NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 817

NITRIDED-STEEL PISTON RINGS FOR ENGINES OF HIGH SPECIFIC POWER

By JOHN H. COLLINS, Jr., EDMOND E. BISSON
and RALPH F. SCHMIEDLIN

1945

For sale by the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C. - - - - - - - - - - Price 20 cents

AERONAUTIC SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Abbreviation	Unit	Abbreviation
Length	l	meter	m	foot (or mile)	ft (or mi)
Time	t	second	s	second (or hour)	sec (or hr)
Force	F	weight of 1 kilogram	kg	weight of 1 pound	lb
Power	P	horsepower (metric)		horsepower	hp
Speed	V	kilometers per hour	kph	miles per hour	mph
		meters per second	mps	feet per second	fps

2. GENERAL SYMBOLS

W Weight $=mg$

g Standard acceleration of gravity $=9.80665$ m/s^2 or 32.1740 ft/sec^2

m Mass $=\frac{W}{g}$

I Moment of inertia $=mk^2$. (Indicate axis of radius of gyration k by proper subscript.)

μ Coefficient of viscosity

ν Kinematic viscosity

ρ Density (mass per unit volume)

Standard density of dry air, 0.12497 kg-m^{-4}-s^2 at 15° C and 760 mm; or 0.002378 lb-ft^{-4} sec^2

Specific weight of "standard" air, 1.2255 kg/m^3 or 0.07651 lb/cu ft

3. AERODYNAMIC SYMBOLS

S Area

S_w Area of wing

G Gap

b Span

c Chord

A Aspect ratio, $\frac{b^2}{S}$

V True air speed

q Dynamic pressure, $\frac{1}{2}\rho V^2$

L Lift, absolute coefficient $C_L=\frac{L}{qS}$

D Drag, absolute coefficient $C_D=\frac{D}{qS}$

D_0 Profile drag, absolute coefficient $C_{D_0}=\frac{D_0}{qS}$

D_i Induced drag, absolute coefficient $C_{D_i}=\frac{D_i}{qS}$

D_p Parasite drag, absolute coefficient $C_{D_p}=\frac{D_p}{qS}$

C Cross-wind force, absolute coefficient $C_C=\frac{C}{qS}$

i_w Angle of setting of wings (relative to thrust line)

i_t Angle of stabilizer setting (relative to thrust line)

Q Resultant moment

Ω Resultant angular velocity

R Reynolds number, $\rho\frac{Vl}{\mu}$ where l is a linear dimension (e.g., for an airfoil of 1.0 ft chord, 100 mph, standard pressure at 15° C, the corresponding Reynolds number is 935,400; or for an airfoil of 1.0 m chord, 100 mps, the corresponding Reynolds number is 6,865,000)

α Angle of attack

ϵ Angle of downwash

α_0 Angle of attack, infinite aspect ratio

α_i Angle of attack, induced

α_a Angle of attack, absolute (measured from zero-lift position)

γ Flight-path angle

REPORT No. 817

NITRIDED-STEEL PISTON RINGS FOR ENGINES OF HIGH SPECIFIC POWER

By JOHN H. COLLINS, Jr., EDMOND E. BISSON
and RALPH F. SCHMIEDLIN

Aircraft Engine Research Laboratory
Cleveland, Ohio

National Advisory Committee for Aeronautics

Headquarters, 1500 New Hampshire Avenue NW., Washington 25, D. C.

Created by act of Congress approved March 3, 1915, for the supervision and direction of the scientific study of the problems of flight (U. S. Code, title 49, sec. 241). Its membership was increased to 15 by act approved March 2, 1929. The members are appointed by the President, and serve as such without compensation.

JEROME C. HUNSAKER, Sc. D., Cambridge, Mass., *Chairman*

LYMAN J. BRIGGS, Ph. D., *Vice Chairman*, Director, National Bureau of Standards.

CHARLES G. ABBOT, Sc. D., *Vice Chairman, Executive Committee*, Secretary, Smithsonian Institution.

HENRY H. ARNOLD, General, United States Army, Commanding General, Army Air Forces, War Department.

WILLIAM A. M. BURDEN, Assistant Secretary of Commerce for Aeronautics.

VANNEVAR BUSH, Sc. D., Director, Office of Scientific Research and Development, Washington, D. C.

WILLIAM F. DURAND, Ph. D. Stanford University, California.

OLIVER P. ECHOLS, Major General, United States Army, Chief of Matériel, Maintenance, and Distribution, Army Air Forces, War Department.

AUBREY W. FITCH, Vice Admiral, United States Navy, Deputy Chief of Naval Operations (Air), Navy Department.

WILLIAM LITTLEWOOD, M. E., Jackson Heights, Long Island, N. Y.

FRANCIS W. REICHELDERFER, Sc. D., Chief, United States Weather Bureau.

LAWRENCE B. RICHARDSON, Rear Admiral, United States Navy, Assistant Chief, Bureau of Aeronautics. Navy Department.

EDWARD WARNER, Sc. D., Civil Aeronautics Board, Washington, D. C.

ORVILLE WRIGHT, Sc. D., Dayton, Ohio.

THEODORE P. WRIGHT, Sc. D., Administrator of Civil Aeronautics, Department of Commerce.

GEORGE W. LEWIS, Sc. D., *Director of Aeronautical Research*

JOHN F. VICTORY, LL. M., Secretary

HENRY J. E. REID, Sc. D., Engineer-in-Charge, Langley Memorial Aeronautical Laboratory, Langley Field, Va.

SMITH J. DEFRANCE, B. S., Engineer-in-Charge, Ames Aeronautical Laboratory, Moffett Field, Calif.

EDWARD R. SHARP, LL. B., Manager, Aircraft Engine Research Laboratory, Cleveland Airport, Cleveland, Ohio

CARLTON KEMPER, B. S., Executive Engineer, Aircraft Engine Research Laboratory, Cleveland Airport, Cleveland, Ohio

---

TECHNICAL COMMITTEES

AERODYNAMICS

POWER PLANTS FOR AIRCRAFT

AIRCRAFT CONSTRUCTION

OPERATING PROBLEMS

MATERIALS RESEARCH COORDINATION

Coordination of Research Needs of Military and Civil Aviation

Preparation of Research Programs

Allocation of Problems

Prevention of Duplication

LANGLEY MEMORIAL AERONAUTICAL LABORATORY
Langley Field, Va.

AMES AERONAUTICAL LABORATORY
Moffett Field, Calif.

AIRCRAFT ENGINE RESEARCH LABORATORY, Cleveland Airport, Cleveland, Ohio

Conduct, under unified control, for all agencies, of scientific research on the fundamental problems of flight

---

OFFICE OF AERONAUTICAL INTELLIGENCE, Washington, D. C.

Collection, classification, compilation, and dissemination of scientific and technical information on aeronautics

REPORT No. 817

NITRIDED-STEEL PISTON RINGS FOR ENGINES OF HIGH SPECIFIC POWER

By John H. Collins, Jr., Edmond E. Bisson, and Ralph F. Schmiedlin

SUMMARY

Several designs of nitrided-steel piston rings were performance-tested under variable conditions of output. The necessity of good surface finish and conformity of the ring to the bore was indicated in the preliminary tests. Nitrided-steel rings of the same dimensions as cast-iron rings operating on the original piston were unsatisfactory and the final design was a lighter, rectangular, thin face-width ring used on a piston having a maximum cross-head area and a revised skirt shape. Results were obtained from single-cylinder and multicylinder engine runs.

The thin, nitrided-steel rings were performance-tested in both nitrided and porous chrome-plated cylinders with good results. The nitrided-steel cylinders were stock production items and the porous chrome-plated cylinders were worn cylinders that had been reclaimed by plating back to size. The use of nitrided-steel rings in chrome-plated cylinders offers attractive possibilities that require further investigation.

Good ring and cylinder performance characteristics were obtained. Runs under dust conditions indicated that the nitrided-steel rings maintained acceptable oil control from three to four times longer than the stock cast-iron rings. Lubricating-oil consumption was somewhat higher with nitrided-steel rings than that usually encountered with the original cast-iron ring assembly when the cast-iron rings are in the best condition of run-in. The lubricating-oil consumption with nitrided-steel rings tended to improve with operating time, whereas the opposite trend was exhibited by cast-iron rings.

INTRODUCTION

When aircraft engines are operated at specific power outputs in excess of their maximum ratings, the piston rings are among the first engine parts to fail. These failures manifest themselves by general increases in specific oil consumption, in blow-by, and in subsequent loss in power caused by rapid ring wear and loss in ring tension. Operational variables that considerably affect piston-ring and cylinder wear are: dust; lubrication, quantity, and quality; brake mean effective pressure; engine speed; and operating temperatures. Because of the early failure of piston rings, an investigation was conducted in order to determine the necessary characteristics of piston rings for operation at extreme conditions of the operational variables particularly including high specific power. Bench and engine tests were made on a variety of piston-ring materials and ring designs. Some of the ring materials were eliminated by the bench tests. Preliminary runs were made in single-cylinder engines. The choice of the procedure for the engine runs reported was made with a view to selecting conditions that would give the required results in the minimum time. The final combination of nitrided-steel rings in both nitrided-steel and chrome-plated cylinders was performance-tested under a wide variety of operating conditions in both single-cylinder and multicylinder engines.

The piston rings were performance-tested in single-cylinder engines at Langley Memorial Aeronautical Laboratory from 1939 to 1942. At the recommendation of the NACA, tests with multicylinder and single-cylinder engines were made by the Bureau of Aeronautics, Navy Department, at the Aeronautical Engine Laboratory of the Naval Aircraft Factory in Philadelphia and by the Army Air Forces, Materiel Command, at Wright Field. The thin, nitrided-steel rings used in these investigations were manufactured by and tested in cooperation with the Borg-Warner Corporation, Spring Division, Bellwood, Ill.

DEFINITIONS

Piston-ring terms to be used in this report are defined as follows:

- face—the part of the piston ring that is adjacent to, or in direct contact with, the cylinder wall
- face width—the width of the ring face
- side—the part of the ring that contacts the piston grooves
- ring assembly—the entire group of rings used on any individual piston, regardless of ring type, material, or position with respect to the piston pin
- radial depth—the radial dimension from the center of the face to the center of the back of the ring
- diametral tension—the force in pounds, which is applied along a radius 90° from the gap, required to close the ring to its nominal diameter. Ring tension as used and measured in this investigation is purely an arbitrary measurement of the characteristics of a piston ring
- unit wall pressure—the force exerted by the piston-ring face against the cylinder wall in pounds per square inch. This unit pressure is obtained by use of the following equation from reference 1:

$$p=\frac{0.76T}{DW}$$

where

p unit pressure, pounds per square inch
T diametral tension, pounds
D ring diameter, inches
W face width, inches

(It is not specified in reference 1 if measurements of ring tension were made with the ring compressed so as to produce the correct ring gap or the correct ring diameter. An analysis made subsequent to the original publication of the data reported herein indicates that the constant 0.76 applies to measurements made with the ring compressed to the correct gap. The analysis showed that a value of 0.88 should be used if measurements are made with the ring compressed to the proper diameter, as was done in obtaining most of these data. Correction of the data reported is felt to be unwarranted, however, because the values obtained were used only for qualitative comparison.)

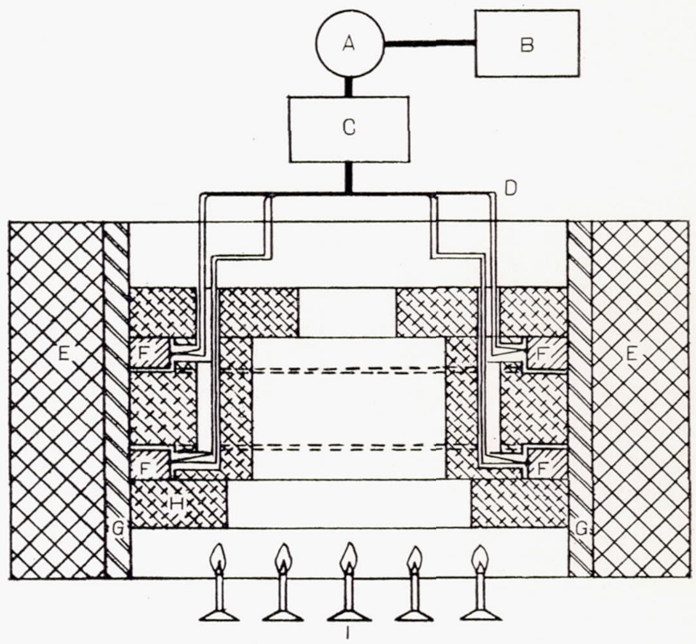
The unit wall pressure as defined herein is only an approximate average value inasmuch as true unit pressures can be determined only by such an instrument as a radial-pressure gage.

APPARATUS AND PROCEDURE

TEMPERATURE INVESTIGATION

The object of the temperature investigation was to eliminate quickly the materials that had the least chance of proving satisfactory for rings intended to operate at high power outputs. The heater shown in figure 1 was designed and built to check the effect of temperature on stress relief. Stress relief of a ring is defined as the loss in tension that occurs in the ring at elevated temperatures. Two piston rings with two thermocouples leading from each ring can be simultaneously tested in this heater. Rings were placed in the heater, which compressed them to their nominal diameter. They were heated to the predetermined temperature and held at this temperature for 10 minutes, after which they were removed from the heater and cooled in still air. The diametral tension of the rings was determined before and after each heat period. Tentative ring materials, such as alloy cast iron, high-speed steel, and several kinds of nitrided steel, were included in this temperature investigation.

In the determination of the effect of prolonged heating at elevated temperatures, rings were compressed to their nominal diameter by installing them in a section of a cylinder of the proper size. This assembly was then placed in a heat-treating oven that had been heated to the predetermined temperature. An unconfined ring was also placed in the furnace in order to evaluate the effect of temperature alone, as compared to the combined effect of temperature and stress. At the expiration of the heating period (1 hr in one run, 6 to 8 hr in another run), the cylinder was removed from the oven and the rings still confined in the cylinder were allowed to cool. The unconfined ring was removed at the same time and allowed to cool in the unconfined condition. Diametral tension was determined before and after each heating period. In this investigation, the same rings were tested over a range from room temperature to 1100° F or until the ring collapsed, that is until the ring lost a large part of its tension.



