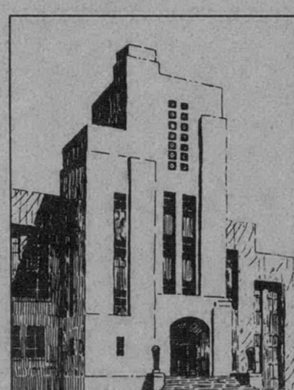
E DAVID W. TAYLOR MODEL BASIN

UNITED STATES NAVY

STATIC STRENGTH TESTS OF DIESEL ENGINE CRANKCASES GMC 16-184 AND EMC 16-184-A FOR 110-FOOT PATROL BOATS

BY J.W. DAY



AUGUST 1943

REPORT 501

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NAVY DEPARTMENT

DAVID TAYLOR MODEL BASIN

WASHINGTON, D.C.

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PERSONNEL

The tests were carried out by F.F. Vane and J.W. Day, with the assistance of C.A. Wagley, J.G. McGinley, E.P. Donoghue, W.S. Preston, Jr., and D.M. Barden of the Taylor Model Basin staff. The report is the work of J.W. Day.

STATIC STRENGTH TESTS OF DIESEL ENGINE CRANKCASES GMC 16-184 and EMC 16-184-A FOR 110-FOOT PATROL BOATS

ABSTRACT

Static strength tests were made on two diesel engine crankcases, General Motors Type 16-184, designed for a unit of 16 cylinders in 4 vertical banks of 4 cylinders each. The crankcases were of welded steel construction, and were built up of forged cylinder deck plates and steel-plate walls and transverse bulkheads.

The tests were carried out for three different conditions of static loading, i.e., in tension, to simulate loads imposed on the cylinder head by cylinder gas pressure; in compression, to simulate loading the main-bearing carriers as beams; and in torsion, to simulate torque loads imposed by the propeller shaft and by piston side thrust. Stresscoat strain-indicating lacquer was used to indicate points of stress concentration on the crankcases.

The static tests indicated that the crankcases had adequate strength for all service requirements. The dynamic and the endurance strength of the crankcases were not investigated in these tests.

INTRODUCTION

This report describes static strength tests of two full-scale crankcases for the Type 16-184, 1200-HP diesel engine used for the propulsion of 110-foot patrol boats of the United States Navy. The tests were carried out at the David Taylor Model Basin at the request (1)* of the Bureau of Ships.

These crankcases embodied the results of experimental development conducted by the General Motors Corporation of Detroit, Michigan, which produced the GMC 16-184 case, and by the Electro-Motive Corporation of La Grange, Illinois, which built the EMC 16-184-A case. The EMC crankcase is a production model of a later type than the GMC crankcase and embodies several changes. It was fabricated by a new technique developed by the Electro-Motive Corporation.

These crankcases are truly remarkable pieces of engineering, and they will well repay careful study by anyone whose work involves mechanical design, welding design, welding technique and weight saving. In addition to the diagrams in the body of the report, a series of large scale views of various parts of this unusual piece of machinery are included as Appendix 1.

This engine is unusual in that the crankshaft axis is vertical, and the cylinders are arranged around it in four X-shaped tiers of four cylinders each, as shown in Figure 1. For ship installations the engine rests on the top of a case containing bevel gears which change the motion 90 degrees to the propeller shaft.

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* Numbers in parentheses indicate references on page 18 of this report.

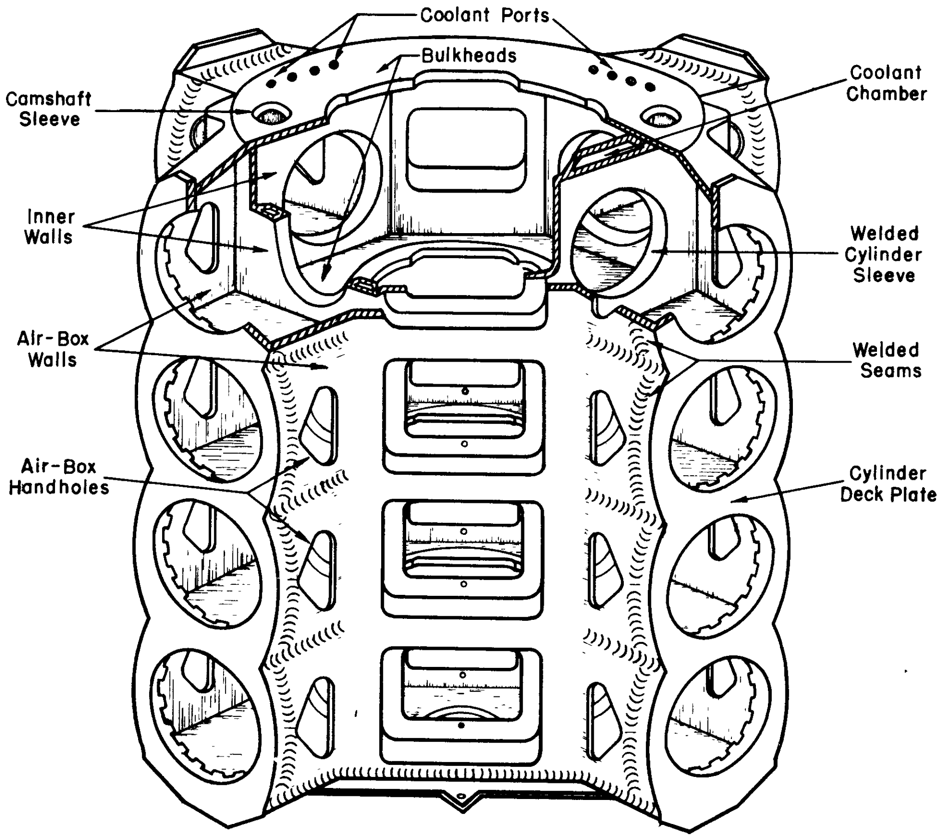

Figure 1 - Isometric Sketch of Crankcase with Two Cylinders sectioned to show Construction of Inner Walls

The static tests at the Taylor Model Basin were carried out to determine points of high stress concentration in the crankcases, and to investigate the ability of the crankcase structures to withstand the working loads of the engine.

