	180		-.05	-0.25	73	.01025	.00079	.0770		8.25	5.55	67.1		150	65	.966	3.0	3.0			30.15	30.35
	1210			16.1		153	181		-0.1	-0.3	74	.01020	.00079	.0773		8.25	5.50	68.5		151	65	.968	2.4	2.9			30.10	30.30
PHOTO RECORD B_1-5	1220			16.3		150	180		-0.1	-0.3	74	.01020	.00079	.0773		8.25	5.50	69.3		151	65	.968	2.0	3.2			30.10	30.30
PHOTO " B_1-6	1240			16.6		149	181		-0.1	-0.20	74	.01023	.00080	.0782		8.30	5.55	70.5		149	65	.968	2.0	3.7			30.20	30.30
	1300			17.3		151	180		-0.1	-0.20	75	.01023	.00078	.0763		8.20	5.55	73.5		150	65	.968	1.7	3.3			30.20	30.30
	1310			18.3		149	180		-0.1	-0.20	75	.01023	.00078	.0763		8.20	5.55	77.6		150	64	.968	2.0	3.2			30.20	30.30
PHOTO " B_1-7	1320			18.8		148	180		-0.1	-0.2	75	.01023	.00078	.0763		8.20	5.55	79.9		150	64	.968	2.0	3.7			30.20	30.30
PHOTO " B_1-8	1335			18.8		150	180		-0.1	-0.2	75	.01029	.00080	.0780		8.3	5.60	79.9		150	64	.968	2.7	3.6			30.20	30.30
CEASED FIRING	1340			(INCREASED SCOPE GAIN TO 40)																								
PHOTO " B_1-9	1342			—		150	180	50	-0.1	-0.2	74	.01051	—	—		—	5.85	—		150	65	.968	2.0	2.4			30.20	30.30
SECURED ENG. REMOVED #2 AND 3 RINGS.	1345																											
RING TENSIONS: #1 : 3.97 lbs															OIL PAST JUNK RINGS TO BEAKER: 320 gms.													
#2 : 3.86 lbs															FLUID TO BLOWBY FLASK : 2.5 gms.													
[redacted] #3 : 3.81 lbs		B_1'		(SCOPE GAIN : 40)																								
COMMENCED MOTORING	1425		1000	18.6				50	-0.10	-0.20	74																	
" FIRING	1430																											
RING GAPS: #1 .015"	1445			19.0		150	179					.01029	.00080	.0778	23°	5.60	8.30	80.7	-0.90	144	65	.968	2.8	5.9		29.5	30.20	30.30
#2 .015"	1455			19.2		150	179					.01024	.00079	.0772		5.55	8.25	81.6		151	64	.968	2.2	5.5				
#3 .015"	1515			19.2		152	180					.01024	.00080	.0782		5.55	8.30	81.6		152	64	.968	2.2	5.6				
	1525			19.1		150	181					.01020	.00080	.0785		5.50	8.30	81.1		152	64	.968	2.0	5.8				
	1536			19.2		150	178					.01024	.00080	.0782		5.55	8.30	81.6		151	64	.968	1.6	6.0				
	1555			19.4		152	180					.01020	.00081	.0794		5.50	8.35	82.5		150	65	.966	2.1	5.1				
CEASED FIRING	1600																											
MOTORING PHOTO B_1'-6	1605		1000	—		150	181									5.75	—			150	65	.966	2.1	2.1				
	1610		1000	—		150	181									5.75	—			150	65	.966	2.1	2.1				
CEASED MOTORING	1615																											

F.A.P. W.C.G. 5-8-51 RUNS B_1-B_1'

SHEET C-3

EXPERIMENT NO. c, c' **TITLE** 3/16" LOW TENSION RINGS **DATE** APRIL 17, 1951 **SLOAN LABORATORY**

ENGINE CFR - FRICTION **FUEL** MARINE WHITE **S.G.** ______ **WET BULB** ______ **DRY BULB** 73°F

BORE 3 1/4 **STROKE** 4 1/2 **COMPRESSION RATIO** 5.05 **BAROMETER (ACT.)** ______ **(CORR.)** 29.75

CONSTANTS BMEP = B.L. X 4.25 BHP = $\frac{B.L. \times RPM}{5000}$ = 0.20 B.L. OSCILLOSCOPE GAIN : 35 (RUN C) ABS. PRESSURES

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P_1	P_E	T_1	AIR CONS.	FUEL CONS.	F/A	S.A.	R	h	BMEP	p_i	T_i	T_h	S_L	g_L	g_b		p_i	p_e	P_1
COMMENCED MOTORING	1040	C	1000					50	0.0	0.0	73				25°													29.75
" FIRING	1050																											
	1100			15.9		151	180			0.0	73	.01015	.00080	.0788		8.30	5.55	67.6	-0.9	150	70	.970	2.3			28.85	29.75	29.75
	1110			16.5		149	180			+.05	72	.01010	.00080	.0790		8.30	5.50	70.3		152	66	.972	1.6				29.70	
	1124			16.7		150	181			0.0	72	.01015	.00078	.0768		8.20	5.55	70.9		151	66	.968	2.4	5.5				
PHOTO RECORD C-1	1130			16.9		150	180			0.0	72	.01010	.00080	.0790		8.30	5.50	71.8		151	66	.971	2.0	5.3				
	1145			17.0		151	179			0.0	72	.01015	.00079	.0778		8.25	5.55	72.2		152	65	.971	1.3	5.1				
PHOTO RECORD C-2	1200			17.7		149	180			0.0	72	.01015	.00080	.0788		8.30	5.55	75.2		151	65	.969	1.9	5.3				
	1215			17.8		152	181			0.0	73	.01015	.00078	.0768		8.20	5.55	75.6		152	65	.969	2.0	6.0				
PHOTO RECORD C-3	1222			18.4		150	180			0.0	73	.01015	.00080	.0788		8.30	5.55	78.2		150	65	.969	2.5	3.8				
*	1230			18.6		149	181			+.05	73	.01020	.00080	.0784		8.30	5.60	79.0		149	65	.969	2.5	3.6			29.80	
	1245			18.5		151	180			0.0	74	.01015	.00080	.0788		8.30	5.55	78.6		150	64	.969						
PHOTO RECORD C-4	1300			18.9		150	180			0.0	74	.01015	.00080	.0788		8.30	5.55	80.3		150	64	.969	1.6	4.2				
	1315			19.0		150	178			0.0	75	.01015	.00080	.0788		8.30	5.55	80.7		150	65	.967	1.7	4.0				
PHOTO RECORD C-5	1330			19.2		150	181			0	74	.01015	.00080	.0788		8.30	5.55	81.6		150	65	.967	2.0	3.4				
	1335			19.2		152	179			0	74	.01015	.00080	.0788		8.30	5.55	81.6		150	65	.967	2.0	3.7				
SECURED ENG. REMOVED #2 + 3 RINGS.	1346																											
RING TENSIONS: #1 4.08 lbs																OIL FAST JUNK RINGS TO BEAKER :							343.5 gms.					
#2 3.97 lbs																FLUID TO BLOWBY FLASK							5.0 gms.					
#3 3.86 lbs.																				OSCILLOSCOPE GAIN :			40	(RUN C')				
[redacted]																												
COMMENCED FIRING	1440	C'	1000					50			76																	
PHOTO RECORD C'-1	1500			19.1		155	180		-.05	+.05	76	.01010	.00080	.0792	25°	8.30	5.50	81.2	-0.9	150	66	.967	1.0	6.0		28.85	29.80	29.70
" " C'-2	1520			19.5		148	181				76	.01010	.00079	.0782		8.25	5.50	82.8		151	66	.962	2.0	5.5				
" " C'-3	1530			19.2		149	181				75	.01005	.00080	.0795		8.30	5.45	81.6		150	65	.962	1.9	5.0				
" " C'-4	1543			19.7		148	180				75	.01005	.00080	.0795		8.30	5.45	83.6		151	65	.962	1.8	4.2				
" " C'-5	1550			19.9		152	180				75	.01005	.00080	.0795		8.30	5.45	84.4		150	65	.962	1.7	4.5				
SECURED ENGINE.	1600																											
RING GAPS : #1 .012"																												
#2 .011"																												
#3 .013"																												
*GAS BUBBLES IN FUEL LINE.																				F.A.P.		W.G.G. 5-8-51		RUNS C-C'				

SHEET C-4

EXPERIMENT NO. D-D' TITLE 3/16" HIGH TENSION RINGS DATE APRIL 20, 1951 SLOAN LABORATORY

ENGINE CFR - FRICTION FUEL MARINE WHITE S.G. ______ WET BULB ______ DRY BULB 74°F

BORE 3 1/4 STROKE 4 1/2 COMPRESSION RATIO 5.05 BAROMETER (ACT.) ______ (CORR.) 30.00

CONSTANTS BMEP = B.L. X 4.25 BHP = $\frac{B.L. \times RPM}{5000}$ = 0.20 x B.L. DYN. CORR ZERO SCOPE GAIN : 35 ABS. PRESS.

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P	P_E	T_1	AIR CONS.	FUEL CONS.	F/A	S.A.	R	h	BMEP	P_i	T_i	T_h	S_L	g_c	g_b		P_i	P_e	P_1
COMMENCED MOTORING	1050	D	1000					50																				
" FIRING	1055																											
PHOTO. D-1	1120			17.7		149	180		-0.1	-.20	74	.01018	.00080	.0787	25	8.3	5.55	75.2	-.90	150	65	.970	1.70	3.9		29.10	29.8	29.9
	1130			18.1		152	181			-.20	74	.01018	.00080	.0787		8.3	5.55	76.8	-.90				1.80	3.6		29.10	29.8	
	1145			18.0		145	180			-.15	74	.01013	.00078	.0770		8.2	5.50	76.3	-.90				2.10	4.0		29.10	29.85	
PHOTO. D-2	1150			18.1		150	180			-.15	73	.01013	.00080	.0789		8.3	5.50	76.8	-.85				1.80	2.8		29.15		
	1200			18.2		150	181			-.15	73	.01013	.00080	.0789		8.3	5.50	77.2	-.85				2.00	2.6		29.15		
PHOTO. D-3	1215			18.1		149	179			-.15	73	.01013	.00078	.0770		8.2	5.50	76.8	-.90				1.90	2.4		29.10		
	1230			18.2		149	180			-.15	73	.01013	.00080	.0789		8.3	5.50	77.2	-.90				1.90	1.6		29.10		
PHOTO D-4	1245			18.2		151	181			-.15	73	.01013	.00080	.0789		8.3	5.50	77.2	-.90				2.00	0.8		29.10		
PHOTO D-5	1310		↓	18.3		150	181	↓	↓	-.15	74	.01018	.00080	.0787	↓	8.3	5.55	77.8	-.90	↓	↓	↓	2.00	0.8		29.10	↓	↓
SECURED ENGINE $2^h 21^m$	1311																											
REMOVED #2 & 3 RINGS.																	OIL PAST	JUNK	RINGS	TO	BEAKER			:		288.0 gms.		
RING TENSIONS:																	FLUID	TO	BLOW-BY	FLASK				:		3.5 gms.		
#1 10.97 lbs																												
2 11.02 lbs				INCREASED		SCOPE	GAIN	TO : 40																				
3 10.92 lbs																												
COMMENCED MOTORING	1355	D'	1000					50																				
" FIRING	1400			18.0		155	180		-0.1	-.15	74	.01018	.0008	.0787	25	8.3						.970	2.0	8.6		29.10	29.85	29.9
PHOTO D'-1	1425			18.5		150	180			-.15	74	.01018	.0008	.0787		8.3	5.55	78.6	-0.90	150	68	.970	2.20	8.0				
[redacted]	1430			18.8		149	180			-.15	74	.01018	.0008	.0787			5.55	79.7			65	.967	2.40	5.3				
PHOTO D'-2	1440			18.8		150	179			-.15	74	.01018		.0787			5.55	79.7				.967	2.80	5.3				
	1445			18.9		150	181			-.15	74	.01013		.0788			5.55	80.4				.969	1.70	4.4				
PHOTO D'-3	1455			18.8		150	181			-.15	74	.01013		.0790			5.50	79.7				.969	1.50	4.1				
	1500			18.9		150	181			-.15	73	.01013		.0790			5.50	80.4				.968	2.00	4.0				
PHOTO D'-4	1505			18.9		150	180			-.15	73	.01013		.0790			5.50	80.4				.968	1.80	3.8		↓		
	1520			19.2		147	180			-.20	72	.01013		.0790			5.50	81.6				.968	2.10	3.8			29.80	
PHOTO D'-5	1530			19.2		150	181			-.20	72	.01018		.0787			5.55	81.6				.968	2.20	3.9			29.80	
PHOTO D'-6	1540		↓	19.3		149	180	↓	↓	-.20	73	.01018	↓	.0787	↓	↓	5.55	82.0	↓	↓	↓	.968	2.00	4.0		↓	29.80	↓
RING GAPS / SECURED ENG.	1545		TIME $1^h 50^m$																									
#1 .013"																												
#2 .014"																												
#3 .013"																												

F.A.P. W.G.G. 5-8-51 RUNS D-D'

SHEET C-5

EXPERIMENT NO. E-E' TITLE 3/16" INTERRUPTED SURFACE RINGS DATE APRIL 28, 1951 SLOAN LABORATORY

ENGINE CFR - FRICTION FUEL MARINE WHITE S.G. ______ WET BULB ______ DRY BULB 77°F

BORE 3 1/4 STROKE 4 1/2 COMPRESSION RATIO 5.05 BAROMETER (ACT.) ______ (CORR.) 30.30

CONSTANTS BMEP = B.L. X 4.25 BHP = $\frac{B.L. \times RPM}{5000}$ = 0.20 × B.L. DYN CORR: 1000 ÷ ZERO, 1346 ÷ +0.5" Hg SCOPE GAIN: 35