A, Selector switch
B, Potentiometer
C, Cold-junction box
D, Thermocouple leads, constantan and iron
E, Insulation
F, Piston rings
G, Section of 5-inch cylinder
H, Aluminum
I, Gas burner

FIGURE 1.—Piston-ring heater.

SINGLE-CYLINDER-ENGINE INVESTIGATION

The following single-cylinder engine setups were used in the NACA tests:

Cylinder	Crankcase	Bore and stroke (in.)	Compression ratio	Fuel system
NACA compression ignition; liquid cooled	NACA universal test engine	5 by 6	12.0	Fuel injection
Air cooled	NACA universal test engine	6⅛ by 7	6.7	Carburetor
Air cooled	[1] Radial engine	6⅛ by 6⅞	6.7	Carburetor
Air cooled	[1] Radial engine	6⅛ by 6⅞	6.7	Carburetor

[1] Rebalanced for single-cylinder operation. (See fig. 2 for typical setup.)

Combustion-air flow to the engines was measured by a sharp-edge, thin-plate orifice assembled according to A.S.M.E. standards. Power was absorbed and measured by a cradle-type electric dynamometer in all single-cylinder-engine runs.

Cooling air was supplied by separately driven blowers and flow was varied to produce the desired cylinder-head and

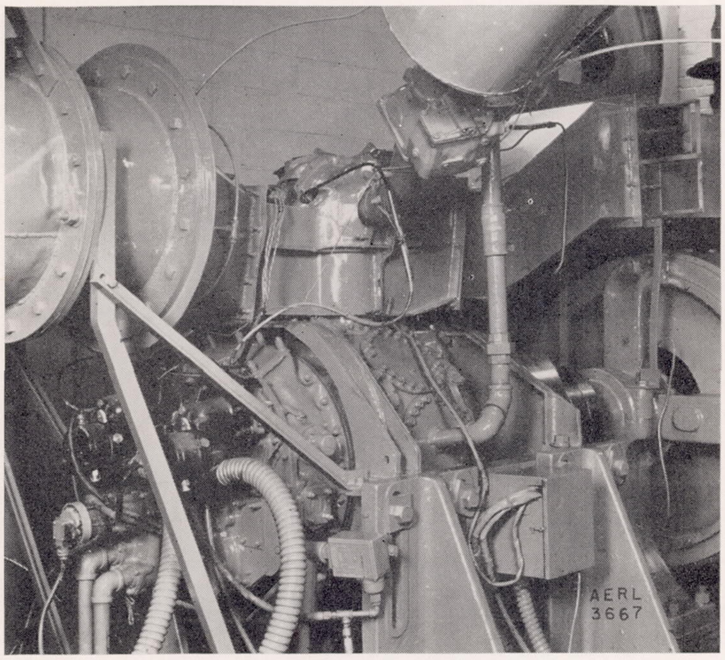


FIGURE 2.—Single-cylinder-engine assembly.

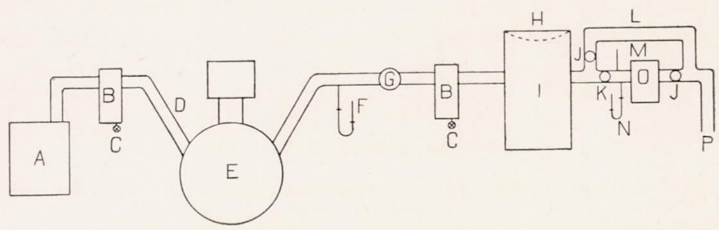
cylinder temperatures. These temperatures were measured by 13 iron-constantan thermocouples distributed over head and cylinder. Temperatures were held to the following limits for all single-cylinder-engine runs at all outputs: rear spark-plug bushing, 450° F; rear center of cylinder, 350° F.

High-power endurance tests were run under speed, lubrication, and temperature conditions conducive to ring failure through scuffing, scoring, feathering, and high wear rates.

Blow-by was measured by a positive-displacement gas meter connected to the crankcase-breather system (fig. 3). As indicated in the diagrammatic sketch, blow-by is piped through a large surge tank with a flexible head, which is installed to damp out pressure fluctuations. By use of this surge tank, the effect of variable speed on the meter calibration becomes negligible. As is the case in all oil-system installations utilizing a dry sump and a large-capacity oil-scavenge pump, some of the blow-by gases are pumped into the oil-storage tank. This condition is compensated for by using a sealed storage tank and by returning this blow-by to the crankcase; thereby all blow-by will eventually pass through the meter.

The crankcase pressure was arbitrarily set at ½ inch of water below atmospheric pressure (to permit leakage corrections to be made) and was maintained at this value throughout a complete run by means of a throttle valve (K in fig. 3) at the vacuum source. A leakage-calibration curve was taken at variable speeds for the crankcase pressure set at ½ inch of water below atmospheric pressure. In this calibration, the engine was motored without compression so that no gas would leak by the rings. Leakage losses in all tests were found to be negligible. Calibration of the leakage was repeated from time to time as a check on the efficiency of the sealing of crankcase and blow-by system.

Ring wear was measured by cleaning and weighing the rings before and after each test. Specific oil consumption was measured by the volume method corrected for temperature. The installation of special fittings and a separate scavenging pump permitted the measurement of the power-section oil flow for the single-cylinder engine using a converted radial-engine crankcase.

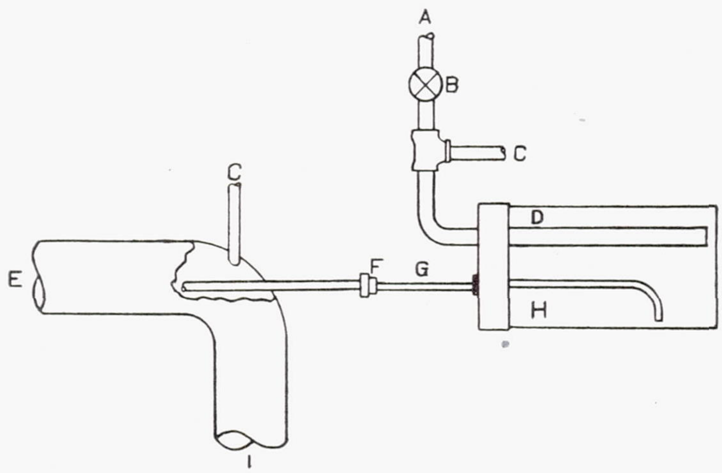


A, Oil-storage tank
B, Surge tank and oil separator
C, Drains
D, Line for returning (to crankcase) blow-by handled by oil-scavenging pump in maintaining dry sump crankcase
E, Crankcase
F, Crankcase-pressure manometer
G, Sight glass
H, Flexible head
I, Surge tank
J, Shut-off valves
K, Throttle valve
L, Bypass
M, Meter inlet thermometer
N, Meter inlet-pressure manometer
O, Gas meter
P, Pipe leading to exhaust trench (approximately 7 in. water vacuum)

FIGURE 3.—Schematic diagram of blow-by system.

Measurement of surface quality was made by measuring surface roughness of the nitrided surfaces of cylinder and rings. Surface roughness was measured with a Brush surface analyzer and a Profilometer; porosity of the chrome-plated surfaces was measured by a replica method developed by the NACA. In this method, a replica of the surface is taken with a plastic, such as stripping lacquer or celluloid, and this replica is photographed at a magnification of 100. It has been found by experiment that a good correlation is obtained between the porosity as measured by the replica method and the porosity obtained by the method of photomicrography of the sample itself. Percentage porosity is defined as the percentage of pits per unit area. The pitted area on the photomicrographs was determined by the method of counting squares and the percentage porosity, based on the nominal surface area, was thus obtained.

The single-cylinder-engine investigation performed by the Bureau of Aeronautics at the Naval Aircraft Factory was



A, Compressed-air supply
B, Globe valve
C, Pipe leading to 20-inch mercury manometer
D, ⅜-inch diameter tube, end plugged, with 5 equally spaced ⅛-inch-diameter holes
E, Pipe leading to carburetor-air scoop
F, Knurled knob clamped to tube
G, ⅛-inch-inside-diameter stainless-steel tube, slip-type fitting on bottle
H, Extra heavy bottle for dust
I, Pipe leading from combustion-air source

FIGURE 4.—Dust-injection equipment used by Bureau of Aeronautics. Pressure differential of 2 pounds per square inch maintained between bottle and manifold. Bottle moved on tube to pick up dust. Hood and slide provided for bottle to aid in operation and to prevent injury in case of excess air pressure.

conducted on nitrided-steel piston rings assembled in a nitrided air-cooled cylinder of 6⅛-by-6⅞-inch bore and stroke to determine the effect of dust on ring wear and on oil control. These dust tests were performed at an engine speed of 2000 rpm, a brake mean effective pressure of 160 pounds per square inch, a carburetor-air temperature of 100° F, an oil-in temperature of 185° F, and a specific fuel consumption of 0.70 pound per brake horsepower-hour. At these conditions, a dust cycle was run consisting of the injection of 2 grams of dust into the combustion air over a 10-minute period every ½ hour for 3 hours; an oil-consumption check run was then made without dust for 4 hours. Figure 4 shows the dust-injection equipment.

The dust used in this investigation was made from a natural Arizona dust, the analysis of which is given in the following table:

Particle size [a]		Chemical composition of sample	
(microns)	(percent)	Compound	(percent)
0–5	39±2	Fe_2O_3	4.58
5–10	18±3	Al_2O_3	15.98
10–20	16±3	CaO	2.91
20–40	18±3	MgO	.77
Over 40	9±3	Na_2O	4.61
		SiO_2	68.47
		Other (ignition loss)	2.68

[a] 100 percent of dust passes through a 200-mesh screen.

MULTICYLINDER-ENGINE INVESTIGATION

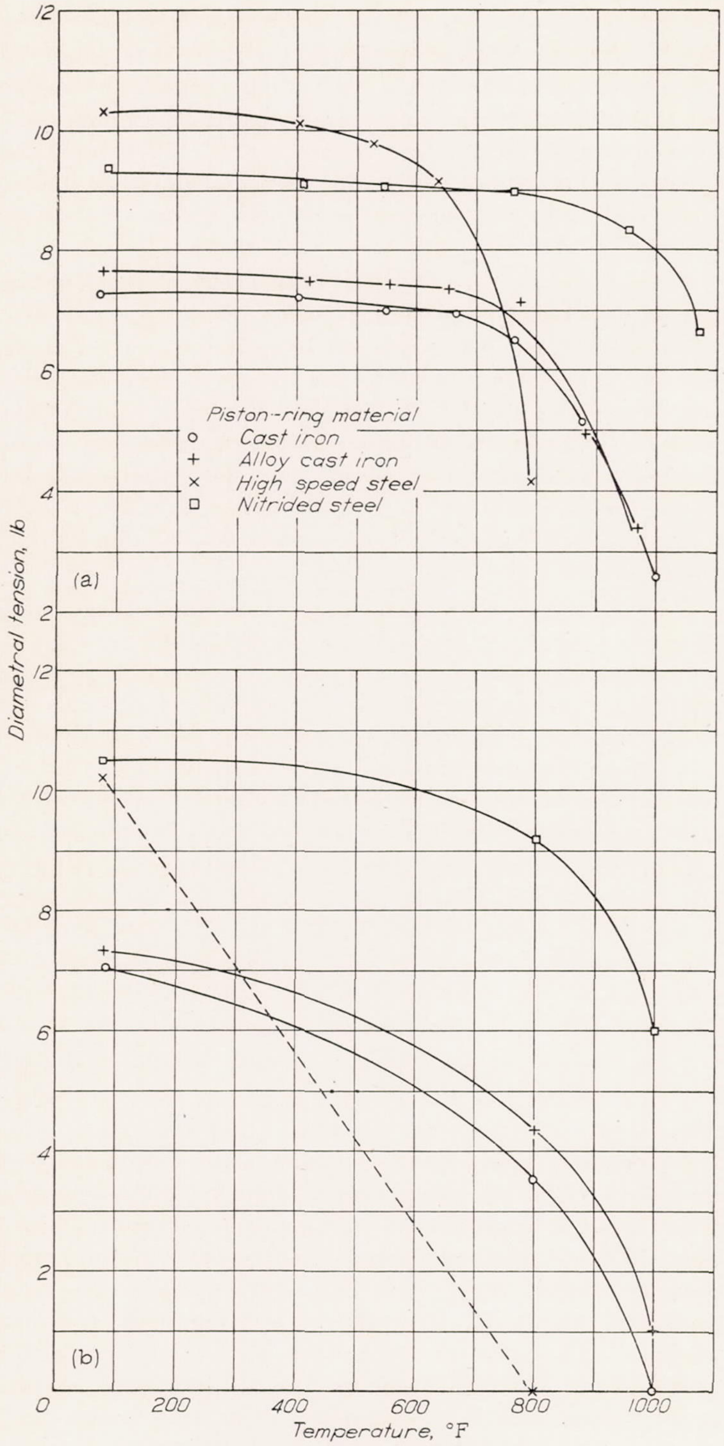
Some of the multicylinder-engine research on nitrided-steel piston rings was performed by the Bureau of Aeronautics at the Naval Aircraft Factory on an 1820-cubic-inch displacement engine. This engine was mounted on a test stand and loaded with a propeller. Cooling-air flow was varied to obtain the desired cylinder temperatures. Specific oil consumption was measured by the weight method using a balanced weighing tank.

The following tests were performed:

1. Continuous operation at take-off power: brake horsepower, 1200; engine speed, 2500 rpm
2. An endurance run made according to the program of the appendix
3. Dust tests: brake horsepower, 740; engine speed, 2000 rpm; brake mean effective pressure, 160 pounds per square inch; and oil-in temperature, 180° F

In the dust tests, the piston rings were run-in after which a 1-hour endurance run without dust was made as an oil-consumption check under the same conditions as those for the runs with dust. Dust was injected at the rate of 18 grams over a 10-minute period every ½ hour for 1½ hours. This period of injections was followed by 4½ hours of endurance without dust injection. Oil-weight readings were obtained every 15 minutes. The dust-injection equipment and dust were the same as those used in the single-cylinder-engine dust tests.