Preliminary to these tests a stress analysis was made by the Bureau of Ships (2) for the EMC crankcase to determine the magnitude and direction of the forces transmitted to the crankcase through the head and the main bearings; a synopsis of this report is given in Appendix 2. As a result of this preliminary analysis, the Bureau of Ships recommended:

1. A static tensile test of each section of the crankcase at a load of 34,000 pounds (2) (4).
2. A compressive test (2), with a 10,000-pound load applied at an angle of 45 degrees to the serrated edge of the bearing carriers.
3. A torsional loading test (3) (4), with no load limits specified.

An analysis of bearing loads was made by the Research Laboratories of the General Motors Corporation (5) which substantiated the recommendations of the Bureau of Ships in Reference (2). A synopsis of the General Motors report is given in Appendix 3.

The pioneer investigation of the Type 16-184 crankcase was a fatigue test on a partial crankcase by the General Motors Research Laboratories in 1938 (6). This test indicated that the crankcase was amply strong for all service load requirements. A synopsis of this report is contained in Appendix 4.

The present static tests of crankcases GMC 16-184 and EMC 16-184-A conducted at the Taylor Model Basin supplement several other tests conducted by the Electro-Motive Corporation.

TEST APPARATUS AND PROCEDURE

Loading fixtures were designed and constructed for making the tension, compression and torsion tests listed in 1, 2, and 3 of the preceding section. The fixtures were designed to take as little space as possible inside the crankcase to permit applying Stresscoat lacquer and observing the formation of strain crack patterns. Figure 1 shows the construction of the crankcase and gives the nomenclature of the parts.

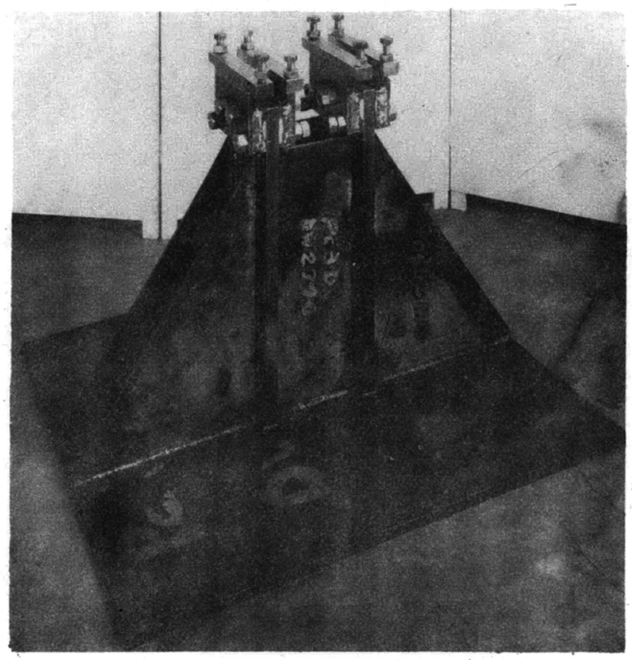

Figure 2 - Pedestal for Compression Test of Crankcase

As shown in Figure 3, the crankcase rests with its inner wall bearing on the eight studs on top of the pedestal. The studs are adjusted in height so that all bear on the crankcase, and are then locked in position. The pedestal was designed to bring the reactions on the crankcase as close as possible to the main bearing carrier when under test.

COMPRESSION TESTS

The details of the compression test pedestal are shown in Figure 2; the method of loading the crankcase in compression is shown in Figure 3. The GMC 16-184 crankcase, mounted for compression testing in the Southwark-Emery 600,000-pound testing machine, is shown in Figure 4.