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P_1	P_E	T_1	AIR CONS.	FUEL CONS.	F/A	S.A.	R	h	BMEP	p_i	T_i	T_h	S_L	g_L	g_b		p_i	p_e	p_1
Commenced Motoring	0950	E	1000					50																				
Motoring Photo E-1	1005					140	181			-0.15	77	.01041	.0008	—	23°	8.3	5.90		-0.90	151	62	.970	2.0			29.4	30.15	
Commenced Firing	1010					150	182																					
Photo E-2	1030			14.3		150	182		-.05	-0	78	.01018		.0786			5.50	60.7	-0.90	151	65	.970	2.0	3.8			30.30	30.25
	1100			14.3		148	178			0	79	.01018		.0786			5.50	60.7		150	65	.970	1.4	4.3			30.30	
	1115			14.8		153	179			0	79	.01018		.0786			5.50	62.9		153	66	.967	2.5	4.0			30.30	
Photo E-3	1125			15.2		150	179			0	79	.01023		.0780			5.55	64.6		148	65		2.0	4.1			30.30	
Photo E-4	1210			16.5		149	181			0	79	.01023		.0780				70.2		152	65		2.0	3.4			30.30	
	1225			15.0		150	180			+.1	80			.0780				63.8		150	65		2.6	3.0			30.4	
Photo E-5	1240			15.0		150	180			+.05	80			.0780				63.8		150	64		2.0	3.1			30.35	
	1250			14.7		150	180			+.05	79			.0780				62.0		151	65		1.7	2.9			30.35	
Photo E-6	1315			15.8		150	180			+.05	78			.0780				67.1		151	65		2.5	2.9			30.35	
Photo E-7	1335			16.2		150	180			0	78			.0780				68.9		148	65		2.5	2.8			30.30	
Ceased Firing	1336		Time 3h 30m																									
Motoring Photo E-8	1345		995	—		150	179			0	78	.01045	—	—		—	5.80	—		152	66	.967		0.8			30.30	30.25
Secured Eng	1346		Time 3h 56m																									
Removed #2 + 3 Rings																			Oil past junk rings to beaker 302 gms									
Ring Tensions:		E'	p_o = 30.29 in Hg																Fluid in blow by flask 3.5 gms.									
#1 10.75 lbs																												
2 10.54																												
[redacted] 3 10.85																												
Commence Motor.	1440	E'	1000					50	-.05																			
" Firing	1443																											
Photo E'-1	1445			15.8		151	183		-.05	-.05	76	.01015	.00080	.0788	23°	8.3	5.45	67.1	-0.90	152	65	.967	—	—		29.39	30.24	30.24
	1450			16.0		150	180		-.05	-.05	76	.01010	.00078	.0787		8.2	5.40	68.0		152		.967	2.9	5.1			30.24	30.24
Photo E'-2	1500			16.3		150	180		-.05	-.05	76	.01015	.00081	.0788		8.35	5.45	67.2		152		.969	2.2	5.1			30.24	30.24
Photo E'-3	1515			16.4		150	180		-.10	-.10	76	.01012	.00080	.0784		8.30	5.45	69.7		150			2.0	5.2			30.10	30.19
	1540			16.4		150	180		-.05	-.05	75	.01015	.00079	.0788		8.25	5.45	69.7		150			2.0	5.0			30.24	30.24
Photo E'-4	1545			16.5		150	179		-.05	-.10	75	.01010	.00080	.0792		8.30	5.40	70.2		150			—	—			30.24	30.19
Ceased Firing	1550		1h-20m																									
Motoring Photo E'-5	1600					149	181			-.05	-.10	75	.01045	—	—	—	—	5.80	—		150			2.0	4.3			29.24	30.19
Ring Gaps: #1 .015"		Secured Engine																	Oil past junk rings to beaker 115 gms.									
#2 .015"																			Fluid in blow-by flask 1.5 gms.									
#3 .013"																				F.A.R.	5-8-51	W.C.G.			Runs E-E'			

APPENDIX D

Bibliography

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(6) Shaw, M. C., and Nussdorfer, T.: Visual Studies of Cylinder Lubrication, Part I The Lubrication of the Piston Skirt. NACA Wartime Report E-66 (Originally issued as NACA ARR No. E5H08, 1945).

(7) Martz, L. S.: Preliminary Report of Developments in Interrupted Surface Finishes. Proceedings, Vol. 161 of the Institution of Mechanical Engineers, 1949.

APPENDIX E

Table of Piston Ring Specifications

APPENDIX E

TABLE I

RING DIMENSIONS

SIZE	3 1/4 x 1/16	3 1/4 x 1/16	3 1/4 x 3/16	3 1/4 x 3/16	3 1/4 x 3/16
RUNS USED	A + A'	B + B'	C + C'	D + D'	E + E'
WIDTH (in)	0.0625 ± 0.00025	0.0625 ± 0.00025	0.18625 ±0.00025	0.18625±0.00025	0.18625±0.00025
WALL (in)	0.1125 ± 0.0025	0.1125 ± 0.0025	0.1125 ± 0.0025	0.1125 ± 0.0025	0.1125 ±0.0025
END CLEAR-ANCE (in)	0.014 ± 0.004	.014 ± 0.994	0.014 ± .004	.014 ± .004	.014 ± .004
DIA.TENSION (LB)	1.0 ± 0.2	3.3 ± 0.7	3.0 ± 0.6	10.0 ± 2.0	10.0 ± 2.0
FACE FINISH	Lapped FC5	Lapped FC5	Lapped FC5	Lapped FC5	Circular Grooves FB20
MATERIAL	7402 - PC"M"	7402 - PC"M"	7402-PC"M"	7402-PC"M"	7402-PC"M"
PROCESS	175-1-28-10	675-1-103-10	175-1-28-10	675-1-103-10	675-1-103-10
P.C.Dwg.No.	20000-IMP	20000-IMP	20000-IMP	20000-IMP	20000-IMP
Type	Plain Compression - Straight Face				
Mfgr.	Perfect Circle Corporation				

RING COMPOSITION

Total Carbon	3.50 - 3.80 %
Silicon	2.20 - 3.10%
Sulfur	0.19 Max. %
Phosphorous	0.15-0.40%
Manganese	0.40-0.80%
Molybdenum	0.50-0.70%
Chromium	0.20-0.40%
Copper	0.50-0.75% (according to section
Hardness, Rockwell D	40-50

TABLE I

PISTON RING SPECIFICATIONS

FIGURES

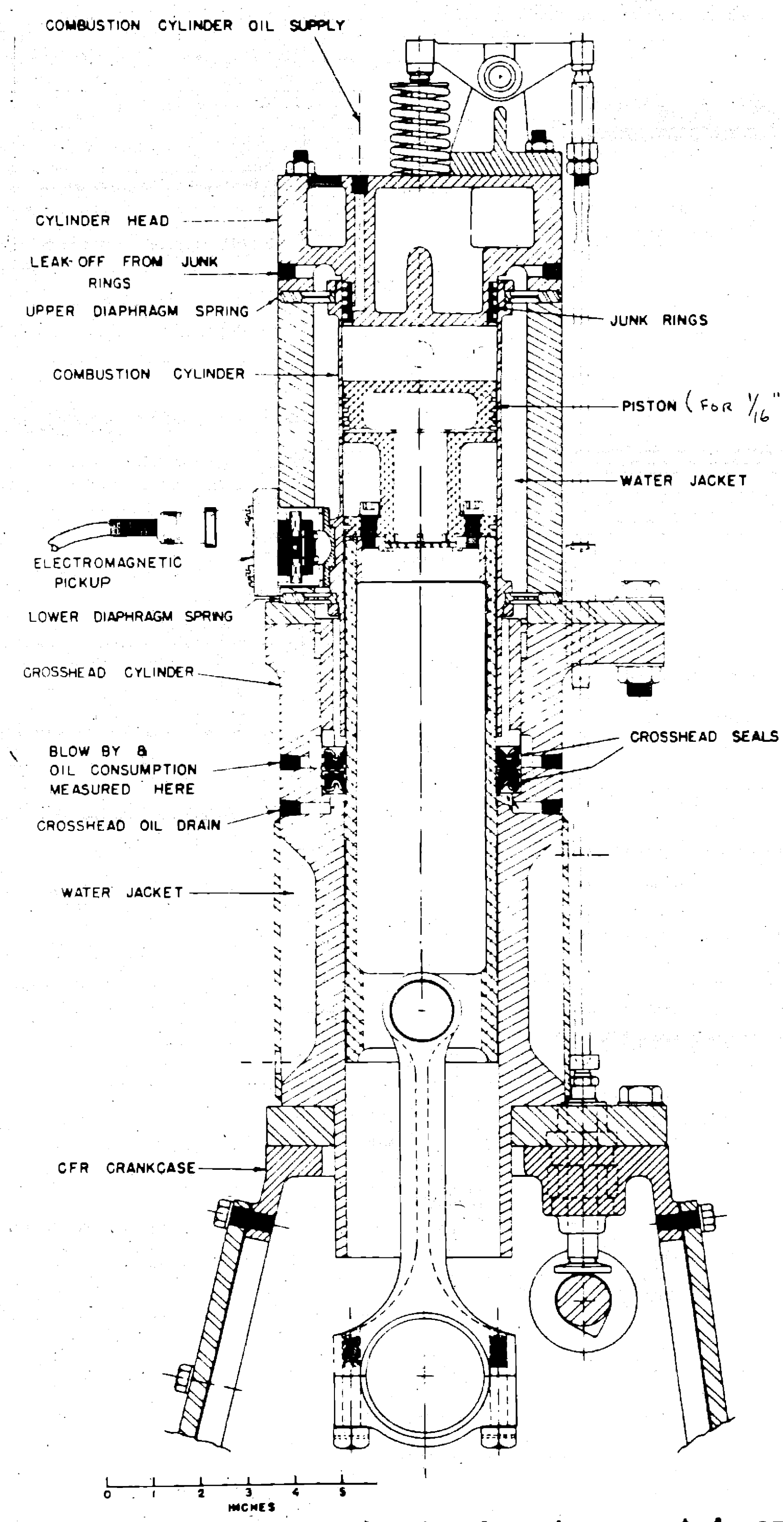


Figure I.- Engine, showing crosshead and spring-mounted combustion cylinder. PISTON FOR 1/16" RINGS. (FROM NACA TN No 1249 BY LIVENGOOD)

FIG. II

FRICTION ENGINE PISTON – 3/16"				
DRAWING NO.	SCALE	MATERIAL	DRAWN	CHECKED
FE 1-51	FULL	AL 17ST	W.C.G.	F.A.P.

MATERIAL :– ALCOA 17 ST

TOLERANCES : RING GROOVES – TENTHS OF THOUSANDTHS

ALL OTHERS – THOUSANDTHS

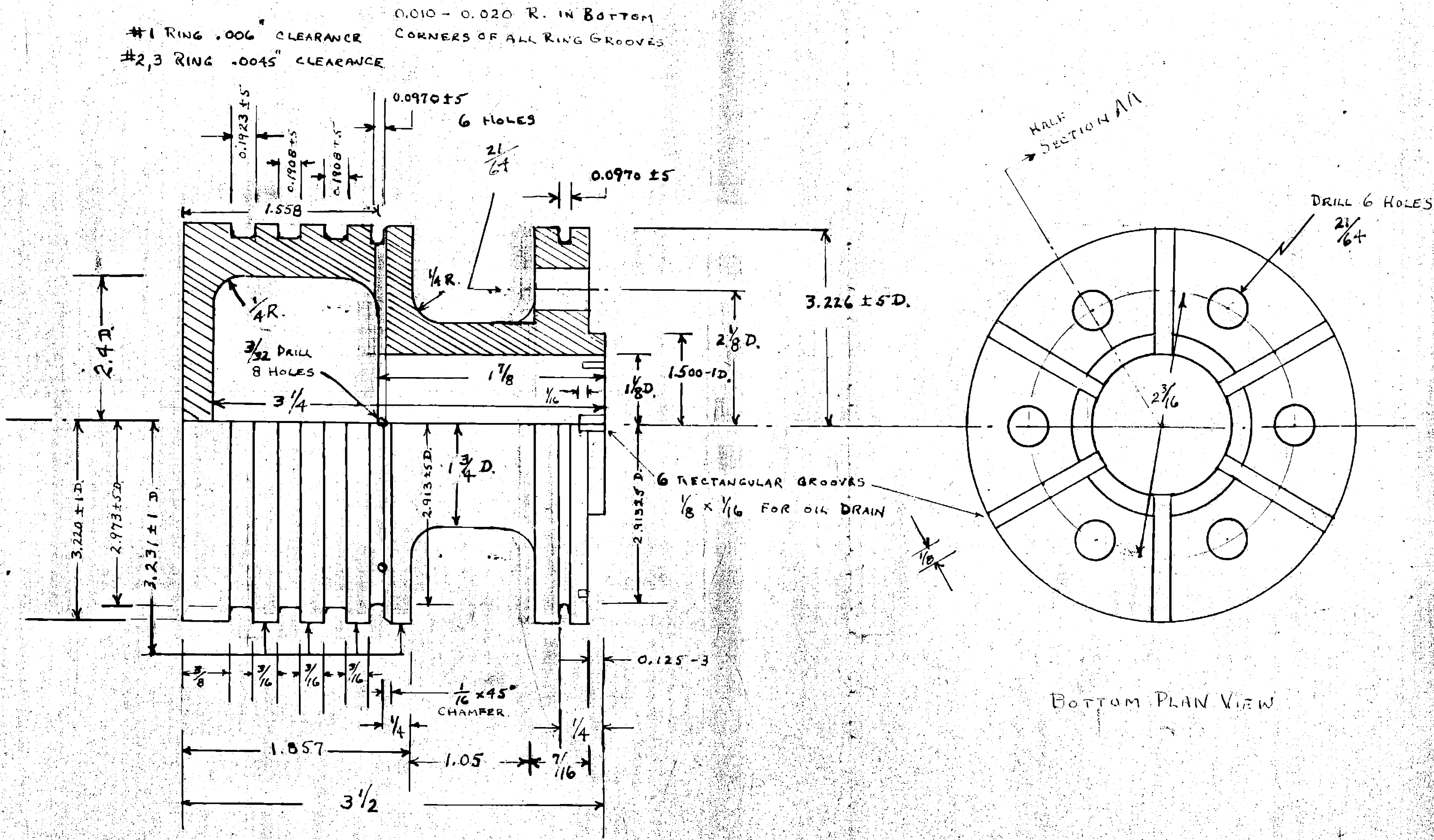


FIGURE II.