Dust tests were also performed on other engines of 1820- and 2600-cubic-inch displacement; the dust cycles were, however, somewhat different from that previously described. The dust was also slightly different. Cast-iron rings were assembled in both engines—in chrome-plated cylinders in the 1820-cubic-inch-displacement engine, and in nitrided cylinders in the 2600-cubic-inch-displacement engine.



(a) Rings held 10 minutes at each temperature.
(b) Rings held 6 to 8 hours at each temperature.

FIGURE 5.—Effect of piston-ring temperature on diametral tension.

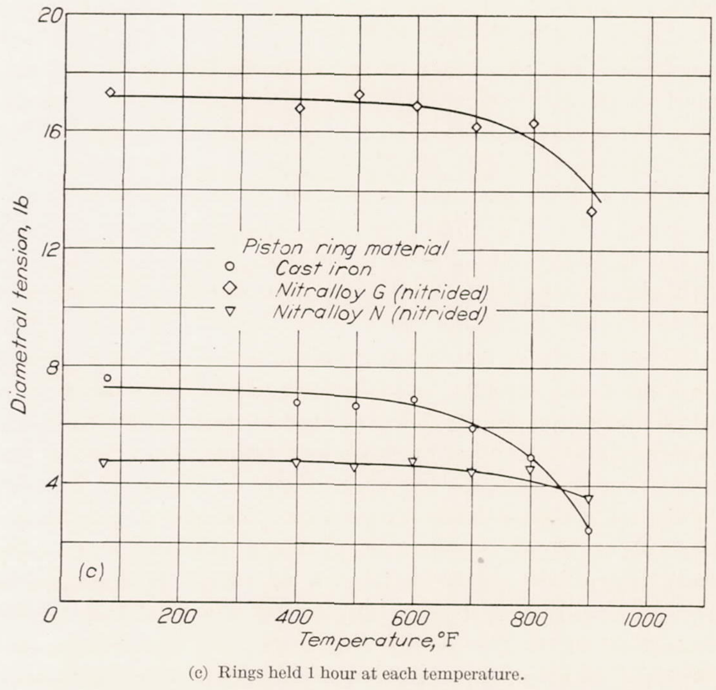
Multicylinder-engine investigations made by the Army Air Forces at Wright Field were in the form of service tests on three 1820-cubic-inch-displacement engines installed on an airplane operating from dusty air fields. Two of the engines were assembled with nitrided-steel cylinders and nitrided-steel ring assemblies, which included the high-unit-wall-pressure oil rings. The third engine was assembled with chrome-plated cylinders and the same ring assemblies. These tests were accelerated by operation of the engines at take-off power for longer than usual periods of time. Total operating time at completion of the tests was 137 hours including 171 take-offs.

RESULTS AND DISCUSSION

TEMPERATURE INVESTIGATION

The analysis of the temperature investigation of ring materials indicated that cast-iron rings did not have the desired physical characteristics under conditions of high operating temperatures. This indication is best shown by study of the results that follow.

Results of this investigation on a variety of piston-ring materials are shown in figure 5. A study of the curves shows



(c) Rings held 1 hour at each temperature.

FIGURE 5.—Concluded. Effect of piston-ring temperature on diametral tension.

that the nitrided-steel rings are more satisfactory on the basis of resistance to stress relief than the rings of other materials investigated at elevated temperatures. Loss of strength was particularly noticeable in the high-speed steel rings and resulted in near or complete collapse at the elevated temperatures.

The cast-iron rings showed a sharp break in the tension curve above 700° F (fig. 5(a)) for the 10-minute heating period. The 6- to 8-hour and the 1-hour heating-period tests (figs. 5 (b) and 5 (c), respectively) showed that the loss in tension, expressed as a percentage of the original tension, was appreciably less with nitrided steel than with cast iron. This difference in tension loss would indicate that prolonged high-temperature operation would have a greater adverse effect on the cast iron than on the nitrided steel.

SINGLE-CYLINDER-ENGINE INVESTIGATION

Compression-ignition-engine runs.—Because the nitrided-steel rings showed greatest resistance to high temperatures, it was considered that these rings would give the best service in an engine of high output. Rings with a nitrided case depth of 0.030 inch were installed in a used, nitrided-steel cylinder liner that was in good condition and, without an attempt to run the rings in, the load on the engine was slowly increased. Excessive breather smoke immediately developed and the rings and the liner were inspected. It was found that the edges of the rings had chipped and the small particles had scratched the nitrided-steel cylinder liner, the rings, and the piston; the small particles could be seen embedded in the aluminum piston at the end of the scratches. The sharp edges of a set of rings were removed by grinding with a small, high-speed emery wheel and the engine was reassembled after the cylinder liner was repolished with abrasive paper. A slightly greater load was obtained before failure and the parts were again inspected. The scratches resulting from the chipping of the rings were much less, but a microscopic examination of the ring faces indicated numerous points where "spot welding" between the ring face and the cylinder wall had taken place during the test.

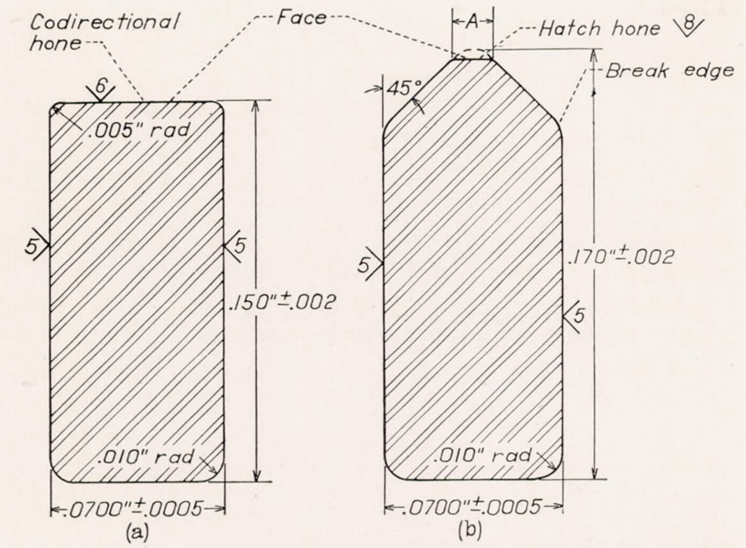
New rings were obtained with a nitrided-case depth of only 0.015 inch instead of the original depth of 0.030 inch. The sharp edges were removed from a set of the new rings and the rings were placed on a lapping jig that was made with ring grooves of the same dimensions as the piston-ring grooves. A strip of steel 0.015 inch thick was embedded in the groove and perpendicular to it to make the ends of the rings butt against the steel and to prevent the rings from turning in the groove. The rings had previously been fitted to the bore for the correct end gap. With the use of a 320-grit, silicon-carbide valve-grinding compound, the rings were lapped in a dummy cylinder by hand with a spiral motion until they showed good contact over the entire face of the ring. The grinding compound used was changed to a 400-grit, silicon-carbide valve-grinding compound, and the lapping was continued. They were finish-lapped with 500-grit aluminum oxide in the bore in which they were to be used. After the lapping operation, the rings were measured and assembled in the engine for testing. Again the engine was slowly brought up to operating conditions and operated at an indicated mean effective pressure of 150 pounds per square inch for 20 hours without a trace of breather smoke. At the end of this run the rings were inspected and measured; no chipping and very little wear were evident.

A new liner was then obtained with a surface finish of less than 3 microinches, rms, and a second set of rings of 0.015-inch-nitrided case was prepared by grinding off all sharp corners. The surface finish on the rings was poor, and they were therefore lapped with 500-grit aluminum oxide in the cylinder that was to be used. When the grinding marks had been removed, levigated alumina was substituted for aluminum oxide and the rings and the liner were lapped to a mirror finish. During the process there was no dimensional change in either the rings or the liner.

The engine was assembled, warmed up under power, and gradually brought up to an output of 240-pound-per-square-inch indicated mean effective pressure at an engine speed of 2000 rpm and a maximum cylinder pressure of 1250 pounds per square inch. The warm-up and the gradual application of the load required approximately 30 minutes. After 25 hours at this output, the rings were inspected and measured with a micrometer and only a trace of wear was discernible.

Under high-output conditions, the assembly of the cylinder and rings that had smooth initial surface finishes proved to be the most successful with respect to wear and blow-by. The attainment of smooth finishes permitted the use of a short run-in time without adversely affecting engine performance and wear. This investigation indicated the necessity for smooth surface finishes on both cylinder and rings for the combination of nitrided-steel rings and a nitrided-steel cylinder.

Preliminary spark-ignition-engine runs.—Nitrided-steel (Nitralloy G) rings (stock wedge-shaped compression-ring design) and a nitrided-steel cylinder were obtained for the runs. These rings had a considerably better surface finish than the original rings. The diametral tension of these rings was quite high (17.3 lb) with a resulting high unit wall pressure. The cylinder was a stock production item with a bore finish of 3 to 5 microinches, rms. The engine was run in for 1 hour and then operated 1 hour at a brake mean effective pressure of 200 pounds per square inch and an engine speed of 2200 rpm. In spite of the improved surface finish of the cylinder, severe scuffing was experienced. (Scuffing may be defined as an area rupture of metal surfaces.) The preparation of the surfaces and the fitting of new rings to the bore were repeated, as previously described, for the last run of the compression-ignition-engine series and, after a run-in period of 1 hour, a test run of 25 hours under the same conditions was completed. The surfaces were in good condition although there was excessive wear of the rings. Temperature tests on similar rings are included in the results of figure 5 (c) for Nitralloy G.



(a) Compression ring. 5/16-inch free gap.

(b) Oil ring; low unit wall pressure, 39 pounds per square inch; 1 1/16-inch free gap, A=0.020 inch: high unit wall pressure, 84 pounds per square inch; 1 5/8-inch free gap, A=0.014 inch.

FIGURE 6.—Nitrided-steel (Nitralloy N) rings for 6 1/8-inch bore. Approximate surface hardness, 1000 Vickers. V indicates surface finish in microinches, rms.

A second type of nitrided-steel ring made of Nitralloy N and shown in figure 6 was obtained for investigation. This type will be referred to as the "thin" ring. Most of the early runs using thin rings were conducted to determine the necessary piston design for use with thin rings. The progressive steps in this part of the research are shown in table I. Most runs were of 1-hour duration at a brake mean effective pressure of 204 pounds per square inch because this period was considered sufficiently long to determine the functioning of the rings and the piston. It will be noted that the run-in period for nitrided-steel piston rings consists of 1 hour rather than the usual 5 to 10 hours required for cast-iron rings.

After some of the initial problems of piston design and ring assembly were solved, runs for periods of 25 hours and longer were made to further check performance. These runs are listed as reference runs 9 to 11 of table I. Figures 7(a) and 7(b) are photographs of the piston and ring assembly after reference run 10 and are representative of the appearance of piston and rings after all three runs.

The final piston design was made from the results of the preliminary investigations. Before acceptance and incorporation in the final piston design, the following features, based on the runs just discussed, were considered:

1. Maximum length of cross head as limited by allowable dimensions. This feature is one of extreme importance inasmuch as length of cross head influences ring performance by effect on piston rocking.

2. Location of rings. The ring belt was located as far from the piston crown as practicable to protect the top ring from the high-temperature combustion gases.

3. Shape of piston. Piston shape was carefully chosen, from the results of the preliminary runs, to conform as nearly as possible to the cylinder bore at operating temperatures. This shape will insure against possibility of piston seizure because of thermal distortions.

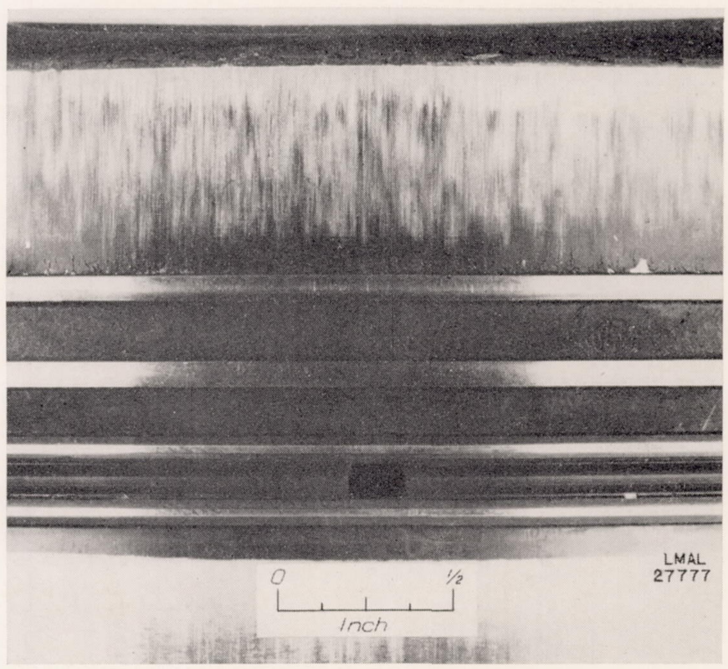
4. Location of oil-drain holes. For maximum effect of oil control, unrestricted passages must be provided and advantage taken of the inertia effect of the oil in the matter of scavenging the grooves.

5. Simplicity of ring assembly. The choice of the number of rings was made to obtain the least number of rings that would result in efficient operation. Two compression and two oil rings, all located above the piston pin, were the final result. The rings were of nominal rectangular cross section because this simple shape was easily machined.

The final piston design is shown in figure 8 and a new, complete piston assembly is shown in figure 9.

(a) Piston assembly.

FIGURE 7.—Appearance of piston and rings after 25 hours at 2200 rpm and 204 pounds per square inch brake mean effective pressure. Reference run 10.



(b) Nitrided-steel rings.