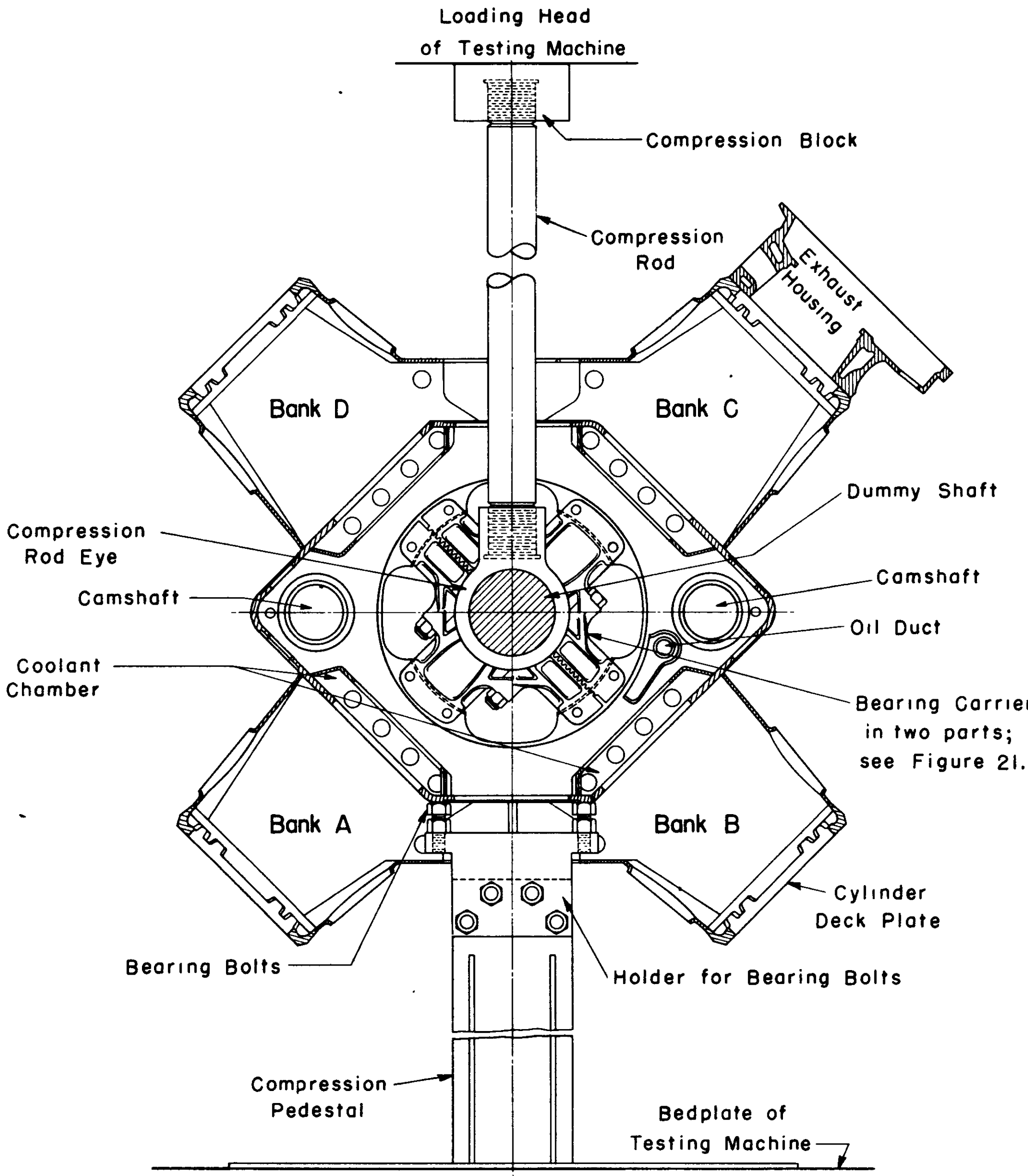

Figure 3 - Section through Crankcase EMC 16-184-A
set up for Compression Test

This section shows in detail the loading fixtures and their application to the crankcase in loading the assembly in compression. The force is transmitted downward from the testing machine through the compression block, rod and eye to the dummy shaft, then through the bearing carriers and the crankcase to the compression pedestal and the bedplate of the testing machine.

The loading in this test simulates the loads transmitted by the shaft to the crankcase, which cause the bearing carriers to be loaded in both bending and compression. The load was applied at an angle of 45 degrees to the serrated joint between the two halves of the carriers because the moment of inertia of the bearing support was least in that section.
The cylinder sleeves which are welded between the walls of the coolant chamber are not shown in the section.