Fig. III

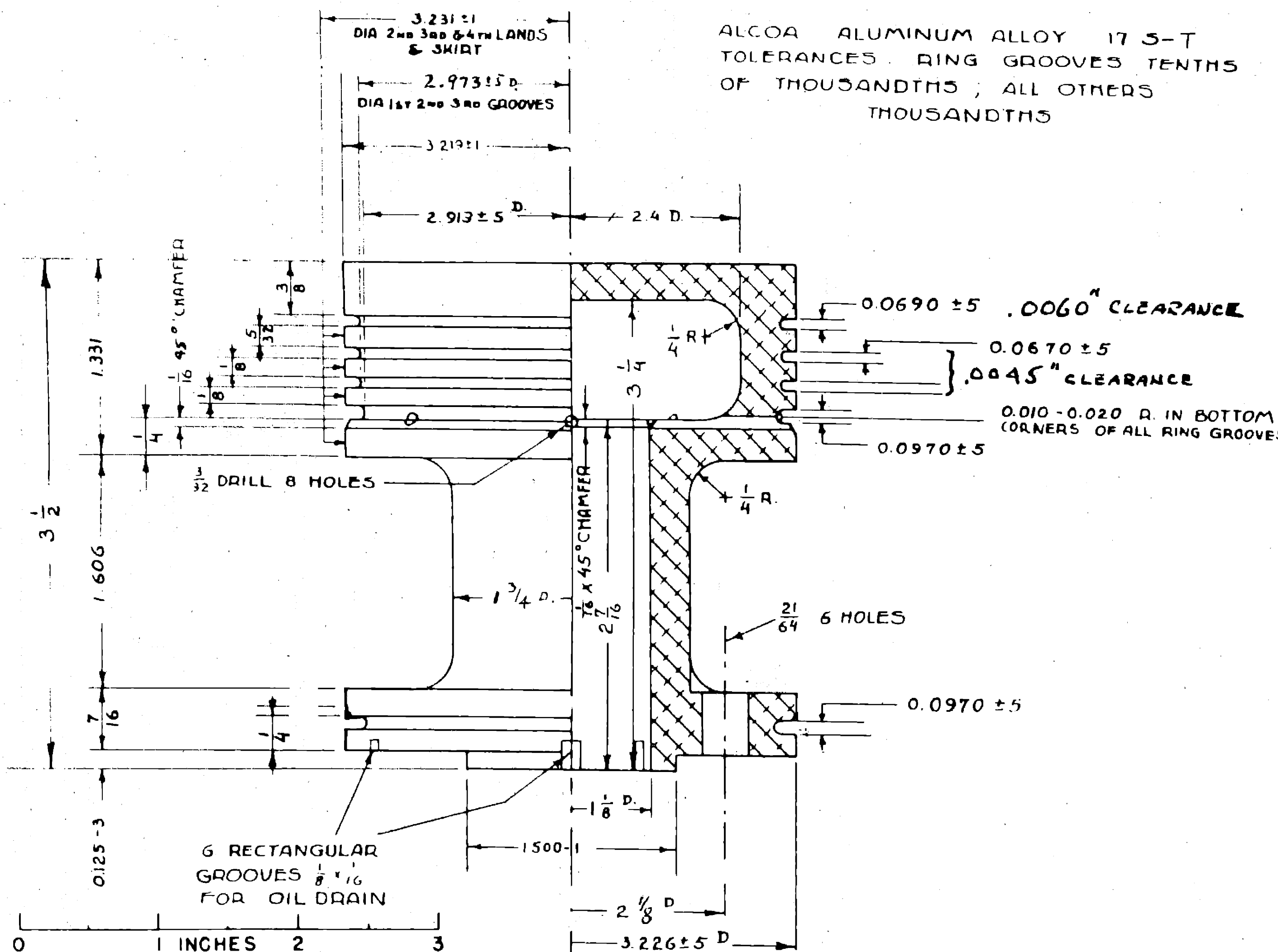


Figure III **Special piston used in the tests.**

(From NACA TN No. 1249 By Livengood)

Fig. IV

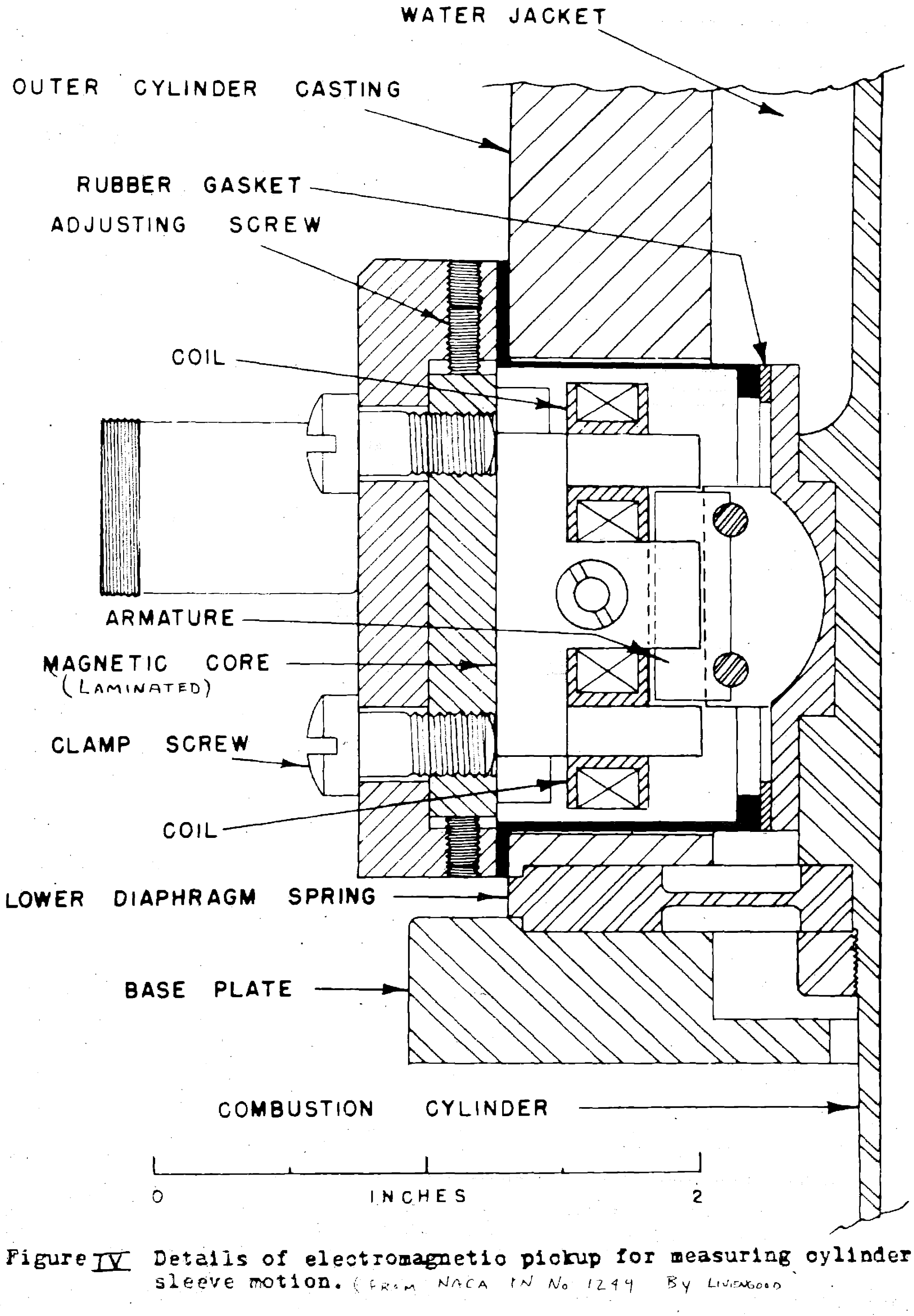


Figure IV Details of electromagnetic pickup for measuring cylinder sleeve motion. (FROM NACA TN No 1249 By LIVENGOOD

Figure V

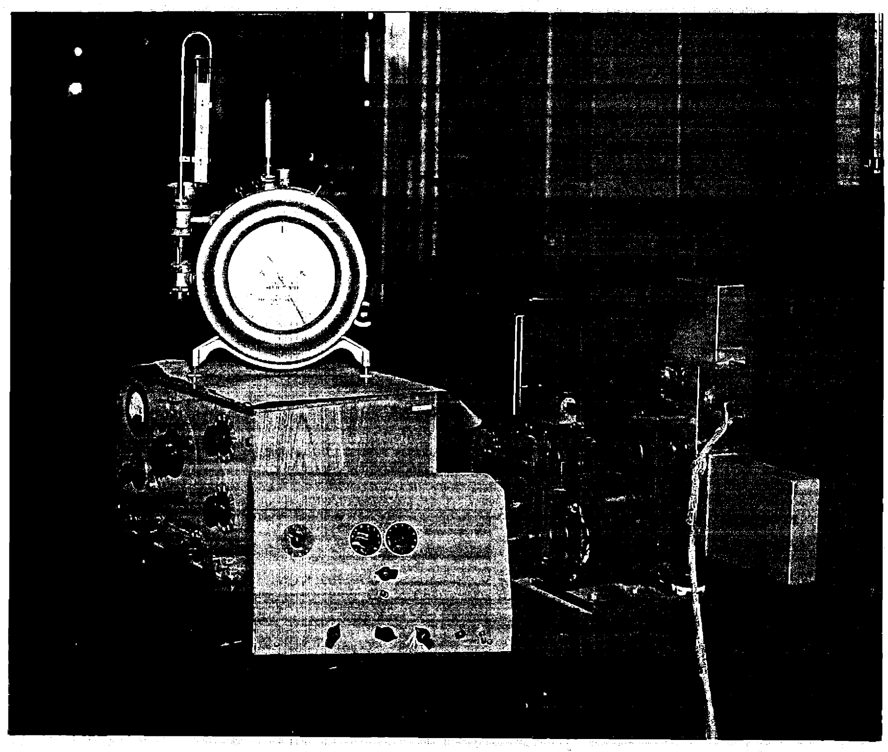
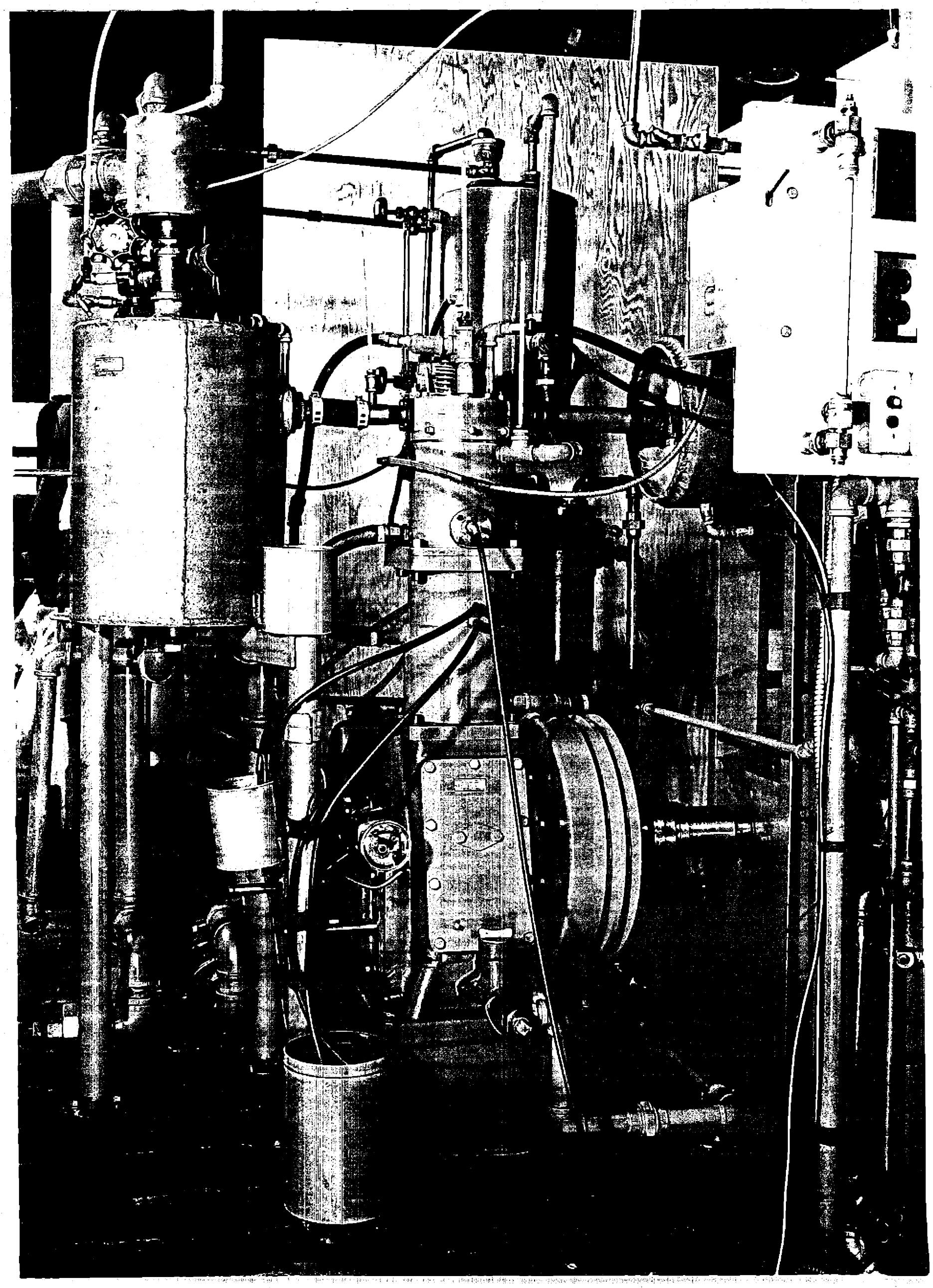