FIGURE 7.—Concluded. Appearance of piston and rings after 25 hours at 2200 rpm and 204 pounds per square inch brake mean effective pressure. Reference run 10.

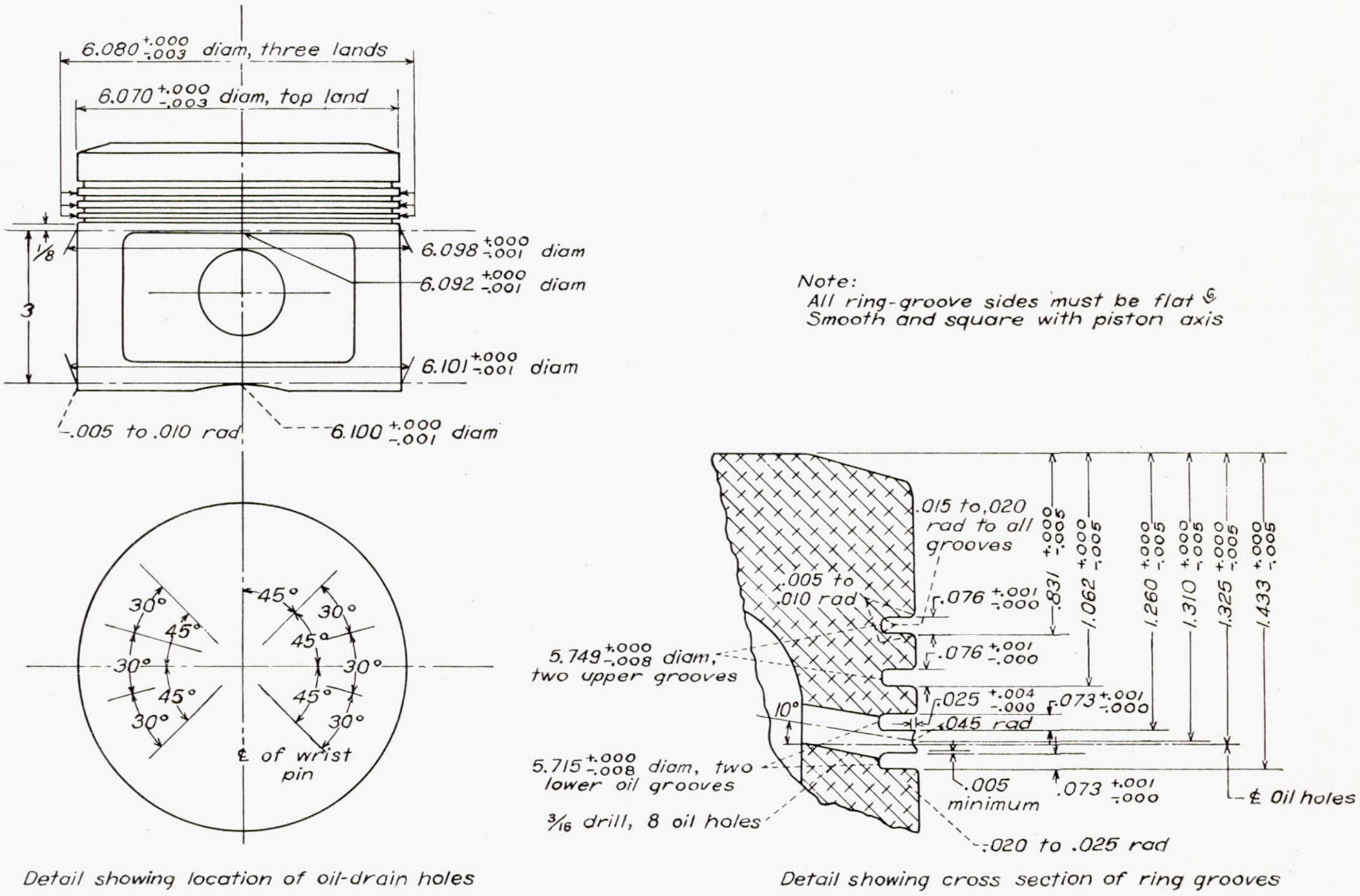


FIGURE 8.—Final piston design for nitrided-steel rings. Bore, 6⅛ inches. V indicates surface finish in microinches, rms. (All dimensions in inches.)

FIGURE 9.—New piston assembly.

In order to check the final piston and ring design, a 150-hour endurance run in accordance with the program of the appendix was performed. This run was a check on both piston and ring performance under the wide range of conditions specified by the endurance run. General engine performance with respect to ring and cylinder wear, blow-by, and power was excellent. Lubricating-oil control (that is, specific oil consumption) was poor. Results are shown in table II and in figure 10. Figure 11 shows the piston and ring assembly after the 150-hour-endurance run.

The excellent condition of ring and cylinder rubbing surfaces indicated that these surfaces were compatible with each other. The extremely low wear of rings as indicated from the low weight loss was also an indication that very successful operation with this ring assembly is possible. As can be noted in table II, a maximum ring-weight loss of approximately 0.6 percent was recorded in this 150-hour endurance run.

Results of runs on the stock cast-iron-ring assembly in a stock nitrided-steel cylinder at a brake mean effective

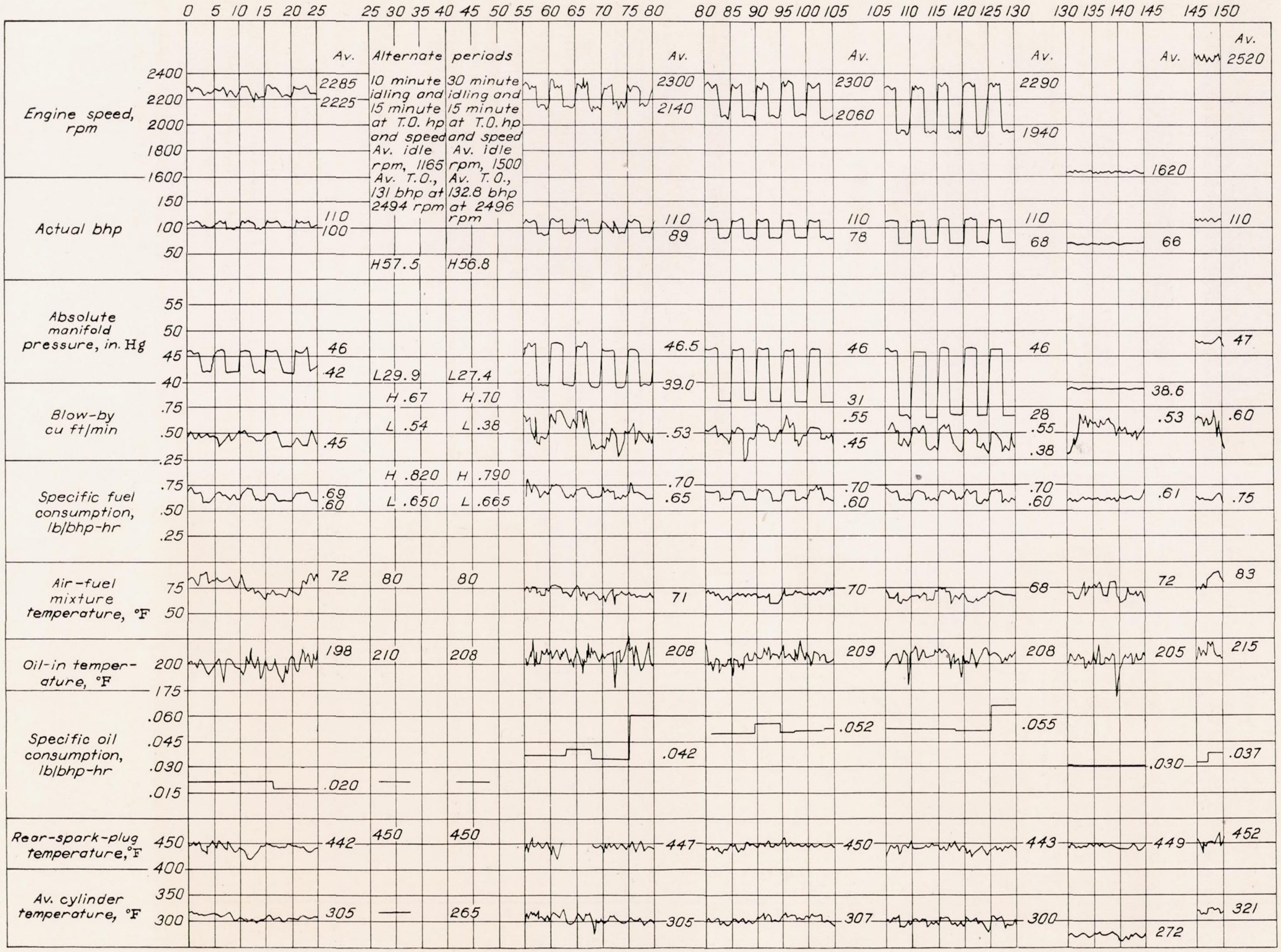
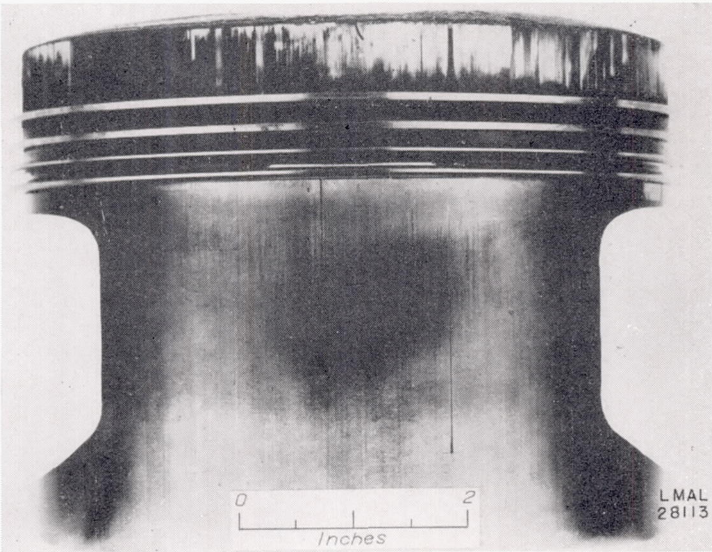


FIGURE 10.—150-hour endurance run of nitrided piston rings in accordance with the program of appendix. Crankcase, multicylinder converted to single-cylinder; fuel, 100 octane; oil, Navy 1120; average barometer, 30.07 inches mercury. (T. O., take-off; H, high; L, low)

pressure of 250 pounds per square inch are shown in table III, reference run 16, and in figure 12. Conditions with respect to speed, output, oil temperature, and cylinder temperature were made extremely severe. It will be noted

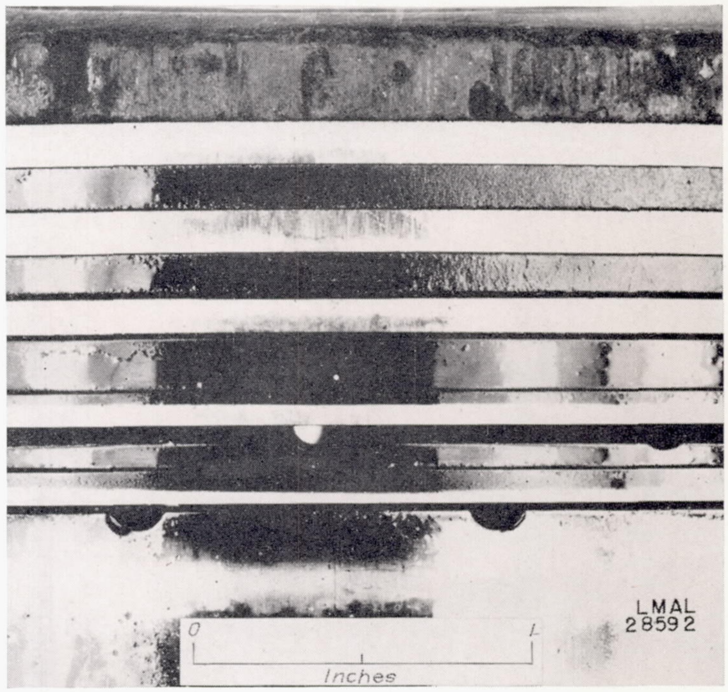


(a) Piston assembly.

FIGURE 11.—Appearance of piston and rings after 150-hour endurance run. Reference run 14.

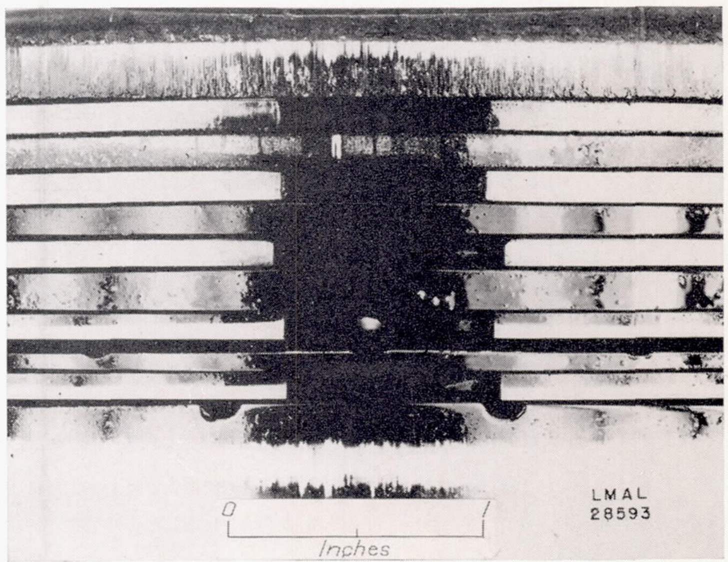
(b) Nitrided-steel rings.

FIGURE 11.—Concluded. Appearance of piston and rings after 150-hour endurance run. Reference run 14.



(a) View 180° from ring gap.

FIGURE 12.—Cast-iron rings after 9½ hours at 2500 rpm and brake mean effective pressure of 250 pounds per square inch. Reference run 16.