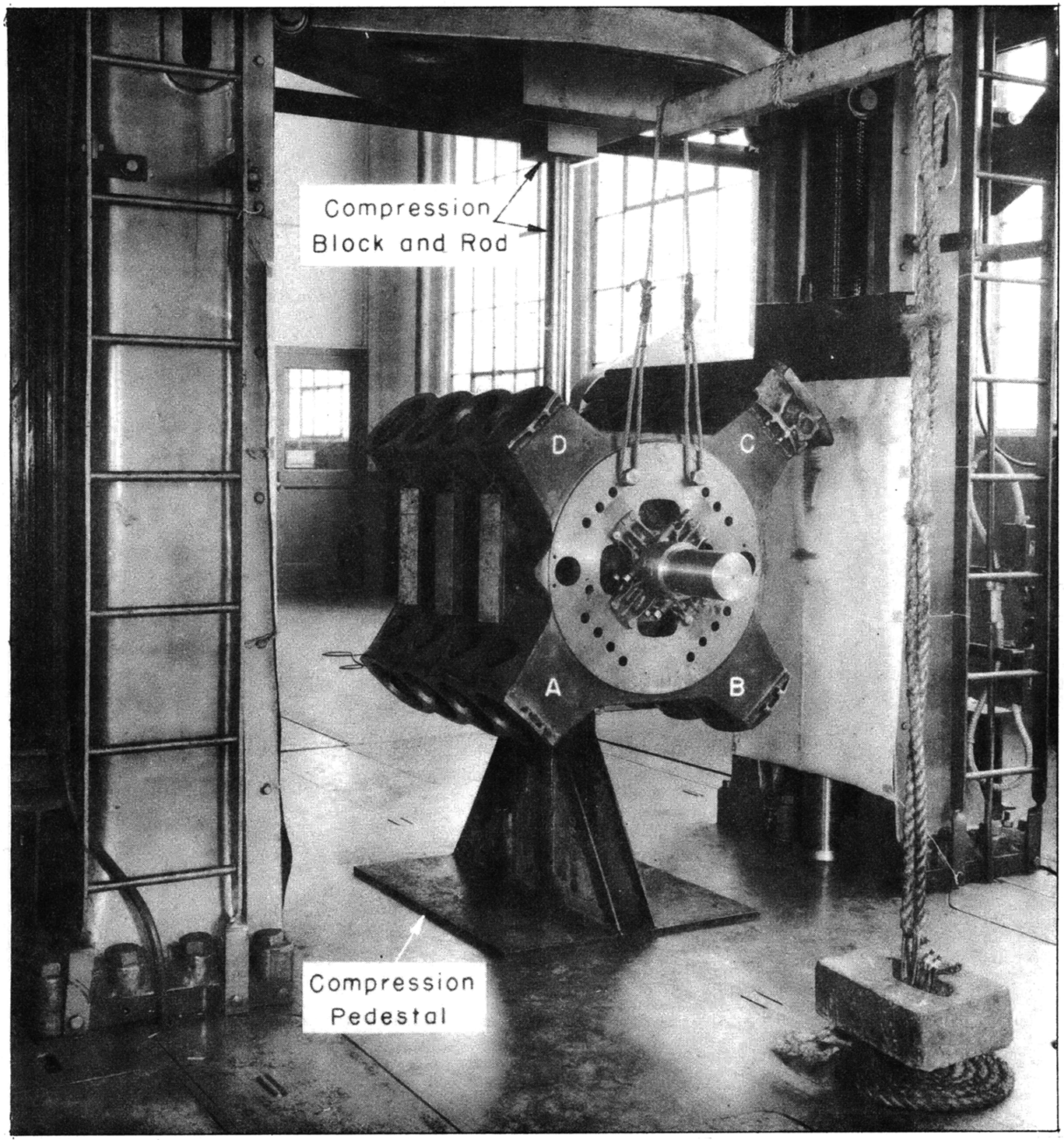

Figure 4 - Crankcase GMC 16-184 mounted in Testing Machine for Compression Test

This is a view of the compression test arrangement shown diagrammatically in Figure 3. The test is being made through one of the inner cylinders.

The counterbalance weights suspended from Bank D neutralize the torsional effect of the exhaust housing mounted on Bank C.

The beam-and-poise tackle attached to the front of the crankcase balances the cantilever effect of the two front tiers of cylinders on the rear portion of the crankcase.