Fig. V Photograph of Measuring Equipment

Figure VI



Photograph of the Friction Engine

FIGURE VII

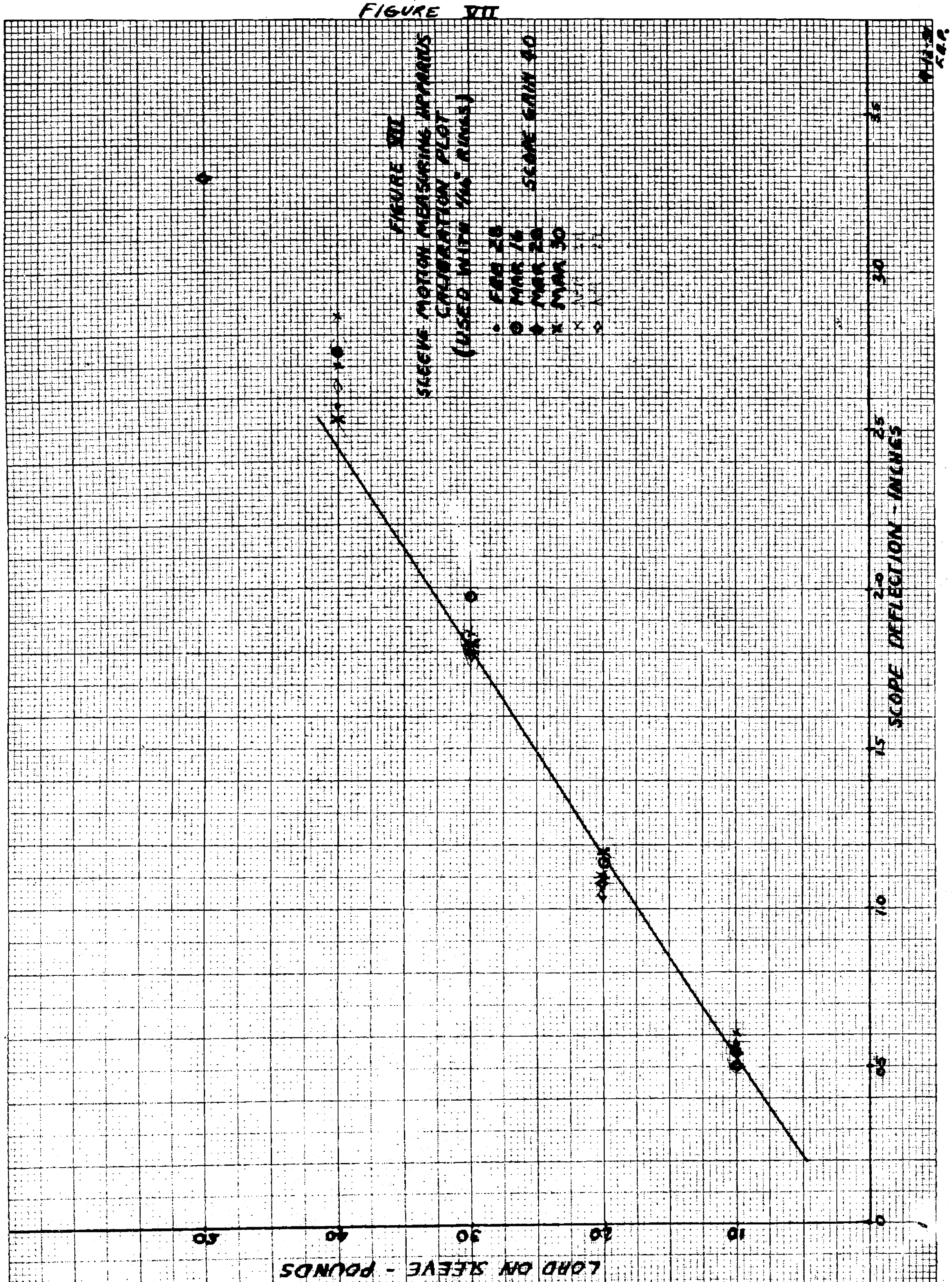


FIGURE VII
SLEEVE MOTION MEASURING APPARATUS
CALIBRATION PLOT
(USED WITH 1/16" RINGS)
FEB 28
MAR 16
MAR 28
MAR 30
SCOPE GAIN 40
SCOPE DEFLECTION - INCHES
0
.5
1.0
1.5
2.0
2.5
3.0
3.5
LOAD ON SLEEVE - POUNDS
10
20
30
40
50

FIGURE VIII

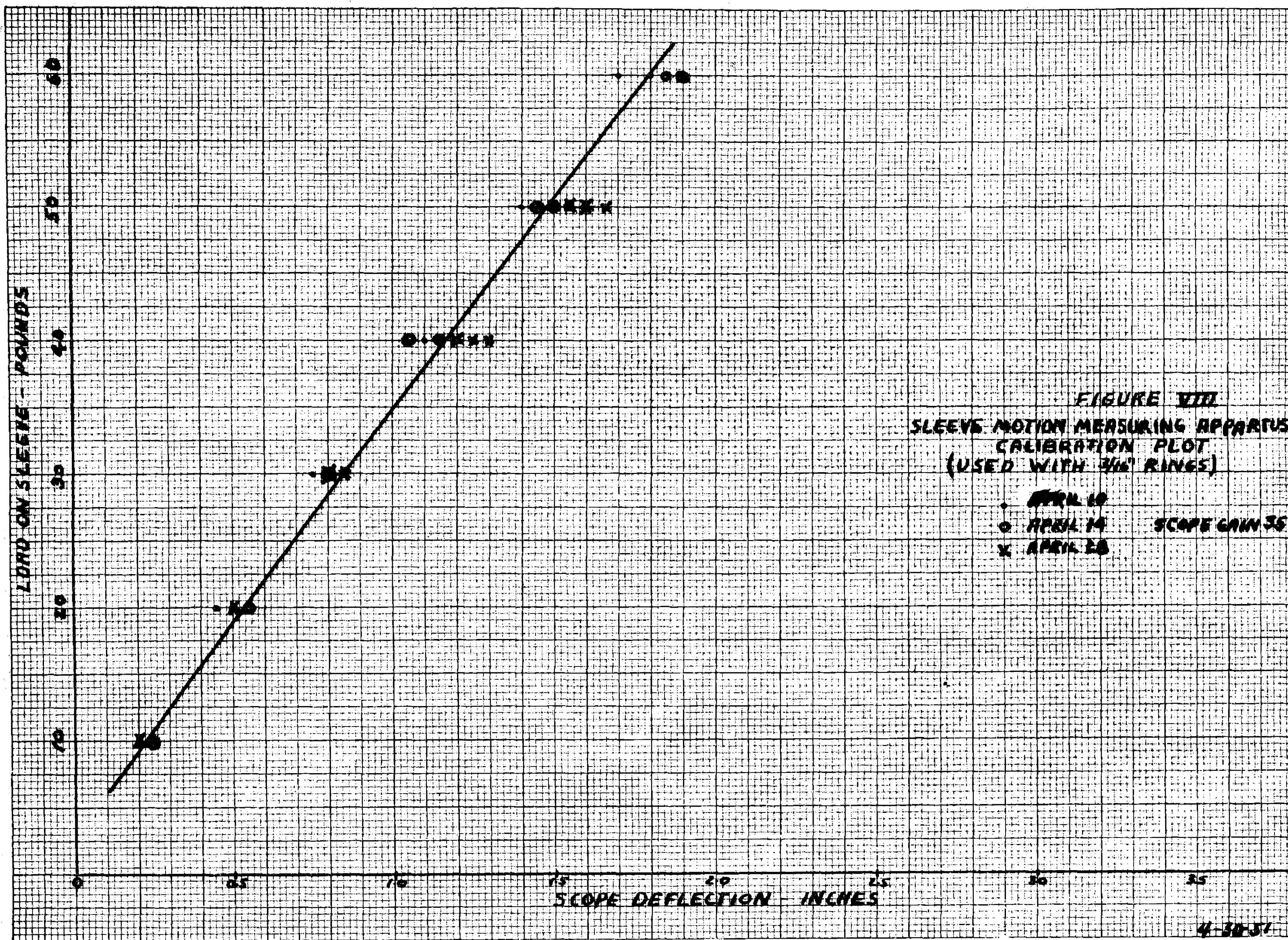


FIGURE VIII
SLEEVE MOTION MEASURING APPARATUS
CALIBRATION PLOT
(USED WITH 3/16" RINGS)
• APRIL 10
o APRIL 14
x APRIL 28
SCOPE GAIN 36
LOAD ON SLEEVE - POUNDS
10
20
30
40
50
60
0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
SCOPE DEFLECTION INCHES
4-30-51
F.A.P.

FIG. IX

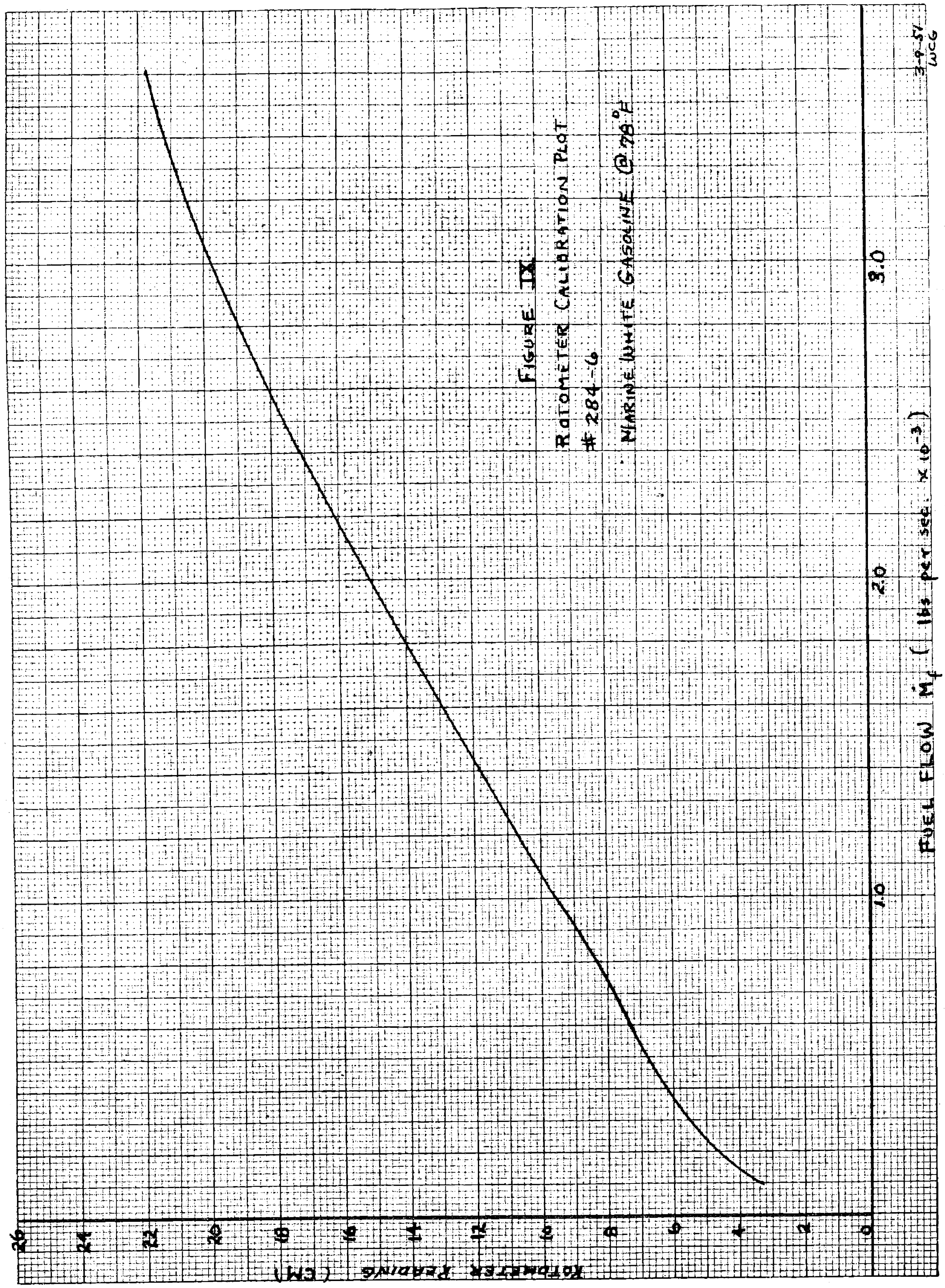
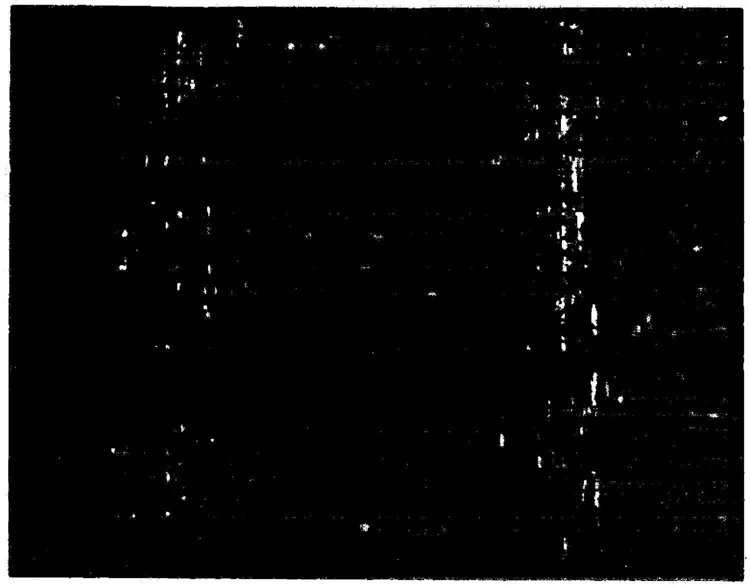
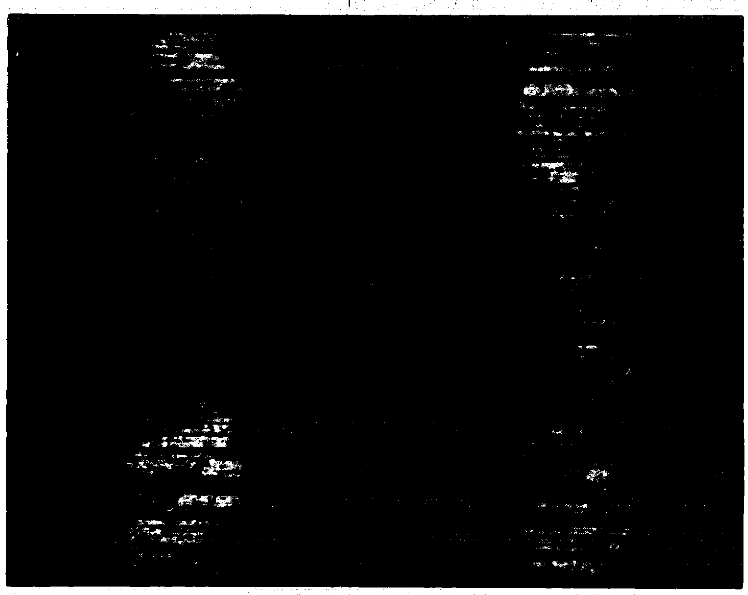


FIGURE IX
ROTOMETER CALIBRATION PLOT
284-6
MARINE WHITE GASOLINE @ 78°F
FUEL FLOW $\dot{M}_f$ (lbs per sec. x 10^{-3})
ROTOMETER READING (CM)
0
1.0
2.0
3.0
0
2
4
6
8
10
12
14
16
18
20
22
24
26
3-9-54
WCG