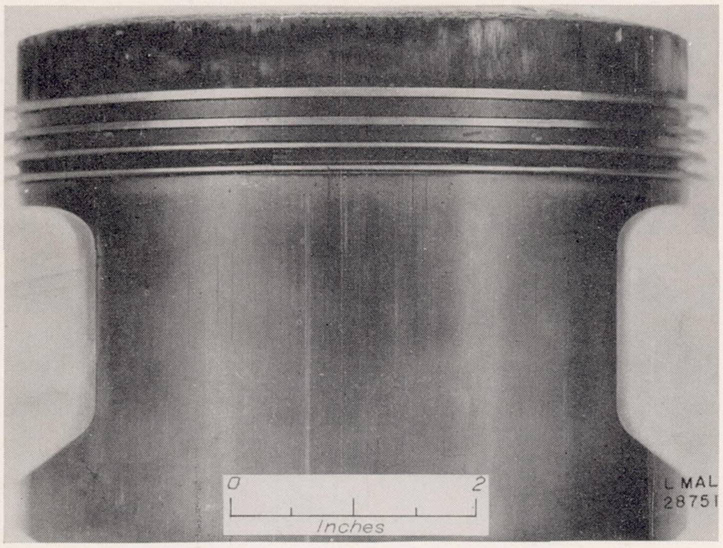


(b) View at gap.

FIGURE 12.—Concluded. Cast-iron rings after 9½ hours at 2500 rpm and brake mean effective pressure of 250 pounds per square inch. Reference run 16.

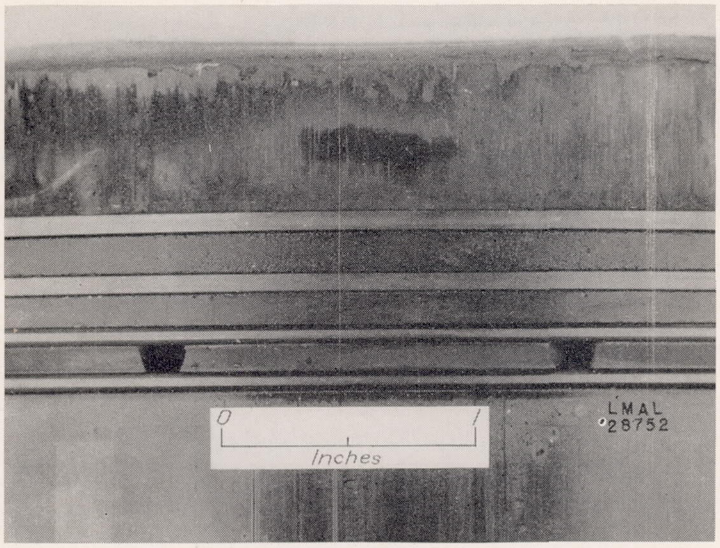
that oil control was poor from the start of the run, possibly indicating that the rings had scuffed and feathered during run-in. Wear of the rings was also very high. The rings after this run were scuffed, scored, and feathered. Table III shows that the top rings had lost tension; this loss was manifested by a loss in free gap of the rings.

Reference runs 17 to 19 of table III show results of runs of the piston and ring assembly from reference run 14 (table II) under the conditions of reference run 16 (table III). These runs were made as a comparative check, at these conditions, of nitrided-steel and cast-iron rings. The run-in period of 1 hour was still maintained at this high output. It will be noted that weight losses are low, when considering the extreme severity of conditions. Figure 13 is included to show the excellent condition of the piston and rings after these runs even though two of the runs resulted in exhaust-valve failures and much of the loose valve-steel particles passed by the faces of the rings and embedded themselves in the piston skirt.



(a) Piston assembly.

FIGURE 13.—Appearance of piston and rings after high-output runs. Reference run 19.



(b) Nitrided-steel rings.

FIGURE 13.—Concluded. Appearance of piston and rings after high-output runs. Reference run 19.

The last reference run listed on table III was successful with respect to wear and general engine performance characteristics. Oil control, however, was still relatively poor although the oil consumption was constant, a condition that is rather unusual in high-output runs of this duration. The large increase in oil consumption for cast-iron rings is shown in reference run 16 of table III.

Because the previously described runs of the nitrided-steel rings indicated that the oil rings (fig. 6, (b) low-unit-wall-pressure oil rings, 39 lb/sq in.) were unsatisfactory on the basis of oil control, runs were made with rings of lesser face width, 0.010 inch, and of diametral tension that was the same as for the original rings. This decrease in face width with the same diametral tension resulted in an approximate initial unit wall pressure of 78 pounds per square inch. The runs indicated that these rings decreased oil consumption at least 50 percent. Rings were accordingly procured that obtained a high unit wall pressure by an increase of diametral tension and a decrease in face width. It was preferred to obtain the high unit wall pressure in this manner rather than by a large decrease in face width alone, because a ring of large face width will have a lower percentage change of face width for an equal amount of wear than a ring of small face width.

Final spark-ignition-engine runs.—The final design of nitrided-steel oil rings was one that had a free gap of approximately $1\frac{5}{8}$ inches rather than the free gap of approximately $1\frac{1}{16}$ inches in the original rings. A decrease in face width from 0.020 to 0.014 inch was also made at this time. The increase in free gap resulted in a change of diametral tension from 6.3 to 9.4 pounds. Values of initial unit wall pressure are 39 and 84 pounds per square inch. Oil rings with both low and high initial unit wall pressure are shown in figure 6.

The results of two runs with these oil rings are presented in table IV. Reference run 31 from table IV can be directly compared with reference run 19 of table III. These two runs furnish a direct comparison of performance with oil rings of high and low initial unit wall pressure. The oil consumption decreased from 0.022 pound per brake horsepower-hour in reference run 19 to 0.013 pound per brake horsepower-hour in reference run 31. In both runs the wear of the compression rings and the cylinders was considered low. As can be seen from a comparison of oil ring wear in reference runs 30 and 31, the wear was relatively high until the rings had seated themselves and apparently reached a stable condition of rate of wear and oil control. In reference run 30, oil consumption decreased from 0.020 to 0.008 pound per brake horsepower-hour. This decrease would apparently correspond to the seating period of the rings inasmuch as reference run 31 showed a constant oil consumption of 0.013 pound per brake horsepower-hour during the entire run.

Figure 14 shows the photograph of the piston and ring assembly after reference run 31. It must be noted that the top oil ring was cold-stuck and the second compression ring was tight at the gap. No ring sticking was apparent in any of the previous high-output runs.

In all runs using oil rings of the high unit wall pressure, the initial wear was relatively high, which resulted in a fairly large increase in face width.

After satisfactory operation with respect to wear and lubricating-oil consumption had been achieved in nitrided-steel cylinders, single-cylinder-engine runs of the nitrided-steel rings in a porous chrome-plated, straight-bore cylinder were conducted. The rings from reference run 31 of table IV were installed directly in a porous chrome-plated, straight-bore cylinder barrel and were operated for three runs according to the operating conditions indicated in runs 1 to 3 of table V.

Results were good with respect to wear and oil control in each of these three runs. Figure 15 shows the piston and ring assembly after runs in the chrome-plated barrel.

In order to investigate the reason for successful performance of the chrome-plated cylinder, porosity was measured after run 3 of table V in two locations. Some replicas were taken in ring travel and others were taken in the section of cylinder above ring travel. Measurements in the section of cylinder above ring travel are considered representative of the porosity before the runs. Porosity range above ring travel was 50 to 65 percent and in ring travel was 30 to 40 percent. Figure 16 shows typical plan-view photomicrographs at 100 magnification of the chrome-plated surface above and in ring travel after the run.

The set of rings used in the runs reported in table V had completed 155 hours of operation after run 3. Figure 17 shows the effect of running time on this particular set of nitrided-steel rings. The curves of figure 17 indicate that the rings had apparently reached a constant rate of wear. This

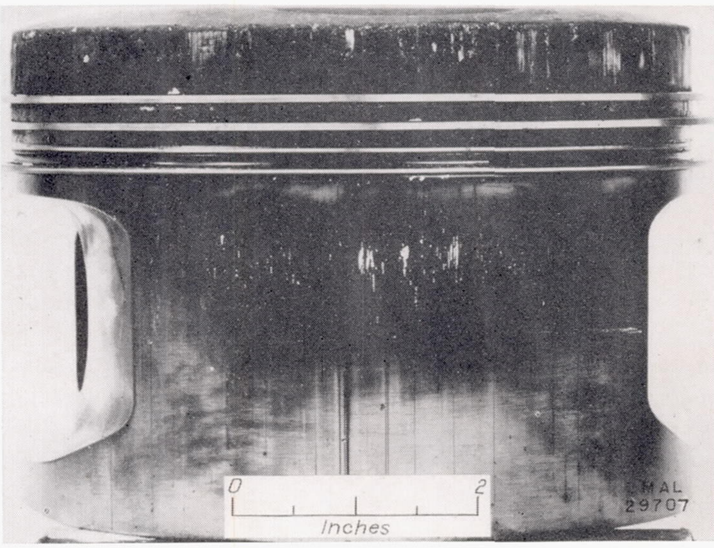
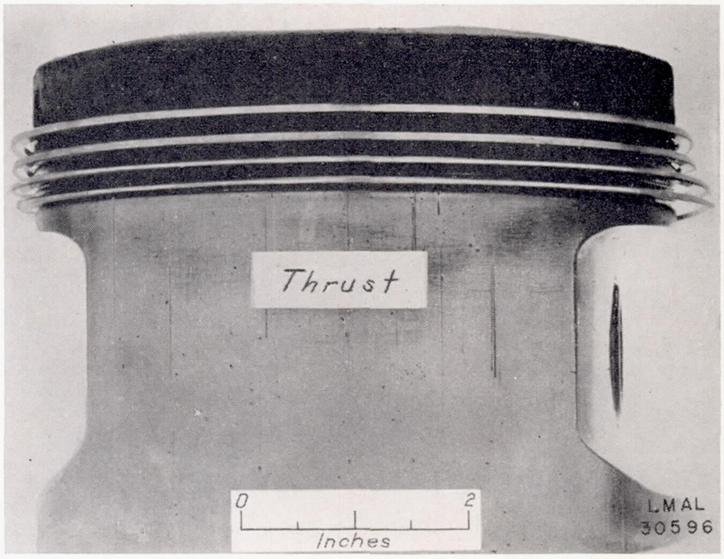
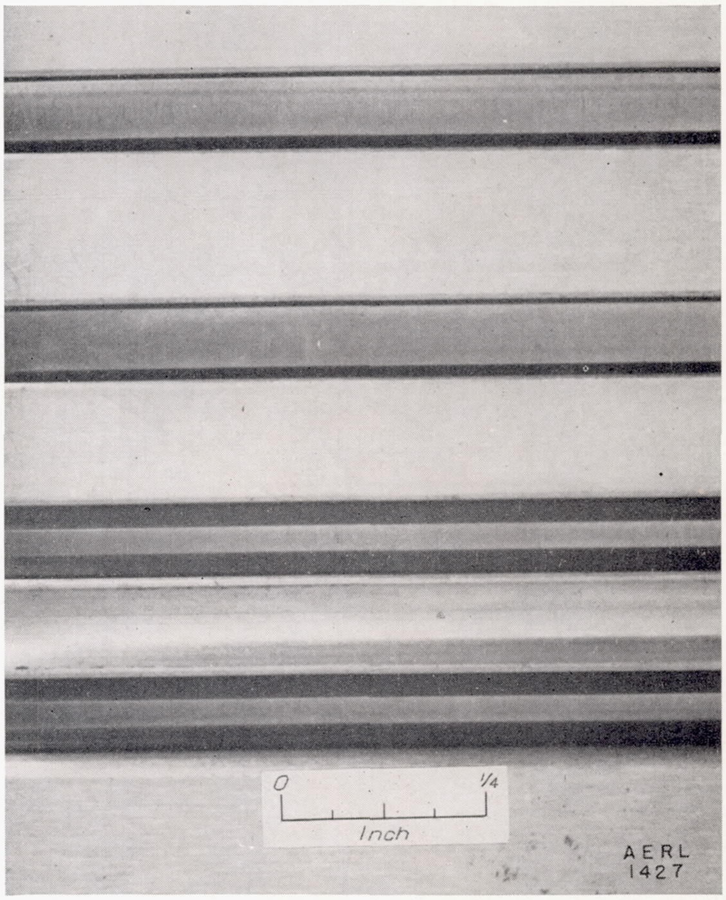


FIGURE 14.—*Piston assembly after high-output runs. High-pressure oil rings. Major thrust face. Reference run 31.*



(a) Piston assembly. Major thrust face.