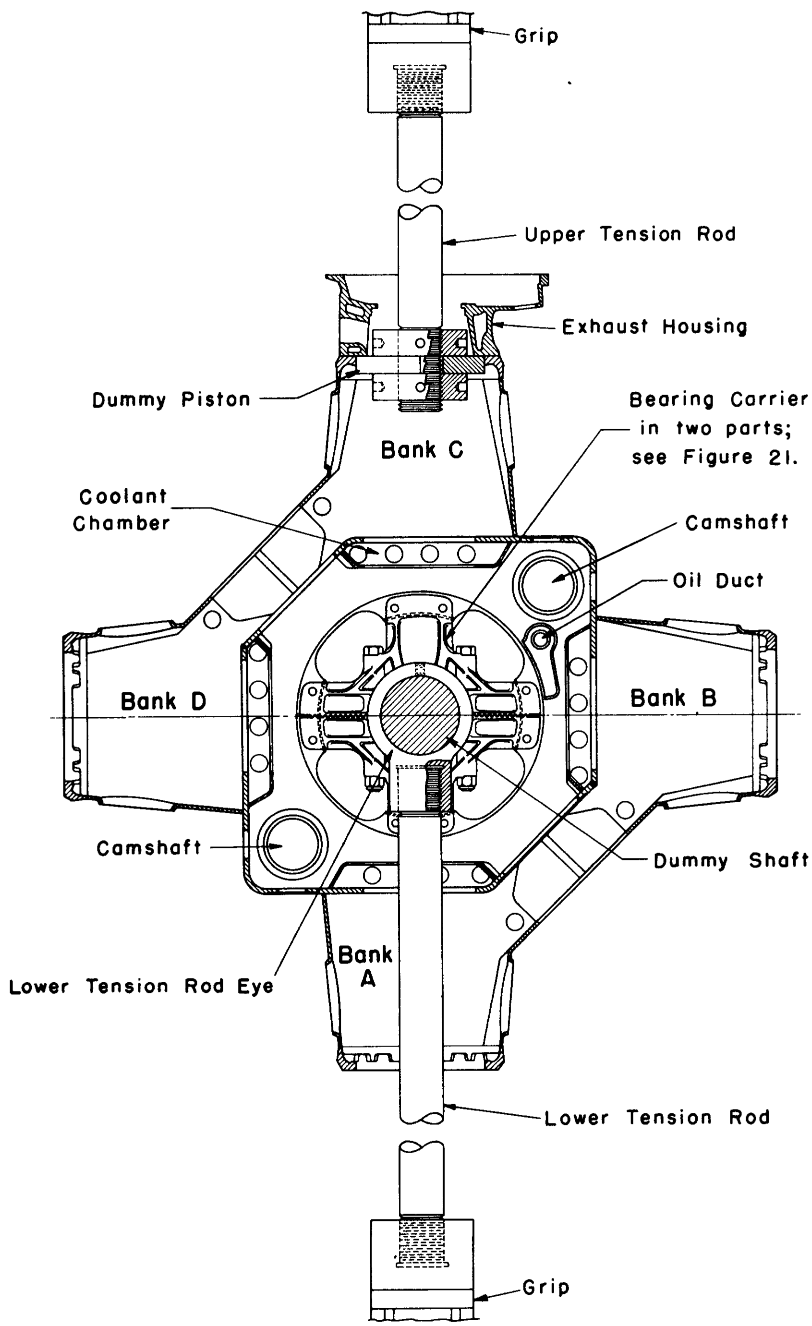

Figure 5 - Section through Crankcase EMC 16-184-A
set up for Tension Test

The section shows in detail the method of loading the crankcase in tension. The pulling force exerted on the upper grip and the upper tension rod is transmitted to the exhaust housing by the dummy piston, which consists of a split ring which bears against the lower side of the housing, and which is held in place on the tension rod by two capstan nuts. The split-ring construction facilitates easy insertion and removal of the loading assembly in the restricted space of the cylinder airbox. The load is received by the crankcase through the studs which secure the exhaust housing to the airbox deck plate. The pulling force on the lower grip and the lower tension rod is transmitted to the shaft by an eye which encircles the shaft and is screwed to the end of the rod. The shaft transmits the load to the main bearing carriers which in turn transmit the load to the crankcase.

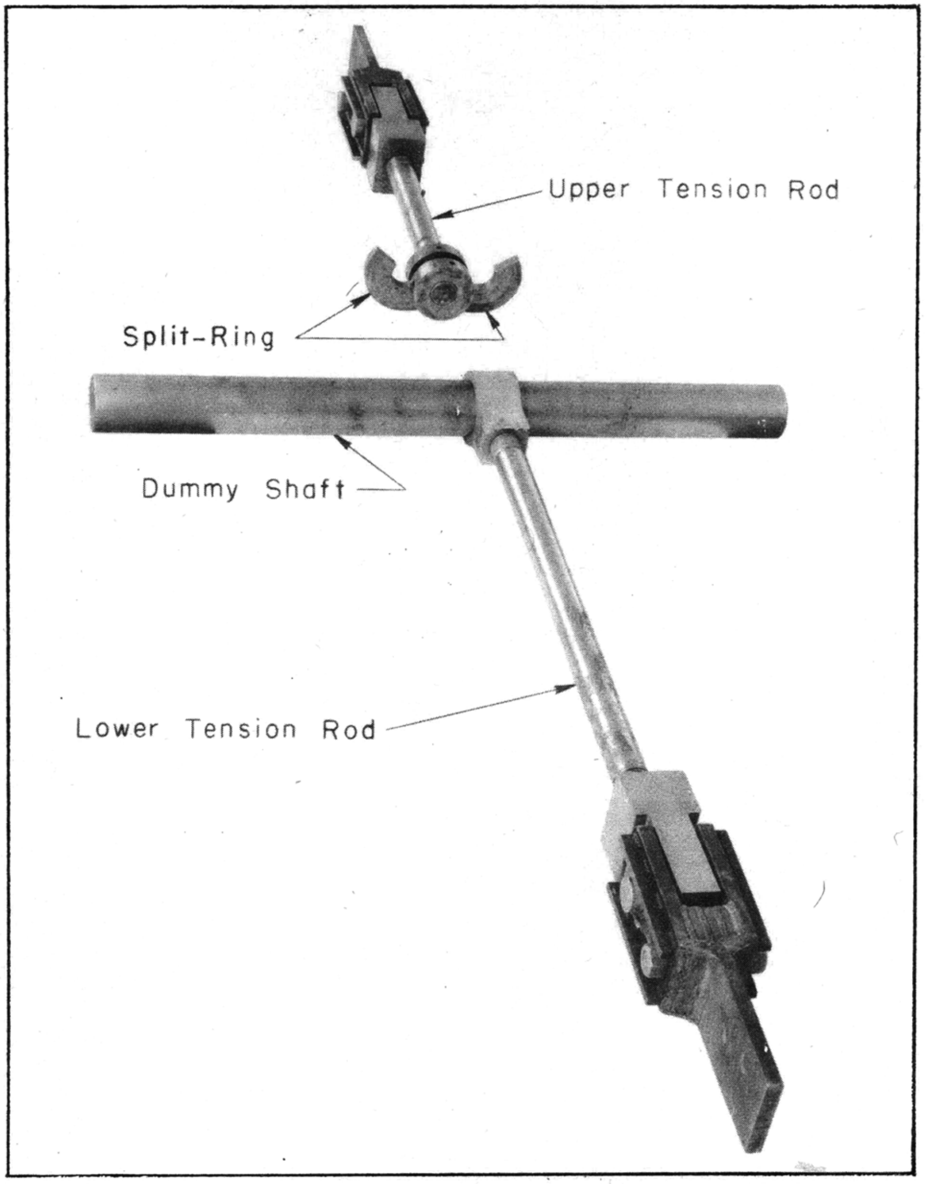

Figure 6 - Loading Fixtures for Tension Tests of Crankcase

The fixtures are shown in their correct relative positions for loading the crankcases in tension. The dummy piston, shown with the split ring open between the circular capstan nuts, fits in the upper cylinder opening. The split ring bears against the under side of the exhaust housing and the pin-ended yoke is gripped in the upper head of the testing machine.

The large dummy shaft is fitted in the bearing carriers of the crankcase. The rod with the eye fits on the shaft and passes through the lower cylinder opening. The pin-ended yoke on the bottom of the rod is gripped in the lower head of the testing machine.

TENSION TESTS

The method of loading the crankcase in tension is shown diagrammatically in Figure 5; the fixtures are shown photographically in Figure 6. Figures 7 and 8 illustrate the tension testing arrangement with the GMC 16-184 crankcase in the machine.