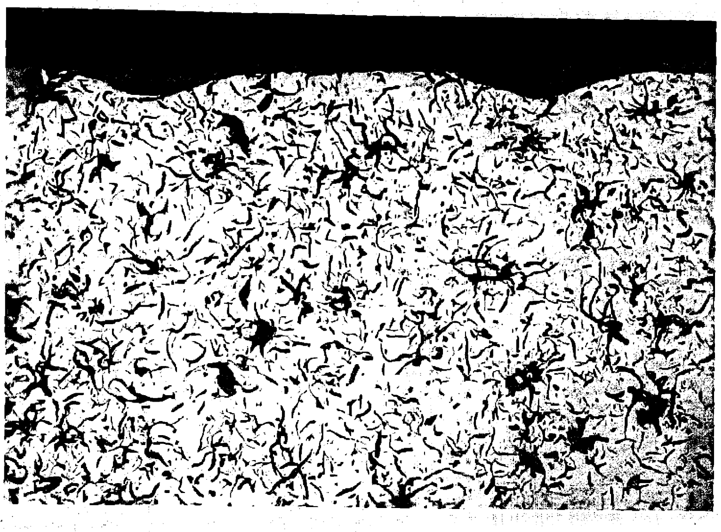


New Ring
2 Lands shown (Light areas)

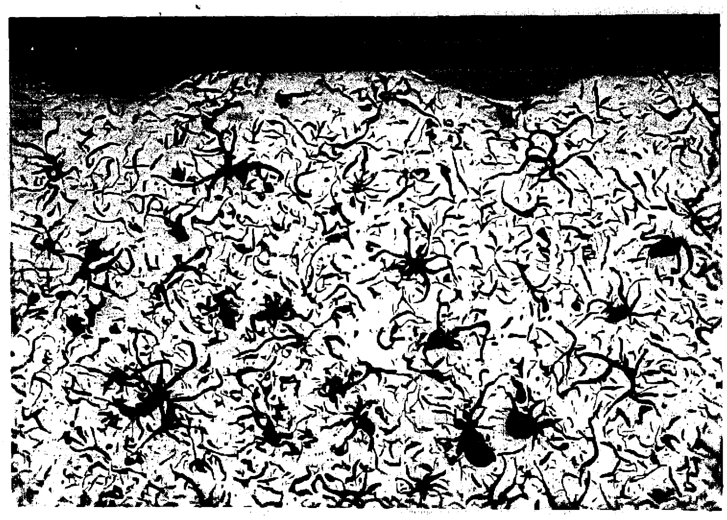


Ring after approx. 6 hours of firing
@ 1000 RPM, mean BMEP 68 psi.
2 Lands shown (Light areas)

Fig. X Photomicrographs 3/16" Interrupted Surface Rings. Plan view near gap. 100X.



New Ring



Ring after approx. 6 hours of firing
@ 1000 RPM, mean BMEP 68 psi.

Fig. XI Photomicrographs 3/16" Interrupted Surface Rings. Profile view near gap. 100X.

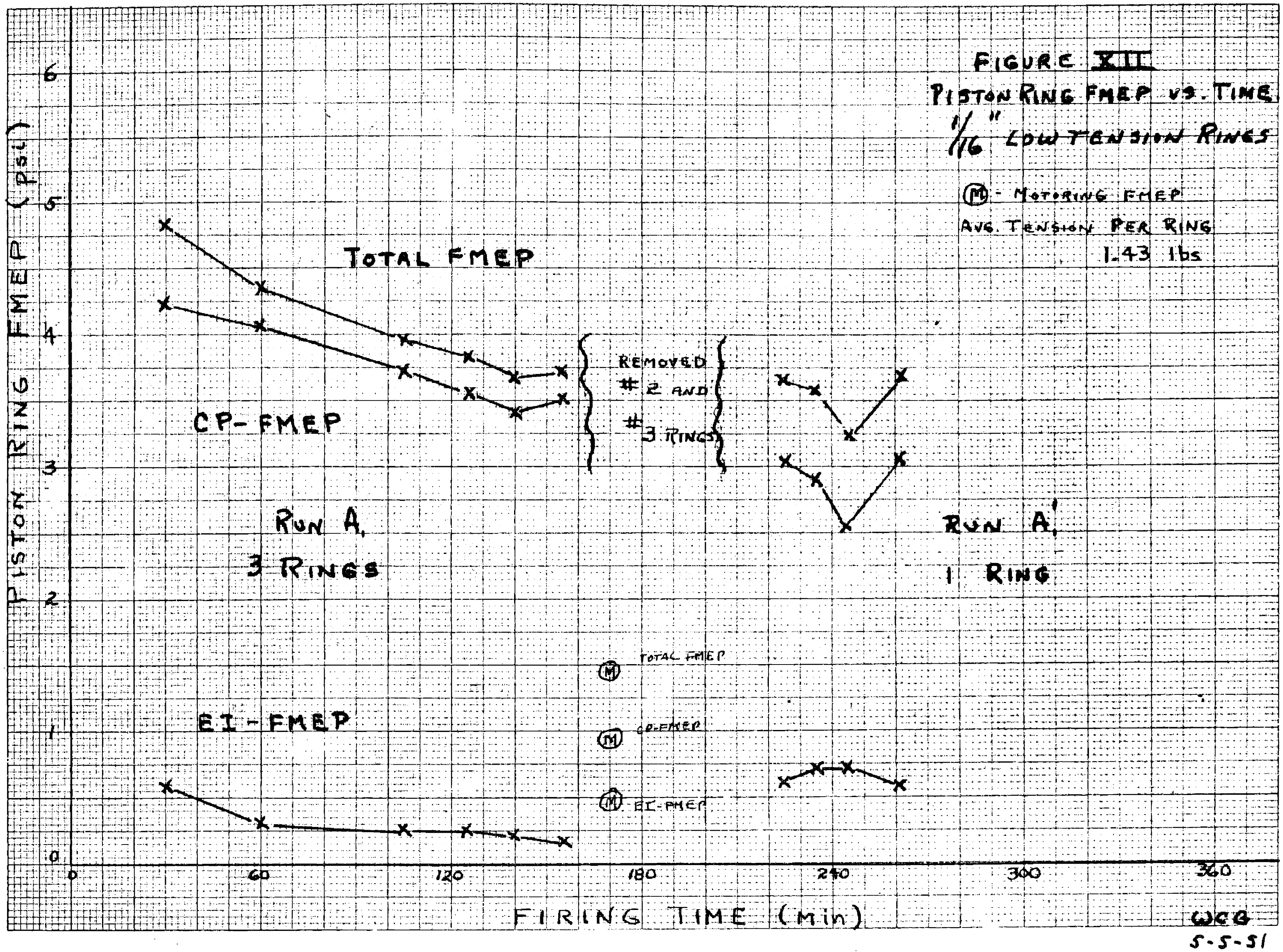


FIGURE XII
PISTON RING FMEP VS. TIME
1/16" LOW TENSION RINGS
(M) - MOTORING FMEP
AVG. TENSION PER RING
1.43 lbs
TOTAL FMEP
CP-FMEP
EI-FMEP
RUN A,
3 RINGS
REMOVED
#2 AND
#3 RINGS
RUN A'
1 RING
TOTAL FMEP
CP-FMEP
EI-FMEP
PISTON RING FMEP (psi)
0
1
2
3
4
5
6
0
60
120
180
240
300
360
FIRING TIME (Min)
WCG
5-5-51

FIG XII

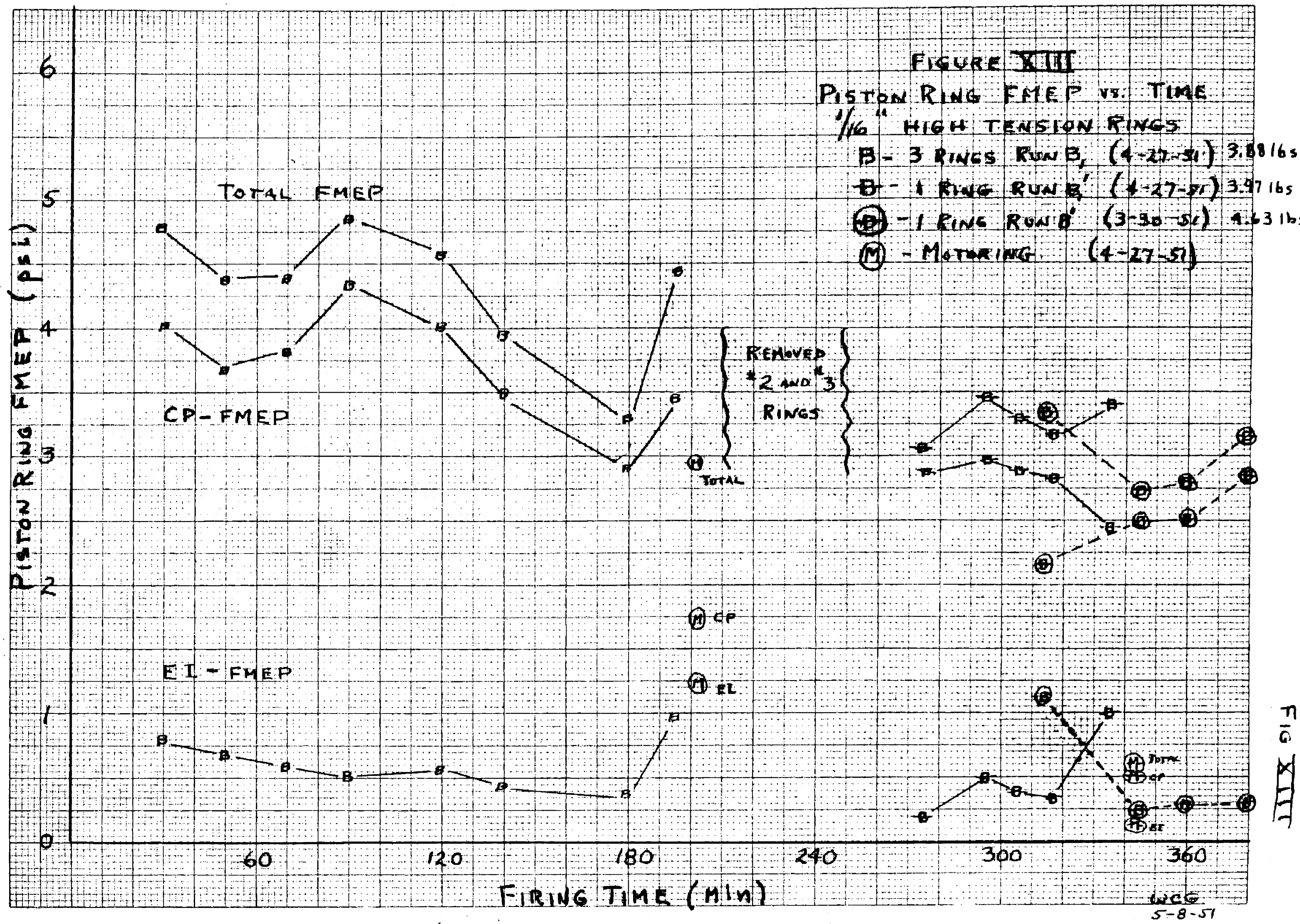


FIGURE XIII
PISTON RING FMEP vs. TIME
1/16" HIGH TENSION RINGS
B - 3 RINGS RUN B, (4-27-51) 3.88 lbs
B - 1 RING RUN B' (4-27-51) 3.97 lbs
B - 1 RING RUN B' (3-30-51) 4.63 lbs
M - MOTORING (4-27-51)
TOTAL FMEP
CP-FMEP
EI-FMEP
REMOVED #2 AND #3 RINGS
TOTAL
CP
EI
PISTON RING FMEP (PSI)
FIRING TIME (MIN)
0
1
2
3
4
5
6
60
120
180
240
300
360
WCG
5-8-51

FIG XIII

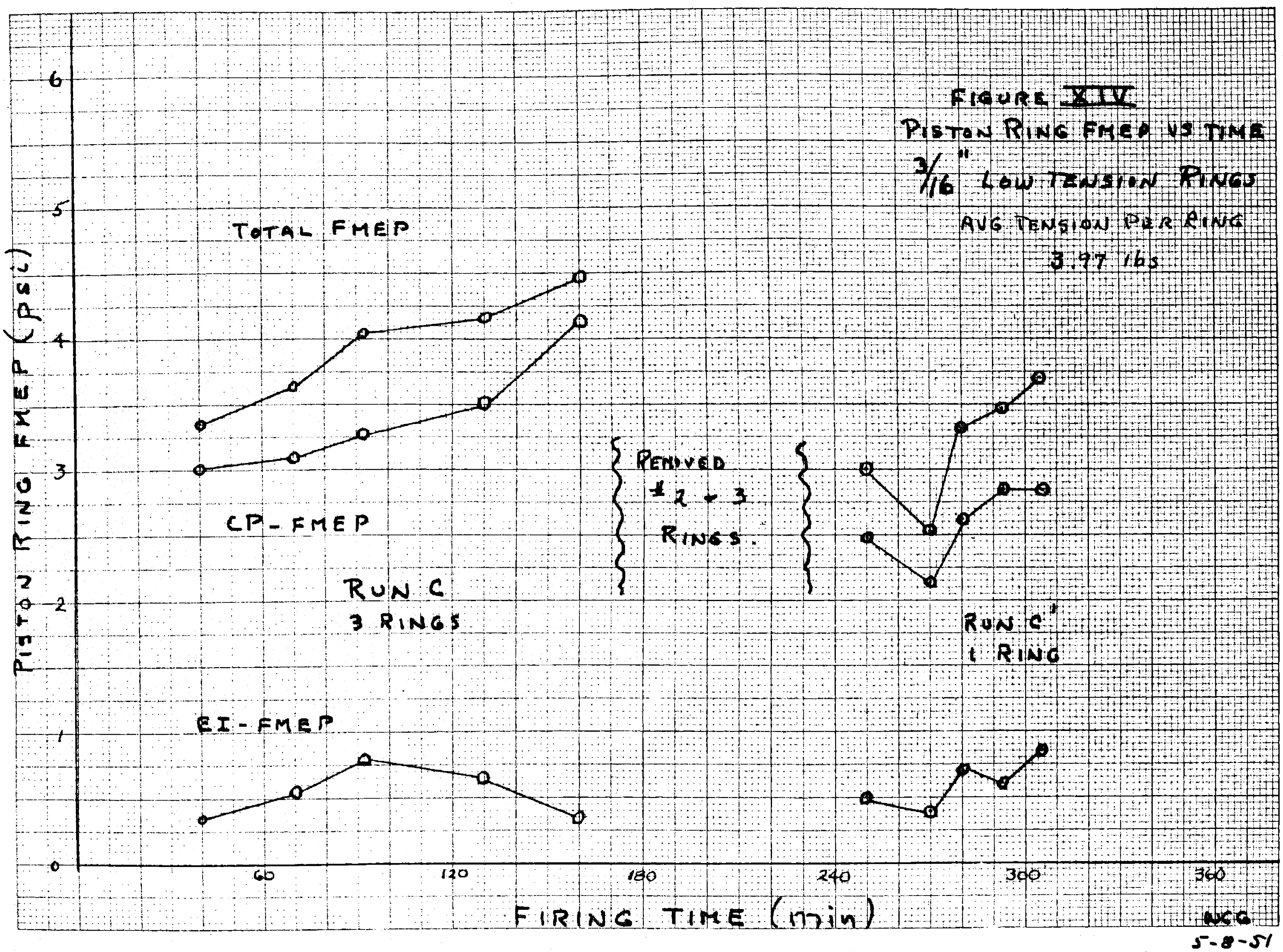


FIGURE XIV
PISTON RING FMEP VS TIME
3/16" LOW TENSION RINGS
AVG TENSION PER RING
3.97 lbs
TOTAL FMEP
CP - FMEP
RUN C
3 RINGS
REMOVED
#2 + 3
RINGS.
RUN C'
1 RING
EI - FMEP
PISTON RING FMEP (psi)
FIRING TIME (min)
0
1
2
3
4
5
6
60
120
180
240
300
360
WCG
5-8-51