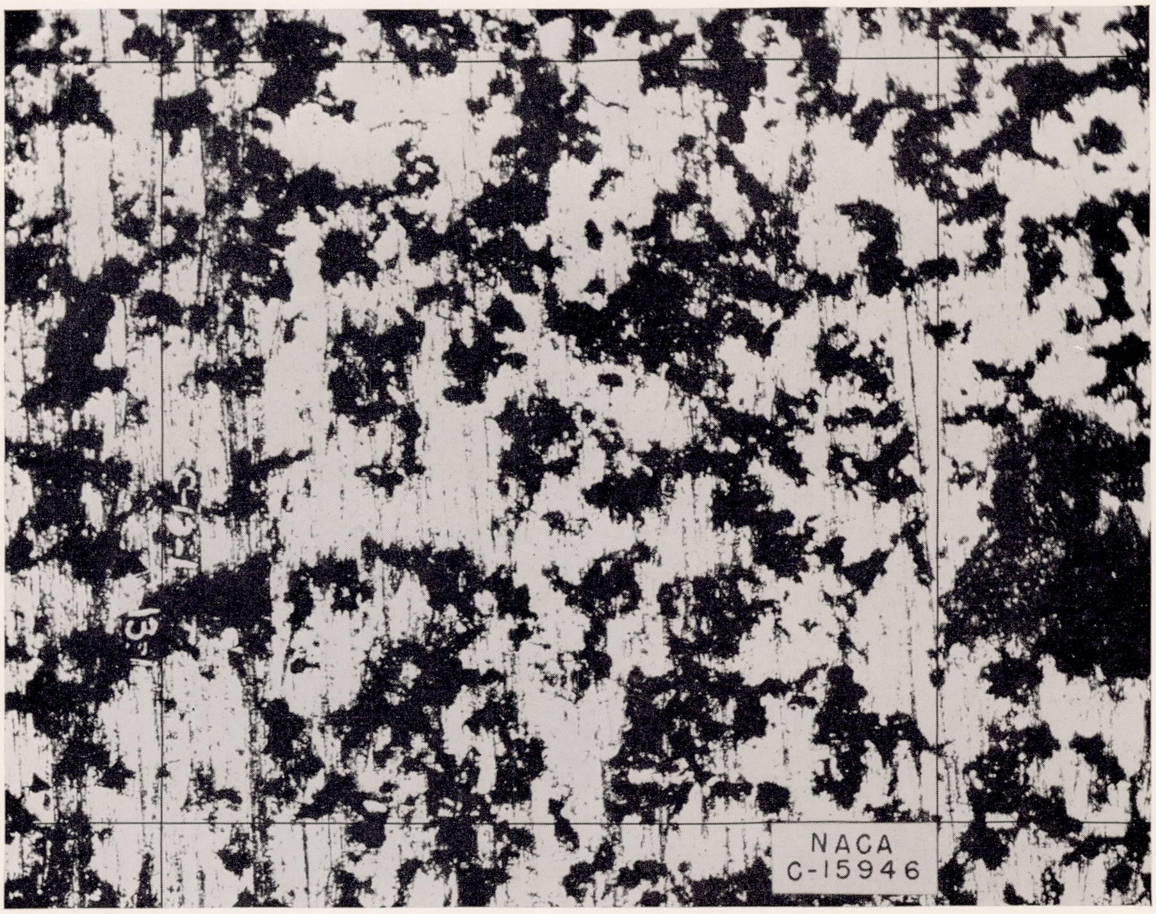
FIGURE 15.—Appearance of piston and rings after runs in porous chrome-plated cylinders. Run 3, table V.



(b) Nitrided-steel rings. (Rings assembled on new piston for photographing.)

FIGURE 15.—Concluded. Appearance of piston and rings after runs in porous chrome-plated cylinder. Run 3, table V.

constant rate of wear is low, as can be computed from the weight loss during the last three runs. Weight loss in run 3 was less than one-quarter of 1 percent of the initial weight of the ring.



(a) Above ring travel.

FIGURE 16.—Plan view of porous chrome-plated cylinder after run. Porosity, 52 percent. X100

(b) In ring travel.

FIGURE 16.—Concluded. Plan view of porous chrome-plated cylinder, after run. Porosity, 40 percent. X100

It would appear from the results of these tests that the most satisfactory combination of rings and cylinders might be that of the nitrided-steel rings in the porous chrome-plated cylinders. This assembly should result, after seating and complete compatibility of the rubbing surfaces has been attained, in a very stable assembly with respect to both rate of wear and oil control. Dust tests of this assembly should be run, however, before a final recommendation can be made.

Dust tests.—Results of the single-cylinder-engine dust tests showed that the nitrided-steel ring assembly in nitrided-steel cylinders resulted in acceptable oil control through the fourth dust cycle (fig. 18). The oil consumption after the fifth dust cycle was approximately that obtained by the stock cast-iron rings after the first dust cycle. The test of the cast-iron-ring assembly was discontinued after the first dust cycle because the slope of the oil-consumption curve indicated that the assembly was wearing out rather than seating in, and consequently no better oil control than that after the first dust cycle (which was unacceptable at the conditions of

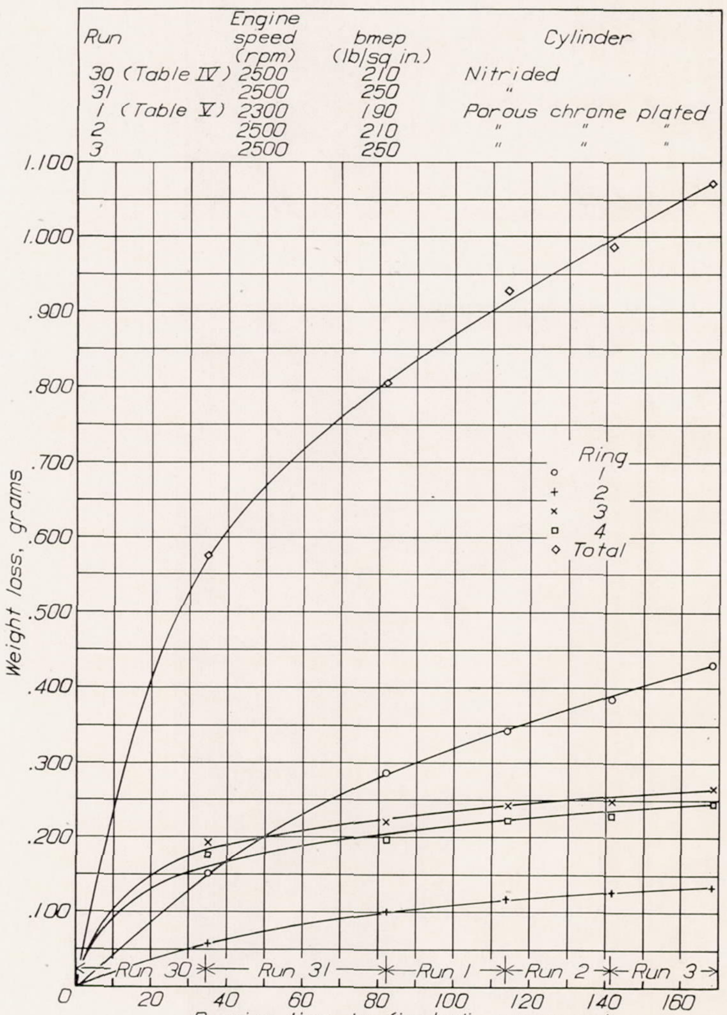


FIGURE 17.—Ring wear of one of the nitrided-steel ring assemblies. Rings cleaned and weighed after each run, then reassembled for next run. Reference runs 30 and 31 from table IV; runs 1, 2, and 3 from table V.

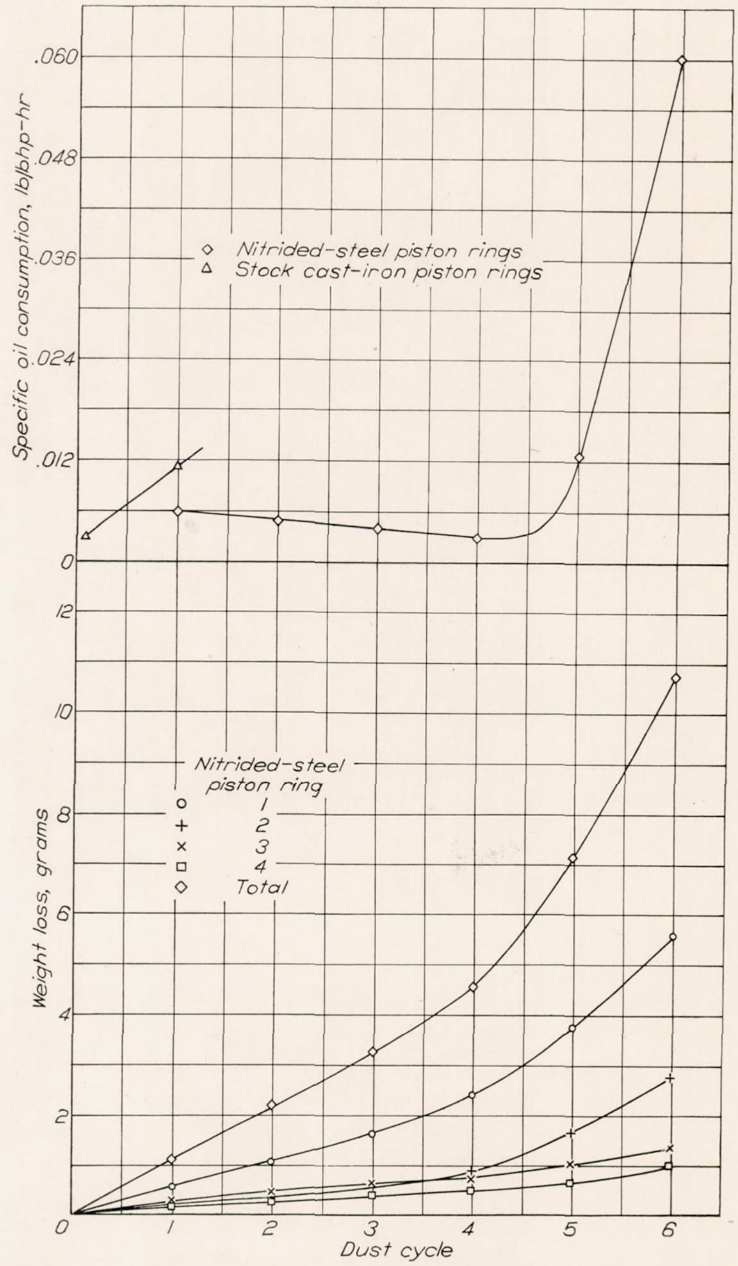


FIGURE 18.—Effect of dust on oil control and wear. Nitrided-steel ring assembly; single-cylinder engine. Data from Bureau of Aeronautics, Aeronautical Engine Laboratory.

this test) could be expected. The wear curves in figure 18 show that oil control was satisfactory as long as the rate of wear of the rings was constant; that is, as soon as the slopes of the curves increased, the oil consumption increased.

Surface finishes.—Surface finishes of 2 to 8 microinches, rms, on the face of the nitrided-steel rings were investigated in preliminary runs and the final result was a ring finished by honing on the face or outside diameter to 5 to 8 microinches, rms.

It was found that for nitrided-steel cylinders a cross-hatch honed finish of 4 to 6 microinches, rms, was suitable for use with the nitrided-steel ring assembly after a number of runs covering a range of surface finishes in the cylinder of 1 to 6 microinches, rms.

Surface finish on the sides of the rings and on the piston-ring lands was set at 5 microinches, rms, in the belief that this finish (5 microin., rms) would be adequately smooth to prevent ring sticking. No runs were made with this finish as a variable, however, because very few of the runs on the nitrided-steel ring assembly showed any signs of ring sticking.

MULTICYLINDER-ENGINE INVESTIGATION

The multicylinder-engine run performed by the Bureau of Aeronautics on an engine of 1820-cubic-inch displacement at take-off power and speed of the original, thin, nitrided-steel ring assembly with standard low-unit-wall-pressure oil rings

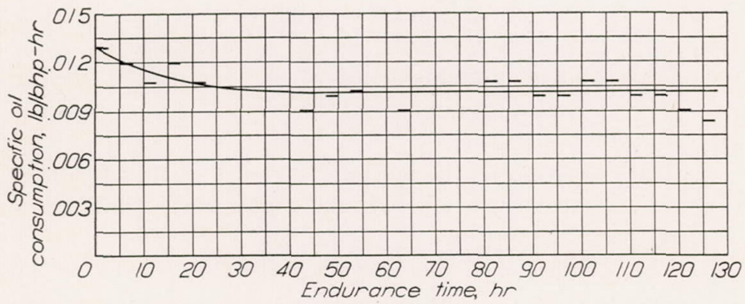


FIGURE 19.—Effect of running time on oil control with nitrided-steel ring assembly. 150-hour endurance run of 1820-cubic-inch displacement engine. All points at normal rated power and speed (1000 bhp at 2300 rpm). Data from Bureau of Aeronautics, Aeronautical Engine Laboratory.

was terminated because of failure of the master-rod bearing after approximately 30 hours at take-off power. Results of this run were good with respect to wear in spite of a complete bearing failure that covered all rubbing surfaces with bearing material and contaminated the lubricating oil. The oil control in this run, however, was relatively poor (specific oil consumption, 0.030 lb/bhp-hr). The trend of oil consumption decreased throughout the run.

Additional multicylinder-engine runs were made by the Bureau of Aeronautics, Navy Department, on the nitrided-steel ring assembly with the oil ring of high unit wall pressure after the NACA single-cylinder-engine runs had shown this assembly to be acceptable with respect to oil control. These single-cylinder-engine runs have been previously described and discussed.

A 150-hour endurance run in accordance with the program of the appendix was performed by the Bureau of Aeronautics, Navy Department, with nitrided-steel ring assemblies including the high-unit-wall-pressure oil rings. This run resulted in good performance with regard to wear and oil control. Specific oil consumption at normal rated power and speed (1000 hp at 2300 rpm) decreased from approximately 0.013 to 0.010 pound per brake horsepower-hour after approximately 40 hours of the endurance run had been completed (fig. 19). Average specific oil consumption at normal rated power and speed during the rest of the endurance run averaged 0.010 pound per brake horsepower-hour. The rings showed no tendency toward sticking. No ring breakage was encountered even though overspeed dive tests were run at 3100 rpm. The general operating characteristics were normal. Specific oil consumption at normal rated power and speed after the overspeed dives was 0.007 pound per brake horsepower-hour. The specific-oil-consumption check completed the 150-hour endurance run. The value of 0.007 pound per brake horsepower-hour is considered satisfactory even though it is higher than is usually obtained with the standard cast-iron ring assembly under these conditions.

The oil flow (total and power section) was accidentally more than 45 percent above the prescribed maximum for most of the run, and runs on this engine showed that decreasing the total oil flow (at normal rated power and speed) 22 percent decreased the specific oil consumption 13 percent. The 22-percent decrease in rate of oil flow resulted in a rate of flow 18 percent higher than the recommended maximum. It is possible that normal oil flow would have resulted in still lower specific oil consumption.

Results of the multicylinder-engine dust tests conducted by the Bureau of Aeronautics indicated that the nitrided-steel ring assemblies were very successful with respect to oil control (fig. 20). Wear and specific oil consumption started to become excessive only after an appreciable number of dust cycles. The top compression ring in most of the cylinders was excessively worn before any great effect on oil consumption was evident. The slope of the oil-consumption curve was not very great even at the end of the seventh dust cycle.