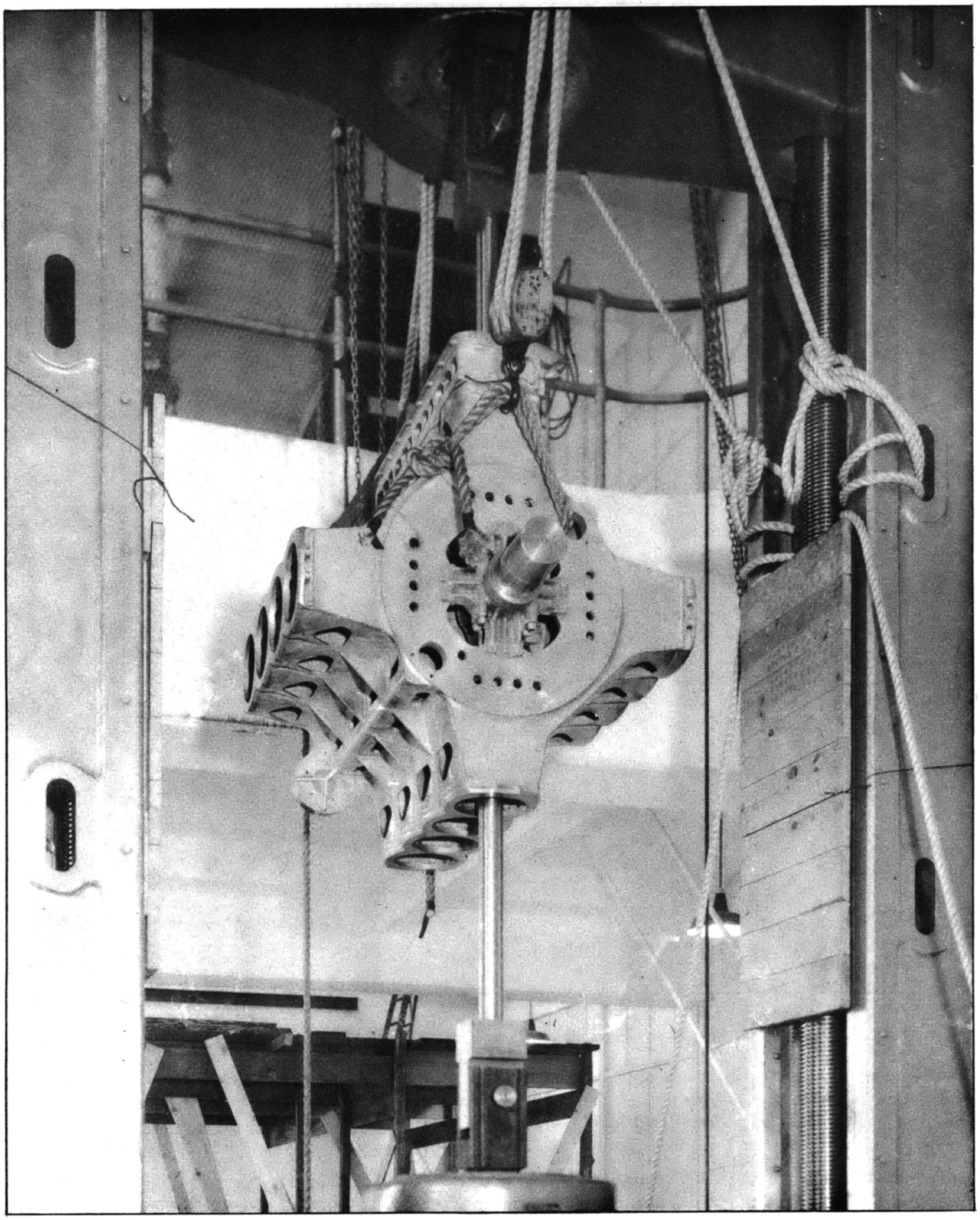

Figure 7 - Crankcase GMC 16-184 mounted in 600,000-Pound Testing Machine for Tension Test

The crankcase is here set up for a tension test, as shown in the diagram of Figure 5.

The weight of the rear part of the crankcase is balanced by a tackle and counterweight; this neutralizes the cantilever action on the front part of the crankcase.