FIG XIV

Fig XV

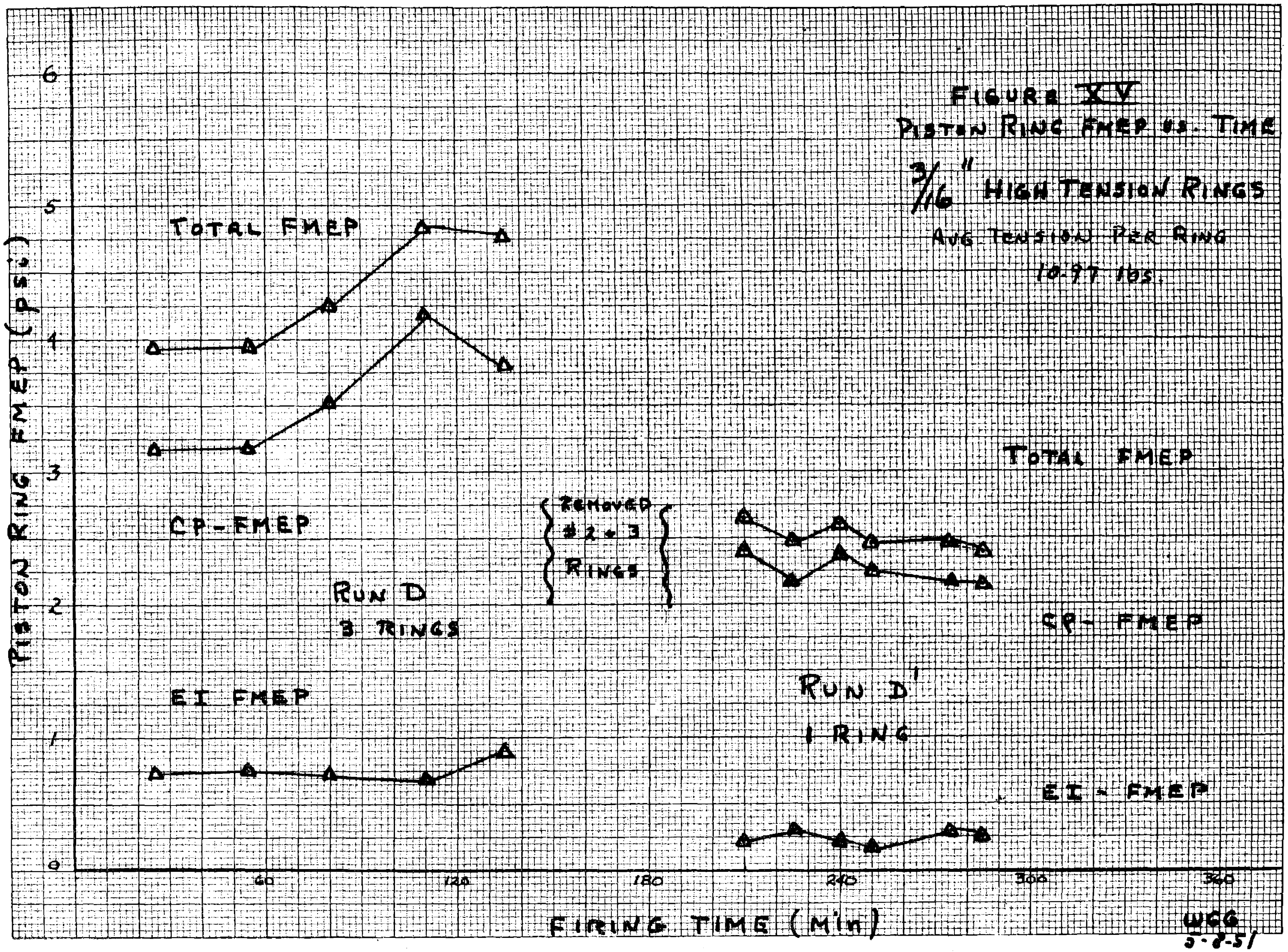


FIGURE XV
PISTON RING FMEP vs. TIME
3/16" HIGH TENSION RINGS
AVG TENSION PER RING
10.97 lbs.
TOTAL FMEP
CP-FMEP
RUN D
3 RINGS
EI FMEP
REMOVED
#2 + 3
RINGS
TOTAL FMEP
CP- FMEP
RUN D'
1 RING
EI - FMEP
PISTON RING FMEP (psi)
FIRING TIME (Min)
0
1
2
3
4
5
6
60
120
180
240
300
360
WGG
5-8-51

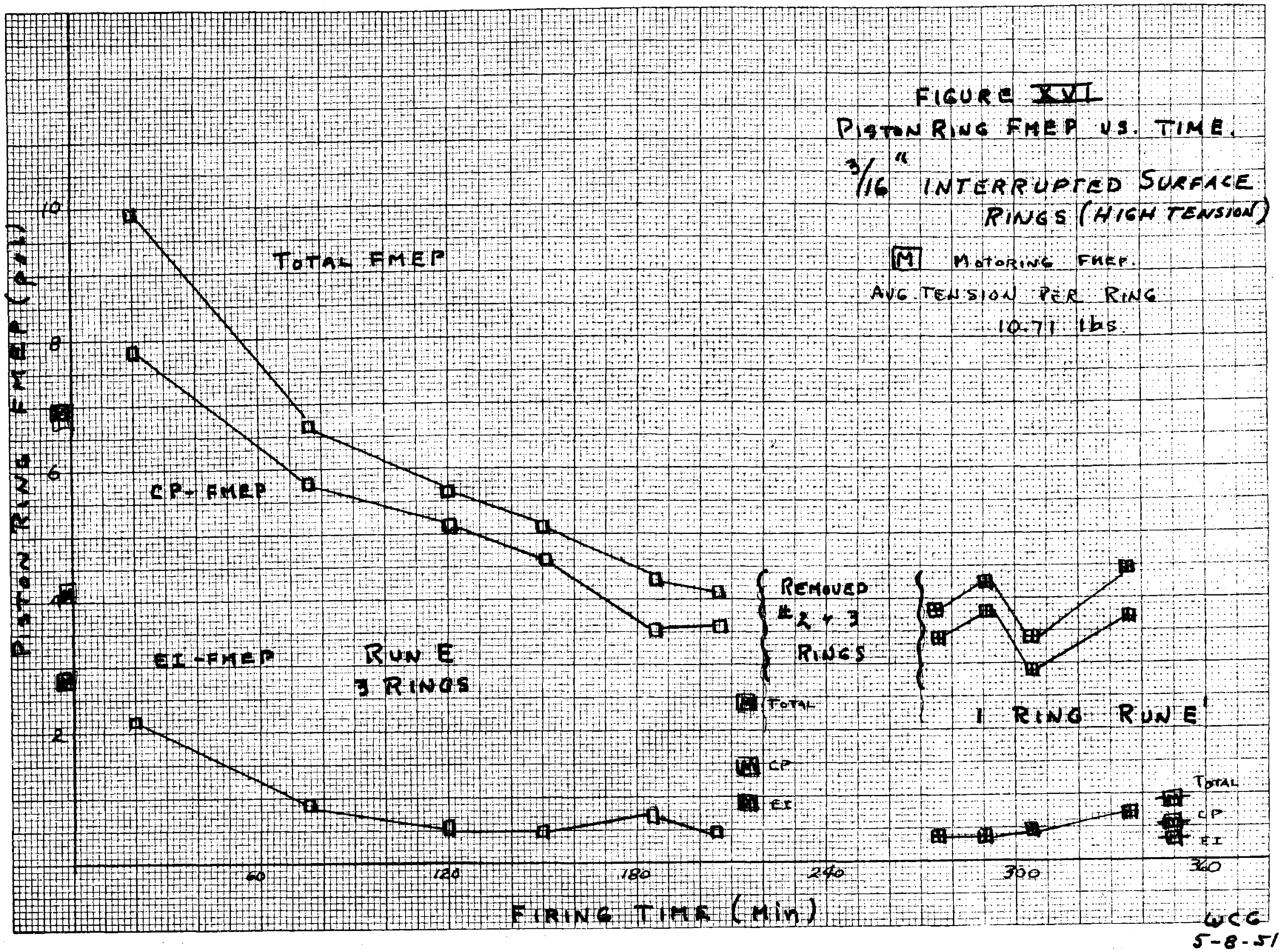


FIGURE XVI
PISTON RING FMEP VS. TIME.
3/16" INTERRUPTED SURFACE
RINGS (HIGH TENSION)
M MOTORING FMEP.
AVG TENSION PER RING
10.71 lbs
TOTAL FMEP
CP-FMEP
EI-FMEP
RUN E
3 RINGS
REMOVED
#2 + 3
RINGS
TOTAL
CP
EI
1 RING RUN E'
TOTAL
CP
EI
PISTON RING FMEP (psi)
10
8
6
2
60
120
180
240
300
360
FIRING TIME (MIN)
WCG
5-8-51

FIG XVI

FIGURE XVII

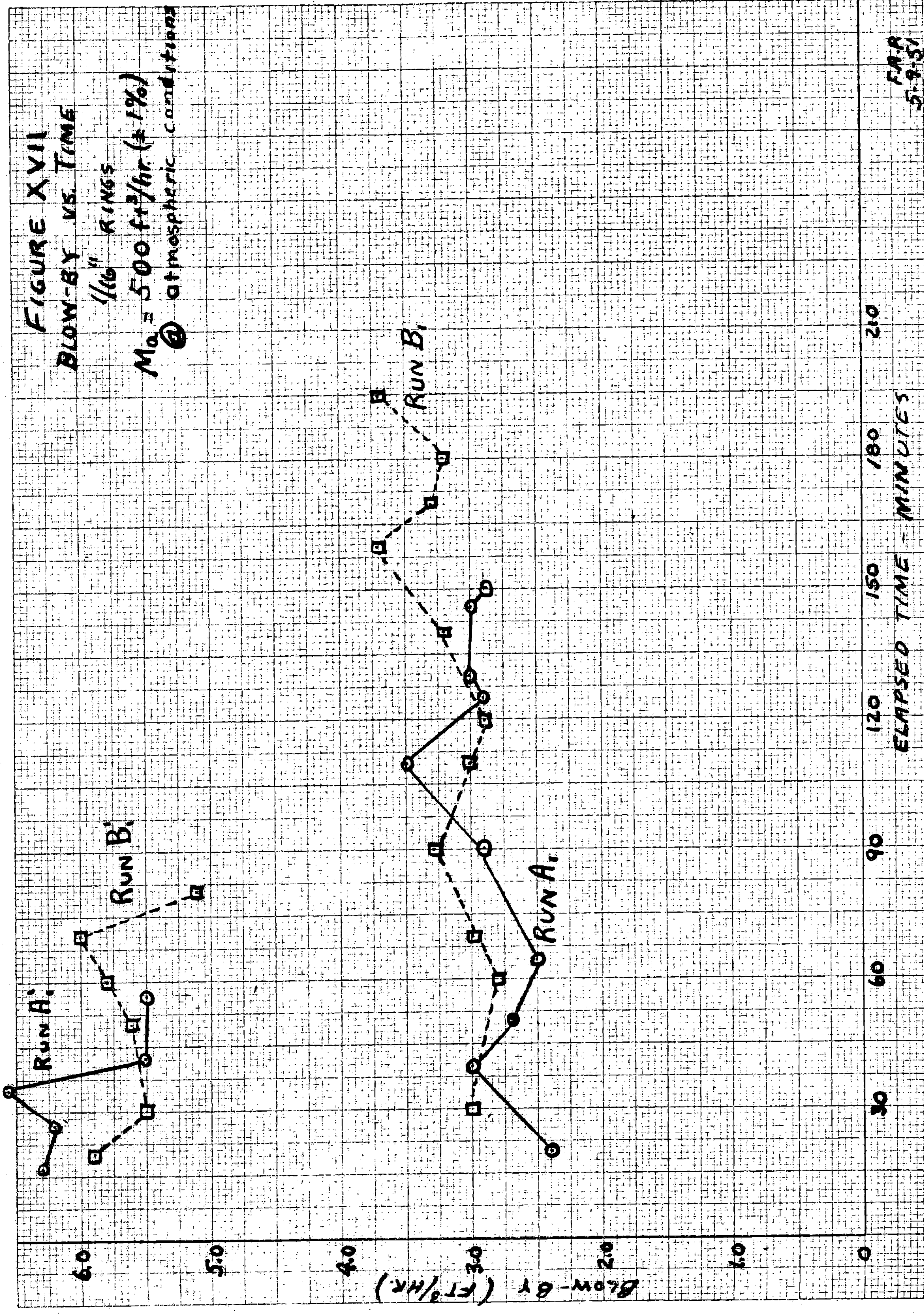


FIGURE XVII
BLOW-BY VS. TIME
1/16" RINGS
Ma = 500 ft³/hr. (± 1%)
@ atmospheric conditions
RUN A1'
RUN B1'
RUN A1
RUN B1
0
1.0
2.0
3.0
4.0
5.0
6.0
BLOW-BY (FT³/HR.)
30
60
90
120
150
180
210
ELAPSED TIME - MINUTES
F.A.R.
5-9-51