The trends of oil consumption for standard piston assemblies of cast-iron rings in engines of 1820- and 2600-cubic-inch displacement are shown in figures 21 and 22, respectively. Both curves show that oil consumption increased very rapidly during the first two dust cycles and became exces-

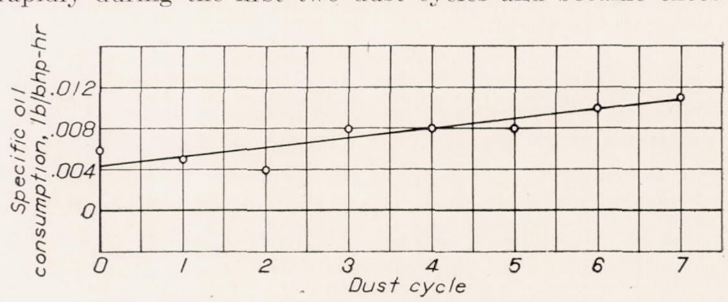


FIGURE 20.—Effect of dust on oil control. Nitrided rings in nitrided cylinders in 1820-cubic inch displacement engine. Data from Bureau of Aeronautics, Aeronautical Engine Laboratory.

sive after the first cycle. It can be seen that this rapid increase is true in both porous chrome-plated cylinders and in nitrided-steel cylinders.

When figure 20 is compared with figures 21 and 22, it is apparent that the nitrided-steel rings should provide acceptable oil control for much longer periods of time than the cast-iron rings, inasmuch as the oil-consumption curve for nitrided-steel rings shows a much more gradual increase than the curve of the cast-iron rings. Although it is true that no quantitative comparisons should be made of figures 20 to 22 because the dust cycles are somewhat different, it is believed that the indicated trends in the three tests can be used on a comparative basis.

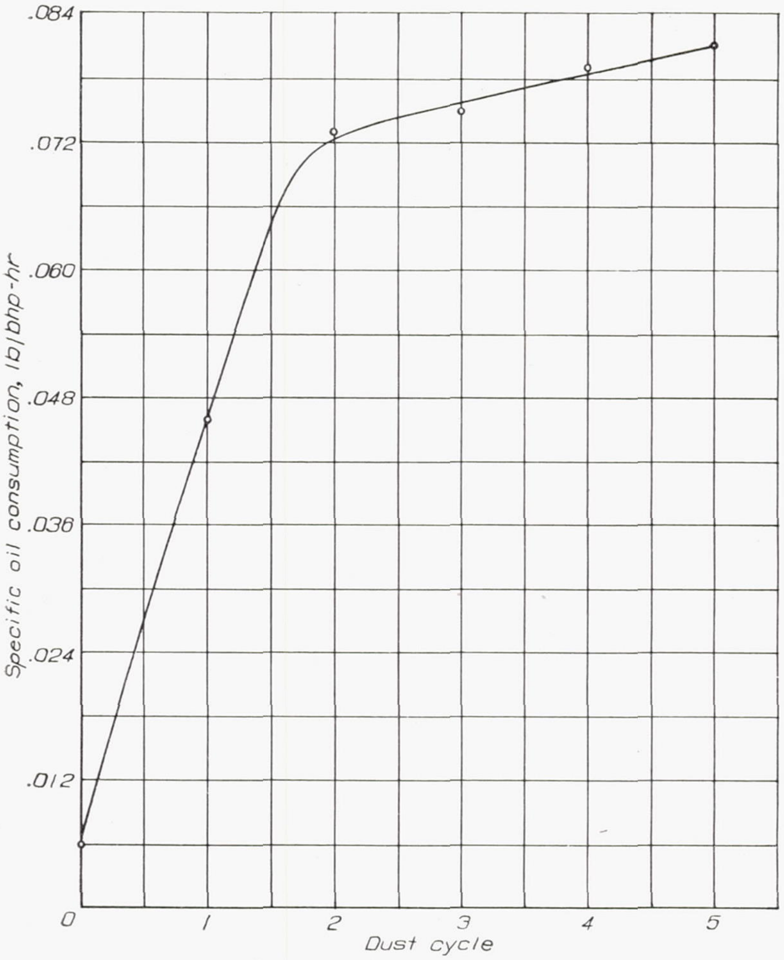


FIGURE 21.—Effect of dust on oil control. Stock cast-iron rings in porous chrome-plated cylinders, 1820-cubic-inch displacement engine. Data from Bureau of Aeronautics, Aeronautical Engine Laboratory.

Multicylinder-engine flight tests of nitrided-steel rings assembled in both nitrided-steel and chrome-plated cylinders as performed by the Materiel Command, Army Air Forces, resulted in good performance of the rings in both types of cylinder with respect to wear and oil control. All rings were free after these tests. Wear was low, considering severity of conditions and amount of operation in dusty atmospheres. The total operating time of 137 hours resulted in more severe operation than this amount of time normally represents because a large number of take-offs (171) were included. Condition of the rubbing surfaces was considered excellent in all three engines investigated. The condition of the pistons, especially the skirts, was very good and the acceptable oil control proved that no bottom ring is necessary in this piston design.

SUMMARIZING REMARKS

The nitrided-steel ring assembly under dust conditions is more successful on the basis of oil control than the cast-iron, taper-faced ring assembly because in the nitrided-steel ring assembly a large part of the oil-control function is performed by the oil rings and little oil control is required from the compression rings. Wear on the compression rings, consequently, has little effect on oil control.

From the results of the runs reported, further investigation of the combination of nitrided-steel rings in porous chrome-plated cylinders should be made because this combination offers attractive possibilities for use. The use of worn cylinders that have been reclaimed by porous chrome plating is attractive from the salvage viewpoint, and any ring assembly that can be successfully operated in these cylinders should be completely engine-tested. These engine tests should include dust tests to check abrasion resistance.

SUMMARY OF RESULTS

Based on the data from the single-cylinder and multicylinder engines, the following results were obtained:

1. Performance characteristics of the nitrided-steel ring assembly of final design were excellent with respect to wear and abrasion resistance.
2. The condition of the rubbing surfaces of nitrided-steel rings and chrome-plated cylinders indicated that these surfaces were compatible.
3. The nitrided-steel ring assembly of final design resulted in average specific oil consumptions of 0.013 and 0.011 pound per brake horsepower-hour at a brake mean effective pressure of 250 pounds per square inch and an engine speed of 2500 rpm in the single-cylinder engine. The average

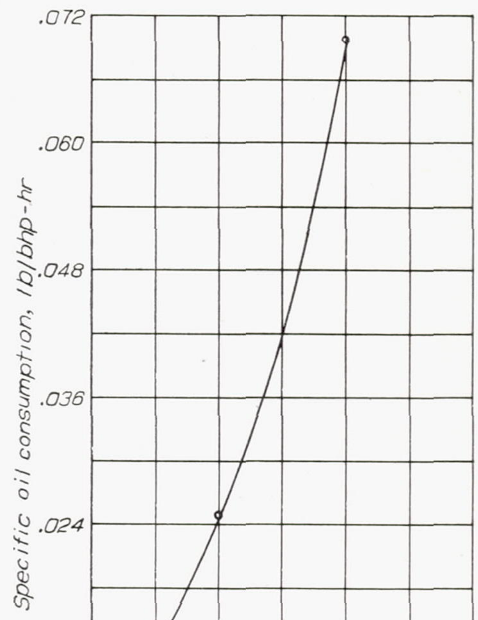


FIGURE 22.—Effect of dust on oil control. Stock cast-iron piston rings in nitrided-steel cylinders, 2600-cubic-inch displacement engine. Data from Bureau of Aeronautics, Aeronautical Engine Laboratory.

specific oil consumption in the 1820-cubic-inch displacement engine was 0.010 pound per brake horsepower-hour at normal rated power and speed (1000 hp at 2300 rpm). In all these cases, the trend of oil consumption was constant or decreasing.

4. Under dust-test conditions, acceptable oil control could be obtained three to four times longer with the nitrided-steel ring assembly than with the stock cast-iron rings.

5. Resultant wear of the cylinder barrels was very low in all runs.

6. Good resistance to ring breakage, at the severe overspeed-dive condition of 3100 rpm on an 1820-cubic-inch displacement engine, was exhibited by the nitrided-steel ring assembly.

7. The use of a shorter run-in time (1 hr) was found to be possible with the nitrided-steel ring assembly.

CONCLUSION

Based on the data from the single-cylinder and multicylinder engine runs, the following conclusion may be drawn:

The nitrided-steel ring assembly in either nitrided-steel or porous chrome-plated cylinders should be very desirable for use in high-output aircraft engines.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, *April 1, 1944*.

APPENDIX

150-HOUR ENDURANCE RUN PROGRAM

The following program represents the steps followed in the 150-hour endurance runs on both the multicylinder and single-cylinder engines. The normal rated power and speed for the multicylinder engine were 1000 brake horsepower at 2300 rpm and for the single-cylinder engine were 111 brake horsepower at 2300 rpm. The take-off power and speed for the multicylinder engine were 1200 brake horsepower at 2500 rpm and for the single cylinder engine were 133 brake horsepower at 2500 rpm.

1. Twenty-five hours of alternate periods of 2½ hours each at normal rated power and speed and at 90-percent normal rated power and 97-percent normal rated speed

2. Fifteen hours of alternate periods of 5 minutes at take-off power and speed and 10 minutes at as low an idling speed as practicable

3. Fifteen hours of alternate periods of 5 minutes at military rated power and speed (or, in the absence of a military rating, at take-off power and speed), and 10 minutes at a selected speed between 55 and 65 percent of that speed without change of propeller pitch

4. Twenty-five hours of alternate periods of 2½ hours each at normal rated power and speed and at 80-percent normal rated power and 93-percent normal rated speed

5. Twenty-five hours of alternate periods of 2½ hours each at normal rated power and speed and at 70-percent normal rated power and 89-percent normal rated speed

6. Twenty-five hours of alternate periods of 2½ hours each at normal rated power and speed and at 60-percent normal rated power and 84-percent normal rated speed

7. Fifteen hours at 60-percent normal rated power and 70-percent normal rated speed

8. Five hours at normal rated power and 110-percent normal rated speed.

The following runs were performed on the multicylinder engine only:

9. Fifty dives at 120-percent normal rated speed

10. Fifty dives at rated dive overspeed (3000 rpm)

11. Fifty dives at rated dive overspeed plus 100 rpm (3100 rpm)

REFERENCE

1. Englisch, C.: Messgerat zur Bestimmung des radialen Anpressdruckes von Kolbenringen. Auto. tech. Zeitschr., Jahrg. 43, Nr. 2, Jan. 25, 1940, pp. 42–44. (Tech. Trans. No. 129, R. A. Castleman.)

TABLE I—SUMMARY OF SINGLE-CYLINDER-ENGINE RUNS FOR NITRIDED-STEEL RINGS WITH NITRIDED-STEEL CYLINDERS

[Oil, Navy 1120; spark timing, 20° B. T. C.; power-section oil flow, 19.5 pounds per minute at 2200 rpm; limiting temperatures: rear spark-plug bushing, 450° F; rear center of cylinder, 350° F]

NACA reference run	Description	Conditions						Results					Remarks
		Run-in (hr)	Endurance (hr)	Speed (rpm)	bmep (lb/sq in.)	imep (lb/sq in.)	Oil-in temperature (°F)	Ring wear			Blow-by (cu ft/min) (a)	Specific oil consumption (lb/bhp-hr)	
								Weight loss (gram)	Percentage of original ring weight	Percentage of total loss			
1	Piston, stock. Rings, stock cast-iron; 3 compression, 3 oil. Cylinder, stock, honed finish.	6	1	2200	204	240	167	--------	--------	--------	0.36	--------	Oil control good; incipient scuffing of top ring.
2	Piston, stock except compression ring belt. Rings, 2 nitrided compression, 3 stock cast-iron oil. Cylinder, stock, polished with abrasive paper.	1	1	2200	204	240	167	0.032 .128	0.13 .52	--------	--------	--------	Oil control poor; more cross-head required on piston; second-ring wear high.
3	Same type assembly as reference run 2 except bottom ring inverted to scrape down. Cylinder, reference run 2, repolished.	1	1	2200	204	240	166	0.090 .048	0.36 .20	--------	0.42	--------	Oil control slightly improved; scuffing still present.
4	Piston, new design. Rings 4 same type, nitrided; 2 as compression, 2 as oil. Cylinder, same as reference run 3, repolished.	1	1	2200	204	240	177	--------	--------	--------	0.33	--------	Rings too high; bearing area on piston insufficient; redesign of piston required; oil control fair.
5	Piston, stock. Rings, stock cast-iron; 3 compression, 3 oil. Cylinder, stock, polished with abrasive paper.	6	1	2200	204	240	181	0.246 .018 .015 .010 .013 --------	0.56 .04 .03 .02 .03 --------	61.6 4.5 3.8 2.5 3.3 --------	0.89	--------	Oil control good; incipient scuffing of top ring.
6	Piston, new design. Rings, nitrided; 2 compression, 2 oil (low pressure).[b] Cylinder, same as reference run 4, without repolishing.	1	1	2200	204	239	175	0.011 .026	0.04 .11	--------	0.32	--------	Oil control improved but still insufficient; condition of piston, rings, and cylinder good.
7	Piston, from reference run 6 with high spots on skirt removed. Rings, nitrided; 2 compression, 2 oil (low pressure).[c] Cylinder, from reference run 6 without repolishing.	1	1	2200	204	238	172	0.012 .004	0.05 .02	--------	0.32	--------	Oil control fair; rings too high; skirt now satisfactory.
8	Piston, new design, ring grooves lowered. Rings, nitrided; 2 compression, 2 oil (low pressure).[b] Cylinder, honed (coaxial).	1	1	2000	204	238	172	0.017 .006	0.07 .02	--------	0.38	--------	Oil control poor.
9	Piston, from reference run 8, hand honed to fit choke. Rings, nitrided; 2 compression, 2 oil (low pressure).[c] Cylinder, from reference run 8, repolished.	1	25	2200	204	235	177	0.043 .024 .051 .062	0.17 .10 .19 .23	23.9 13.5 23.1 34.5	0.33	0.026 to .020 [a] .023	Oil control fair; general condition good.
10	Piston, machined to shape of one in reference run 9. Rings, 2 compression, 2 oil (low pressure).[c] Cylinder, stock, honed finish.	1	25	2200	204	235	176	0.046 .009 .005 .008	0.19 .04 .02 .03	67.6 13.0 7.8 11.7	0.33	0.048 to .028 [a] .034	Oil control poor; condition of piston, rings, and cylinder satisfactory.
11	Continuation of reference run 10.	½	27	2200	204	237	179	0.059 .019 .013 .012	0.24 .07 .05 .05	57.1 18.6 12.2 12.1	0.34	0.046 to .091 [a] .070	Oil control poor; general conditions very good.
12	Piston, stock. Rings, stock cast-iron; 3 compression, 3 oil. Cylinder, same as in reference run 5, repolished.	6	2	2200	204	239	178	0.041 -------- -------- -------- .001 .003	0.10 -------- -------- -------- .02 .06	--------	0.95	--------	Oil control good, general conditions good.
13	Piston, more skirt taper to increase clearance in choke, less and larger drain holes. Rings, nitrided; 2 compression, 2 oil (low pressure).[c] Cylinder, stock, honed finish.	1	1	2500	206	244	197	0.089 .033 .031 .027	0.36 .13 .12 .10	49.6 18.1 17.4 14.9	0.50	--------	Oil control fair; general conditions good.