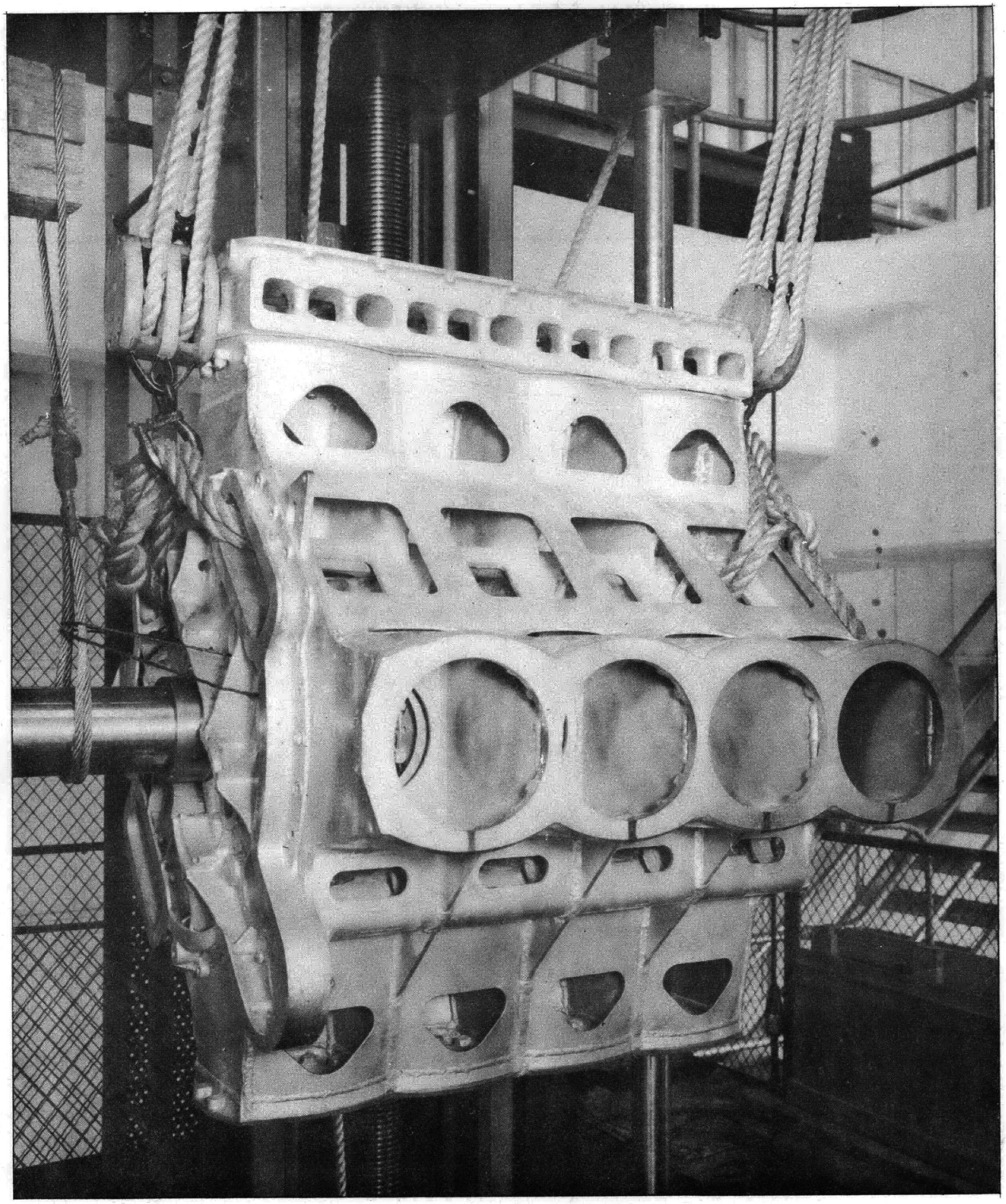

Figure 8 - Close-up View of Crankcase GMC 16-184
mounted for Tension Test

This view shows in considerable detail the structure of the crankcase and the method of support during the test. Stresscoat lacquer has been applied to the inside and outside of the crankcase.

TORSION TEST

For this test the crankcase was bolted through one of the end bulkheads to a heavy steel plate, which was in turn bolted to the base plate of the testing machine, as shown in Figure 9. The note under this figure explains in detail the method of applying the test torque.