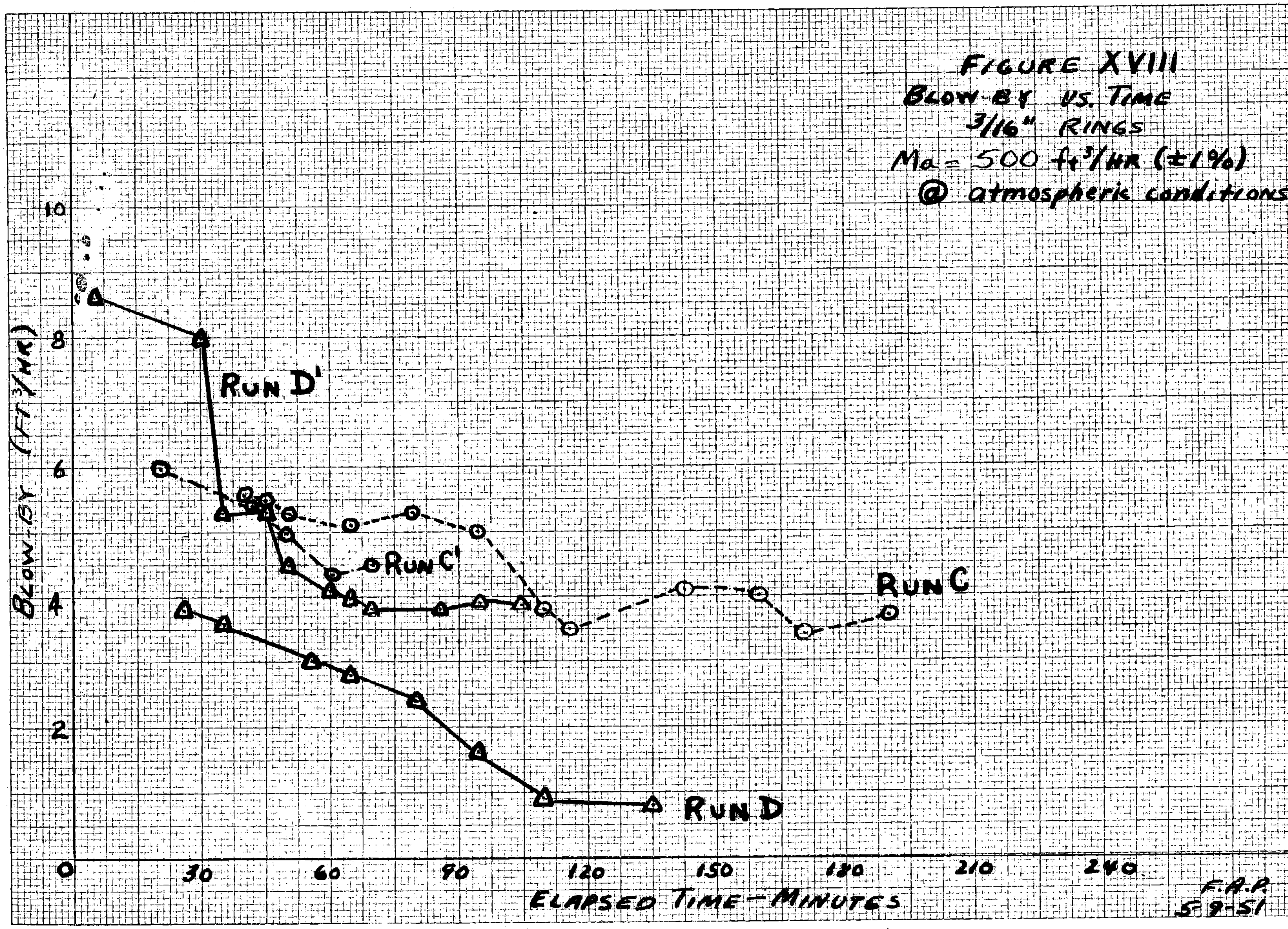


FIGURE XVIII

FIGURE XIX

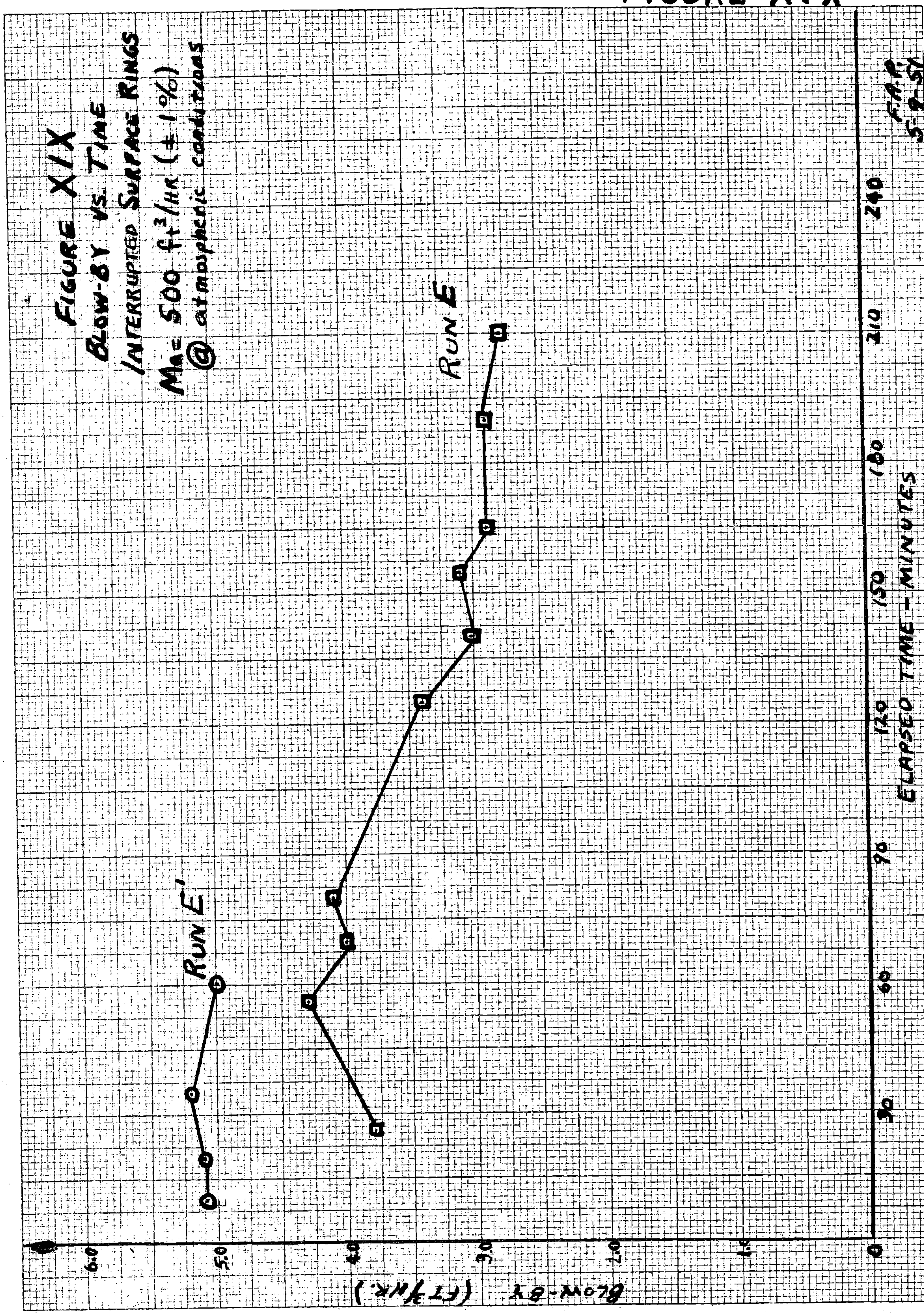
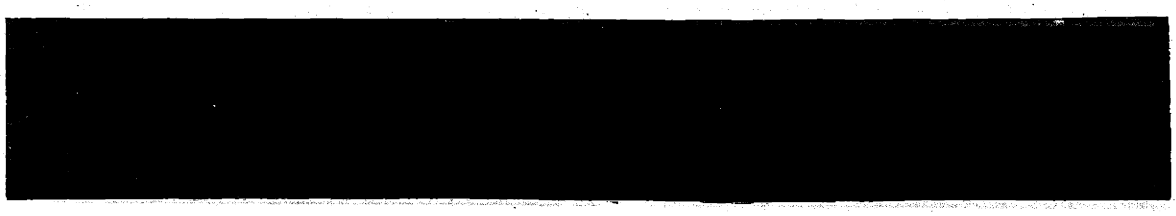


FIGURE XIX
BLOW-BY VS. TIME
INTERRUPTED SURFACE RINGS
Ma= 500 ft³/HR (±1%)
@ atmospheric conditions
RUN E'
RUN E
BLOW-BY (FT³/HR.)
6.0
5.0
4.0
3.0
2.0
1.0
0
30
60
90
120
150
180
210
240
ELAPSED TIME - MINUTES
F.A.P.
5-9-51

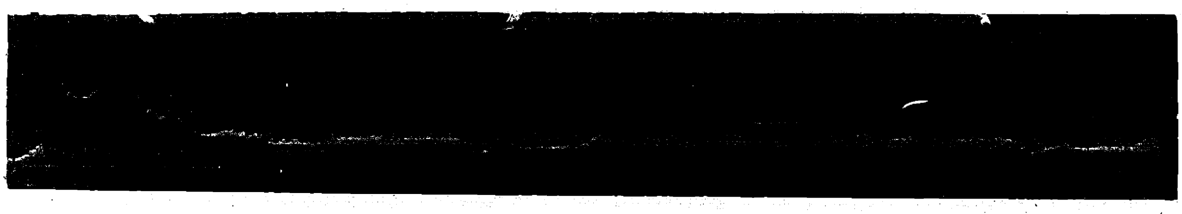
$\theta \rightarrow$

A,2

A,3



A,5



A,'1

A,'3

A,'5

Fig. XX Sample Photographic Records of Runs A,and A'