[a] Average value.
[b] Lowest pressure oil rings: radial depth, 0.150 in.; free gap, approx. 1 1/16 in.; initial unit wall pressure, approx. 28 lb/sq in.
[c] Standard low-pressure oil rings: radial depth, 0.170 in.; free gap, approx. 1 1/16 in.; initial unit wall pressure, approx. 39 lb/sq in.

TABLE II—RESULTS OF 150-HOUR ENDURANCE RUN[a] IN SINGLE-CYLINDER ENGINE WITH NITRIDED-STEEL RINGS AND NITRIDED-STEEL CYLINDER

[Oil, Navy 1120; spark timing, 20° B. T. C.; power-section oil flow, 19.0 pounds per minute at 2500 rpm and 185° F oil-in temperature; limiting temperatures: rear spark-plug bushing, 450° F; rear center of cylinder, 350° F]

NACA reference run	Description	Conditions						Results					Remarks
		Run-in (hr)	Endurance (hr)	Speed (rpm)	bmep (lb/sq in.)	imep (lb/sq in.)	Oil-in temperature (°F)	Ring wear			Blow-by (cu ft/min) (b)	Specific oil consumption (lb/bhp-hr)	
								Weight loss	Percentage of original ring weight	Percentage of total loss			
14	Piston, from run 13, cleaned. Rings, nitrided; 2 new compression, 2 oil, from run 13. Cylinder, stock, honed finish.	1	150	1165 to 2520	75 to 206	125 to 234	205	0.147 .062 .042 .069	0.60 .25 .16 .26	46.0 19.4 13.0 21.6	0.35 to .65	0.018 to .061	General conditions of piston, rings, and cylinder very good; oil control poor.

[a] See appendix.
[b] Average value.

TABLE III—RESULTS OF HIGH-OUTPUT RUNS FOR NITRIDED-STEEL RINGS WITH NITRIDED-STEEL CYLINDER

[Oil, Navy 1120; spark timing, 20° B. T. C.; power-section oil flow, 19.0 pounds per minute at 2500 rpm and oil-in temperature of 185° F; limiting temperatures: rear spark-plug bushing, 450° F; rear center of cylinder, 350° F]

NACA reference run	Description	Conditions						Ring unit wall pressure (lb/sq in.) (a)	Results							Remarks
		Run-in (hr)	Endurance (hr)	Speed (rpm)	bmep (lb/sq in.)	imep (lb/sq in.)	Oil-in temperature (° F)		Ring wear			Free gaps		Blow-by (cu ft/min) (b)	Specific oil consumption (lb/bhp-hr)	
									Weight loss (gram)	Percent of original ring weight	Percentage of total loss	Before test (in.)	After test (in.)			
16	Piston, stock, Rings, stock cast-iron, 3 compression, 3 oil. Cylinder from reference run 15, repolished.	6	9½	2500	250	288	210	6 6 10 21 26 37	2.721 2.452 .873 .788 .787 .753	6.28 5.67 2.02 1.87 1.87 1.79	32.5 29.3 10.4 9.4 9.4 9.0	0.98 1.00 1.00 .91 .91 .84	0.79 .81 .88 .82 .82 .80	0.90	0.022 to .112 [b].071	Oil control poor; rings worn, scuffed and feathered slightly; cylinder condition good; exhaust valve guide boss broken.
17	Piston, from reference run 14 with ring grooves cleaned. Rings, from reference run 14. Cylinder, from reference run 14 without repolishing.	1	8	2500	250	286	210	------- ------- ------- -------	0.099 .026 .026 .017	0.41 .11 .10 .06	58.9 15.6 15.6 9.9	------- ------- ------- -------	1.03 1.03 1.00 1.00	0.36	0.045 to .024 [b].030	Complete exhaust-valve failure; particles found in ring grooves and piston skirt.
18	Piston, from reference run 17; cleaned completely, stoned lightly on thrust faces. Rings from reference run 17. Cylinder from reference run 17, without repolishing.	1	15½	2500	250	286	210	------- ------- ------- -------	0.191 .051 .045 .044	0.79 .21 .17 .17	57.6 15.5 13.5 13.4	1.03 1.03 1.00 1.00	1.03 1.00 .98 .98	1.07	0.040 to .022 [b].028	Exhaust-valve failure; cylinder discarded because of broken valve guide and boss; second ring, partly cold stuck.
19	Piston, from reference run 18, cleaned. Rings, from reference run 18. Cylinder, stock, honed finish.	1	50	2500	250	285	210	8 9 25 24	0.103 .014 .015 .010	0.43 .06 .06 .04	72.7 9.7 10.5 7.1	1.03 1.00 .98 .98	1.03 1.02 .98 .98	0.6 to 1.3	[c]0.022	Rings, piston, and cylinder in very good condition; piston boss cracked; incipient exhaust-valve failure.

[a] After run.
[b] Average value.
[c] Constant value.

TABLE IV—RESULTS OF HIGH-OUTPUT RUNS WITH HIGH-PRESSURE NITRIDED-STEEL OIL RINGS WITH NITRIDED-STEEL CYLINDER

[Oil, Navy 1120; spark timing, 20° B. T. C.; power-section oil flow: reference run 30, 17 pounds per minute; reference run 31, 19 pounds per minute at 185° F oil-in temperature; limiting temperatures: rear spark-plug bushing, 450° F; rear center of cylinder, 350° F]

NACA reference run	Description	Conditions						Ring unit wall pressure (lb/sq in.) (a)	Results							Remarks
									Ring wear			Free gaps				
		Run-in (hr)	Endurance (hr)	Speed (rpm)	bmep (lb/sq in.)	imep (lb/sq in.)	Oil-in temperature (° F)		Weight loss (gram)	Percentage of original ring weight	Percentage of total loss	Before test (in.)	After test (in.)	Blow-by (cu ft/min) (b)	Specific oil consumption (lb/bhp-hr)	
30	Piston, new. Rings, nitrided; 2 compression, 2 oil, high pressure, 1.6 in. free gap. Cylinder, stock, standard honed finish.	1	25	2500	210	242	185	.... 57 62	0.151 .059 .194 .177	0.62 .22 .73 .67	26.0 10.2 33.4 30.4	1.10 1.08 1.62 1.62	1.08 1.06 1.61 1.61	1.25 to .75	0.020 to .008 [a].014	Incipient compression-ring scuffing; steel particles from overhaul in piston skirt.
31	Piston, from reference run 30, cleaned. Rings, from reference run 30. Cylinder, from reference run 30, without repolishing.	1	49½	2500	250	278	210	....	0.137 .041 .026 .020	0.56 .17 .10 .08	61.2 18.3 11.6 8.9	1.08 1.06 1.61 1.61	1.07 1.06 1.61 1.61	1.22 to .75	[c]0.013	General condition good; incipient compression-ring scuffing; second ring partly cold stuck; third ring completely cold stuck; broken valve spring.

[a] Diametral tension measured by closing ring to correct gap (as measured in cylinder before run). These values not strictly comparable to unit wall pressures of other tables.
[b] Average value.
[c] Constant value.

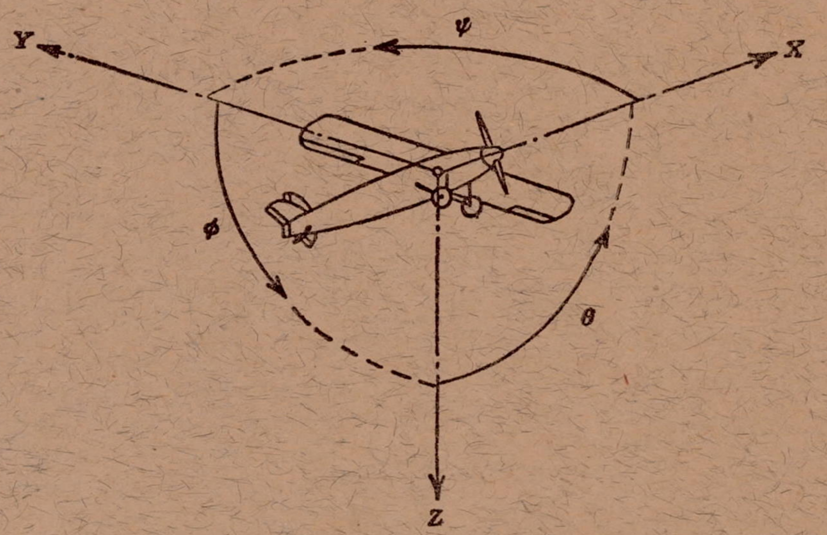
TABLE V.—SUMMARY OF RUNS FOR CHROME-PLATED CYLINDER WITH NITRIDED-STEEL PISTON RINGS

[Oil, Navy 1120; spark timing, 20° B. T. C.; power-section oil flow, 34 pounds per minute (maximum) at 2500 rpm and at 185° F oil-in temperature; limiting temperatures: rear spark-plug bushing, 450° F; rear center of cylinder, 350° F]

Run	Description	Condition						Ring unit wall pressure (lb/sq in.) (a)	Results							Remarks
									Ring wear			Free gaps				
		Run-in (hr)	Endurance (hr)	Speed (rpm)	bmep (lb/sq in.)	imep (lb/sq in.)	Oil-in temperature (° F)		Weight loss (gram)	Percentage of original ring weight	Percentage of total loss	Before test (in.)	After test (in.)	Blow-by (cu ft/min) (b)	Specific oil consumption (lb/bhp-hr)	
1	Piston, from reference run 31. Rings nitrided, from reference run 31. Cylinder, straight bore, chrome-plated.	6	25	2300	190	226	185	....	0.055 .019 .024 .025	0.24 .08 .09 .10	44.7 15.4 19.5 20.4	1.08 1.06 1.61 1.61	1.09 1.08 1.63 1.63	0.65	0.012 to .007 [b].008	Cylinder, piston, and rings in excellent condition.
2	Continuation of above run; piston and cylinder untouched.	1	25	2500	210	250	185	....	0.042 .007 .005 .007	0.17 .03 .02 .03	68.8 11.5 8.2 11.5	1.09 1.08 1.63 1.63	1.09 1.06 1.64 1.64	0.80	0.010 to .007 [b].008	Cylinder, piston, and rings in excellent condition; porosity on cylinder evident; piston cracked through pin boss; top ring has convex surface.
3	Continuation of above run. Piston from run 2 replaced with used, cleaned piston.	1	20½	2500	250	283	210	10 10 45 42	0.045 .009 .015 .014	0.19 .04 .06 .05	54.2 10.8 18.1 16.9	1.09 1.06 1.64 1.64	1.09 1.06 1.59 1.63	0.95	[c]0.011	Cylinder, piston, and rings in excellent condition; porosity evident on entire cylinder; compression rings have convex surface; head cracked between fifth and sixth fin, exhaust side, causing oil leak from rocker box and termination of run.

[a] After run.
[b] Average value.
[c] Constant value.

U. S. GOVERNMENT PRINTING OFFICE: 1948



Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Symbol		Designation	Symbol	Positive direction	Designation	Symbol	Linear (component along axis)	Angular
Longitudinal	X	X	Rolling	L	$Y \longrightarrow Z$	Roll	ϕ	u	p
Lateral	Y	Y	Pitching	M	$Z \longrightarrow X$	Pitch	θ	v	q
Normal	Z	Z	Yawing	N	$X \longrightarrow Y$	Yaw	ψ	w	r

Absolute coefficients of moment

$C_l=\frac{L}{qbS}$ (rolling) $\qquad C_m=\frac{M}{qcS}$ (pitching) $\qquad C_n=\frac{N}{qbS}$ (yawing)

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D Diameter
p Geometric pitch
p/D Pitch ratio
V' Inflow velocity
V_s Slipstream velocity
T Thrust, absolute coefficient $C_T=\frac{T}{\rho n^2 D^4}$
Q Torque, absolute coefficient $C_Q=\frac{Q}{\rho n^2 D^5}$

P Power, absolute coefficient $C_P=\frac{P}{\rho n^3 D^5}$
C_s Speed-power coefficient$=\sqrt[5]{\frac{\rho V^5}{P n^2}}$
η Efficiency
n Revolutions per second, rps
Φ Effective helix angle$=\tan^{-1}\left(\frac{V}{2\pi r n}\right)$

5. NUMERICAL RELATIONS

1 hp=76.04 kg-m/s=550 ft-lb/sec
1 metric horsepower=0.9863 hp
1 mph=0.4470 mps
1 mps=2.2369 mph

1 lb=0.4536 kg
1 kg=2.2046 lb
1 mi=1,609.35 m=5,280 ft
1 m=3.2808 ft