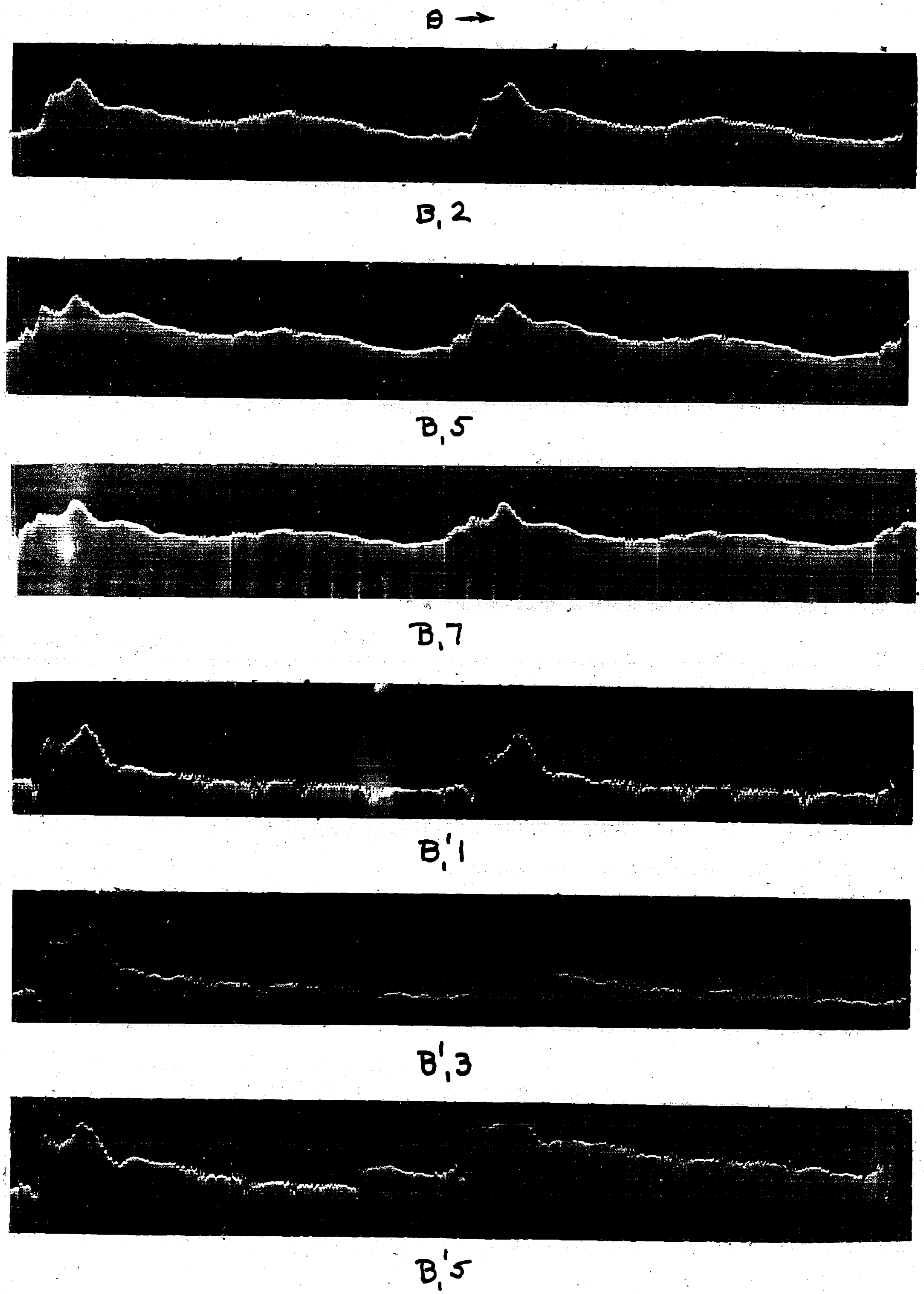


Fig. XXI Sample Photographic Records of Runs B_1 and B'_1

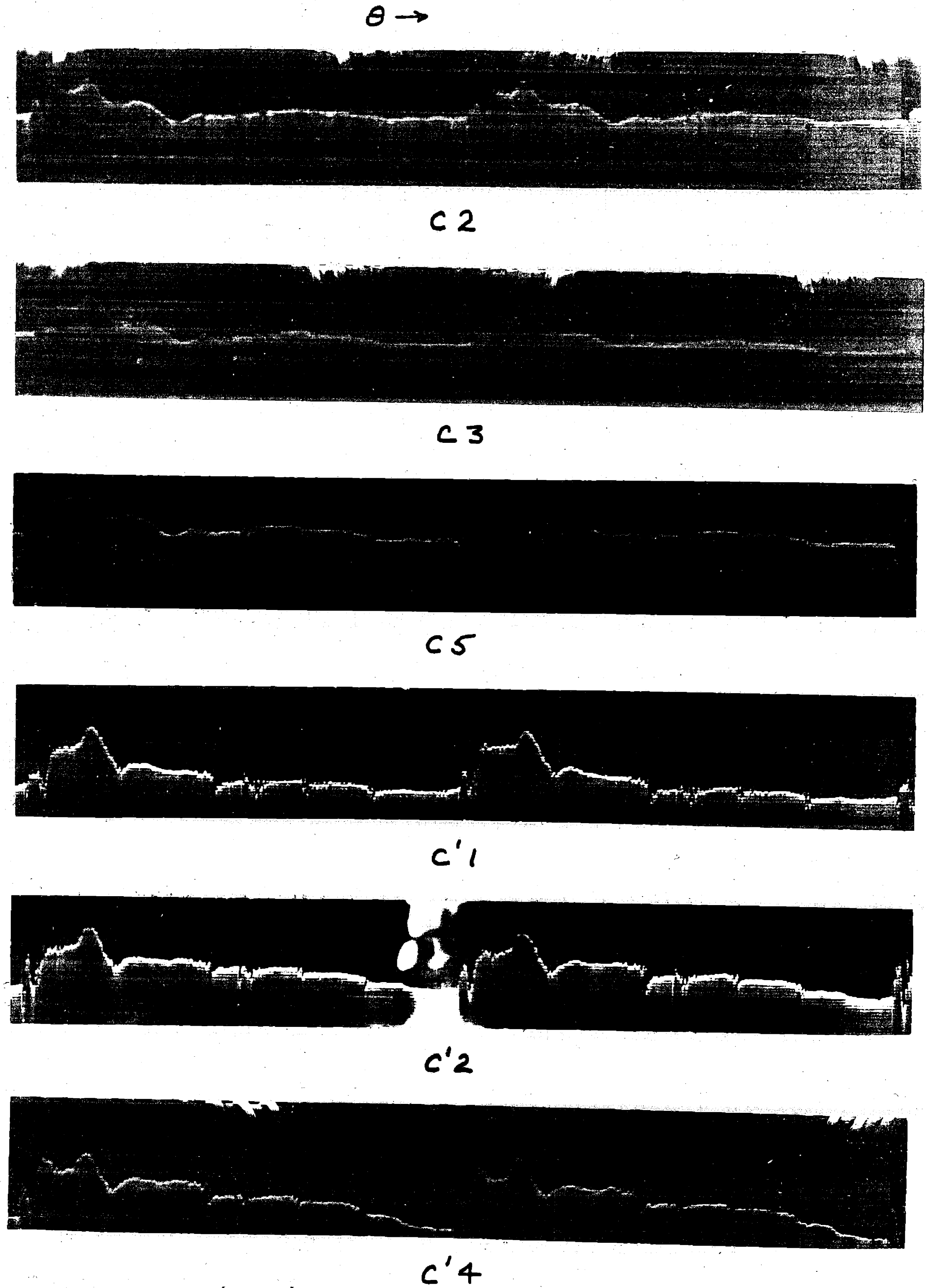
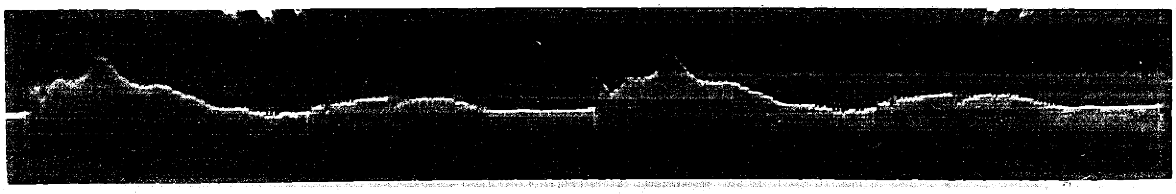


Fig. XXII Sample photographic Records of Runs C and C'

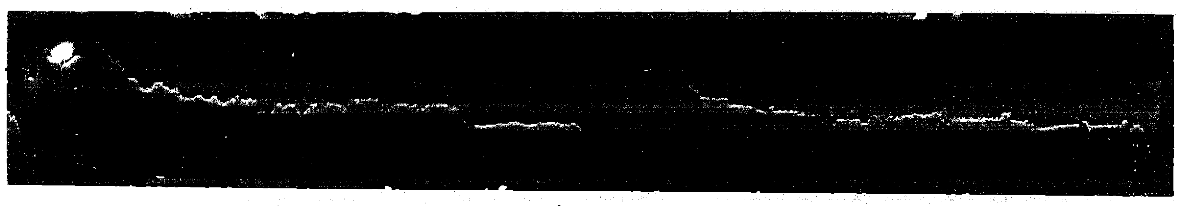
$\theta \rightarrow$



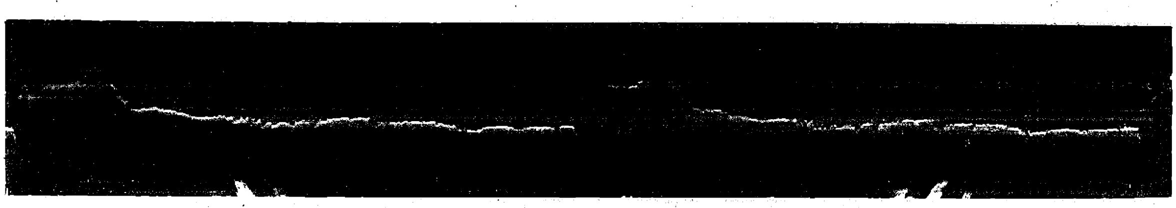
D2

D3

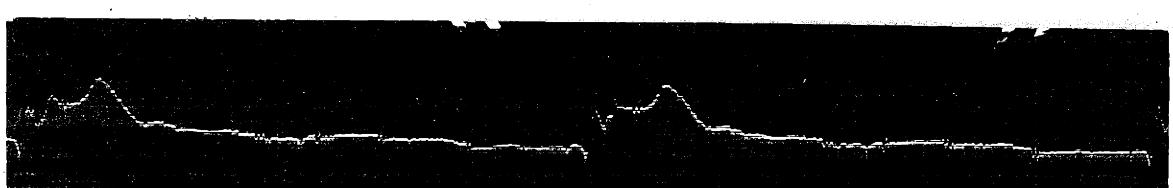
D5



D'1



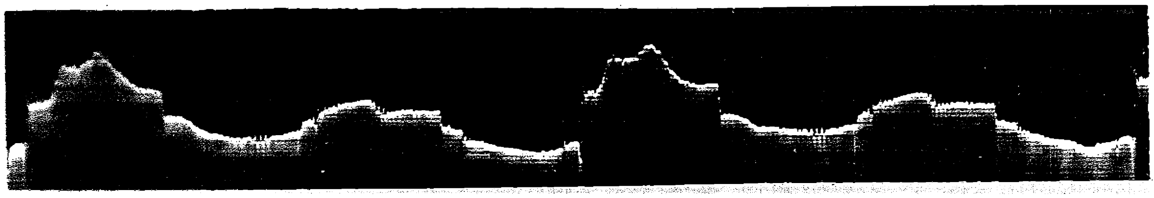
D'3



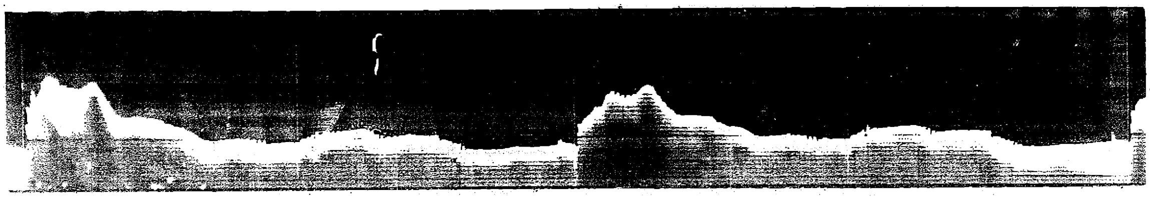
D'6

Fig. XXIII Sample Photographic Records of Runs D and D'

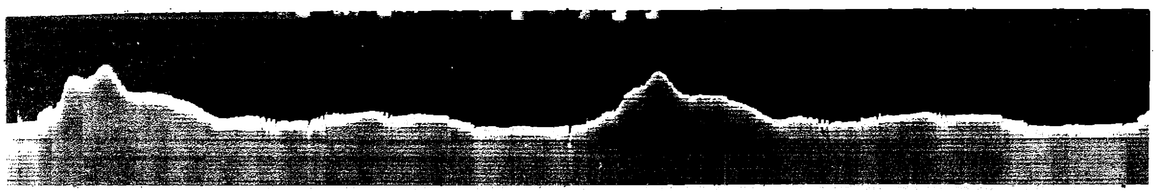
$\theta \rightarrow$



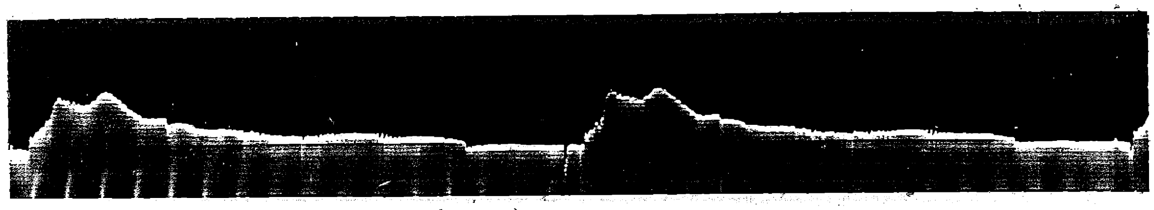
E2



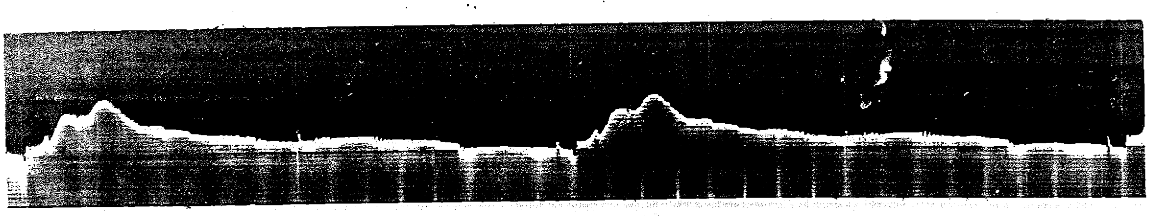
E4



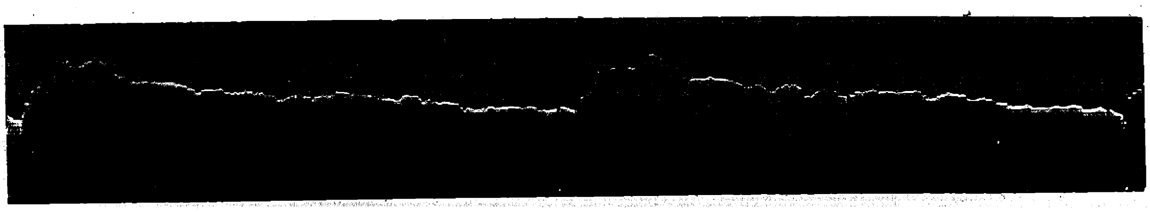
E7



E'2



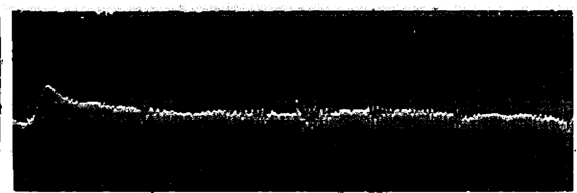
E'3



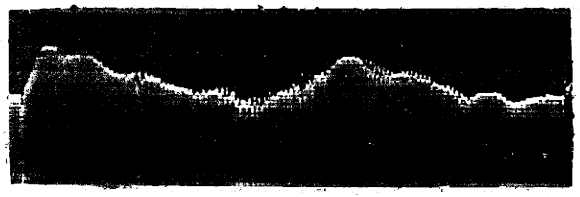
E'4

Fig. XXIV Sample Photographic Records of Runs E and E' θ →

A,7



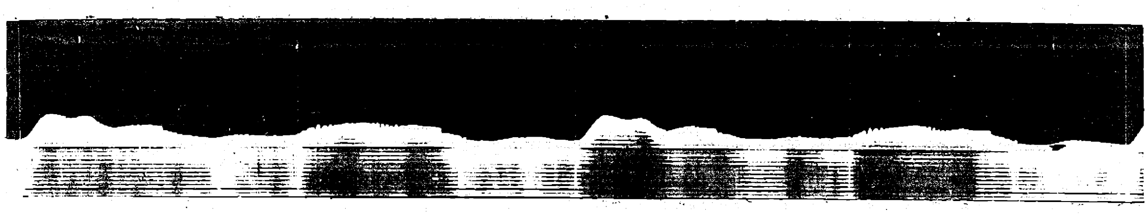
A,'6



B,9

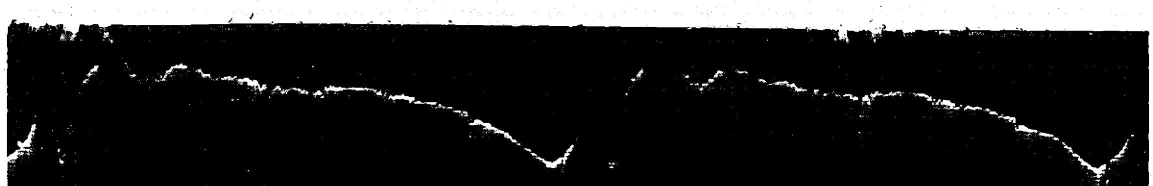
B,'6

E1



E8

E,5



Record of Run A 5 showing possible sticking of junk rings

Fig. XXV Sample Photographic Records of Motoring Runs