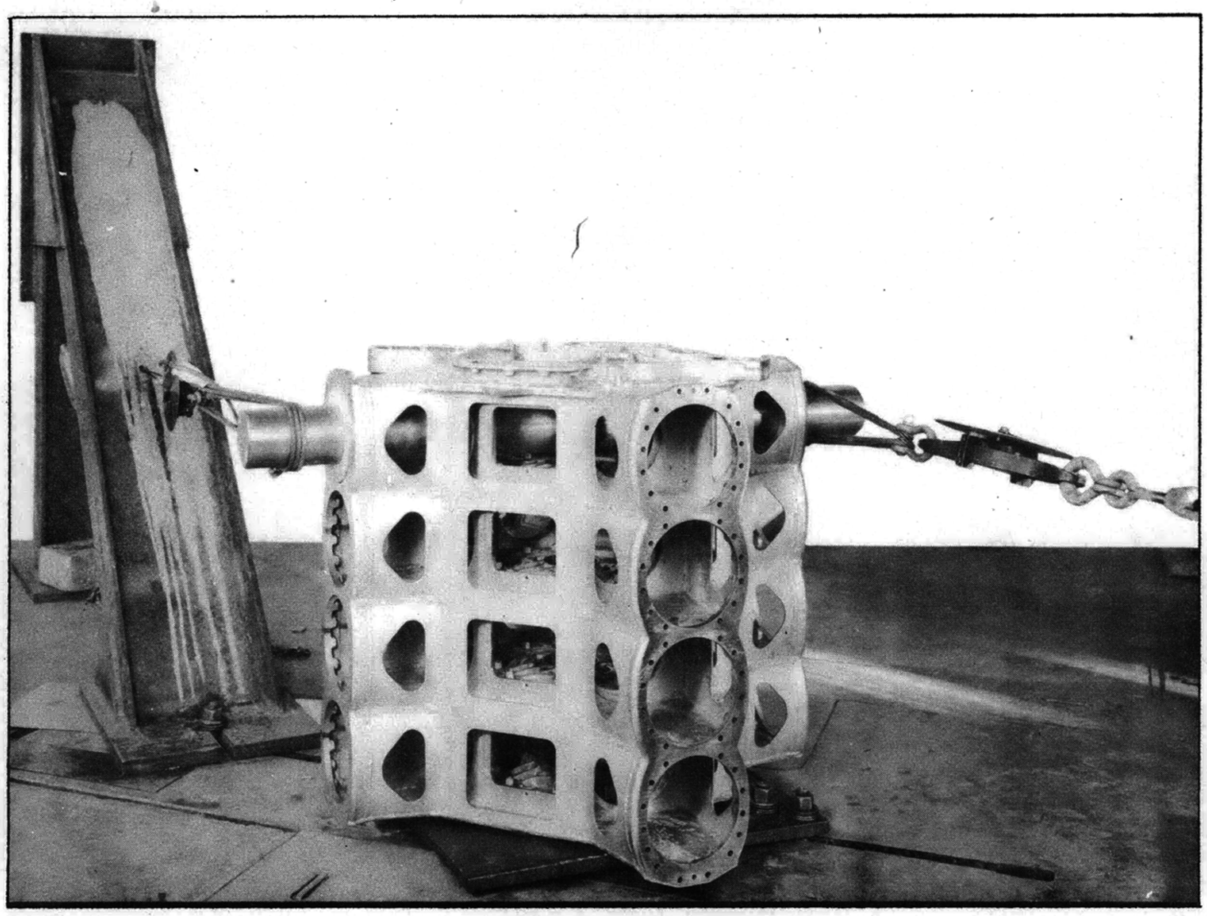

Figure 9 - Crankcase GMC 16-184 set up for Torsion Test

The crankcase was bolted through Bulkhead 5 to a heavy plate which was in turn bolted to the base plate of the testing machine.

A heavy bar was placed through the upper opposite cylinders and was held in place by hardwood bushings fitted in the deck plate holes. Pulling tackles were attached to the shaft ends. A spring dynamometer in each tackle indicated the force applied. The force on the right side was applied by the turnbuckle, the end of which appears in the view. The force on the left side was applied with a differential chain fall, the edge of which is shown between the members of the A-frame. Equal forces were applied simultaneously by each tackle, so that a force couple was applied to the crankcase. This test simulated the loading of the crankcase which would be caused by piston side-thrust and torsional action of the shaft load.

STRESSCOAT INVESTIGATION

Before applying the undercoat lacquer the inside and outside surfaces of the crankcases were thoroughly cleaned by removing all loose scale, grease and oil. The undercoat of aluminum-pigmented lacquer was applied with a compressed-air sprayer to the inside and the outside surfaces of the crankcase and exhaust housing. The Stresscoat, a brittle strain-indicating lacquer, was applied after the undercoat had dried.

The minimum stress in steel which can be indicated by the Stresscoat method is about 18,000 pounds per square inch.

The gear described previously in this section was used for applying tension and compression to the crankcases, and for developing cracks in the Stresscoat at points of large strain.

TEST SCHEDULE AND RESULTS

After loading to a maximum compressive load of 30,000 pounds, the crankcase GMC 16-184 was dismantled from the compression mount and set up in the testing machine for testing in tension, as shown in the photographs, Figures 7 and 8. The sequence of loading during the test is indicated in Table 1. The maximum load applied in tension was 60,000 pounds, 75 per cent over the maximum load recommended by the Bureau of Ships.

Crankcase GMC 16-184 was loaded to a torque of 160,000 pound-inches, the capacity of the testing equipment. Crankcase GMC 16-184 was then dismantled and Crankcase EMC 16-184-A was tested in a similar manner in tension and compression.

Representatives from the Bureau of Ships requested that the tests on Crankcase EMC 16-184-A be carried to destruction. The compression test was carried to a load of 300,000 pounds, at which point the pedestal fixture shown in Figure 2 failed. The tension test was carried to a maximum load of 170,000 pounds with Exhaust Housing 1, and to a maximum load of 157,000 pounds with Exhaust Housing 2; the housings failed at the respective loads. The tests were then discontinued.

DISCUSSION OF RESULTS

Of the three tests, compression, tension, and torsion, conducted on each crankcase, the tension test was the only one that produced Stresscoat crack patterns.

TENSION TEST

The strain patterns produced on Main Bearing Carrier 5 in the tension test of Crankcase EMC 16-184-A are shown in Figure 10; this was the outside carrier on the end nearest the tension rod. The strain patterns indicated tension and occurred on the side of the carrier opposite to that shown in Figure 10. The occurrence of strain cracks on one side of the carrier and not the other indicated the presence of bending in the carrier. The bending tensile stress added to the membrane tensile stress resulted in a combined stress sufficient to form strain patterns on the fillets of the

TABLE 1

Test Loading Log

Date	Crankcase	Type of Loading	Load pounds	Time Load Was Held	Loaded at Cylinder	Stresscoat Crack Patterns at Release of Load
1941						
17 Dec	GMC 16-184	Compression	20 000	30 seconds	2	none
17 Dec			20 000	3 1/2 hours		
20 Dec			20 000	30 seconds		
20 Dec			25 000	30 seconds		
20 Dec			30 000	30 seconds		
20 Dec			20 000	3 3/4hours		
27 Dec	GMC 16-184	Tension	34 000	30 seconds	4	none
27 Dec			34 000	3 1/2 hours	4	
29 Dec			34 000	30 seconds	1	
30 Dec			34 000	3 1/2 hours	1	
30 Dec			50 000	30 seconds	1	
30 Dec			60 000	30 seconds	1	
1942			pound-inches			
7 Jan	GMC 16-184	Torsion	60 000	30 seconds	1	none
7 Jan			75 000	5 hours		
7 Jan			100 000	over night 16 hours		
8 Jan			0	4 hours		
8 Jan			100 000	30 seconds		
8 Jan			125 000	30 seconds		
8 Jan			150 000	30 seconds		
8 Jan			160 000	30 seconds		
10 Feb	EMC 16-184-A	Tension	34 000	30 seconds	1	none
10 Feb			50 000	30 seconds		
11 Feb			50 000	3 1/2 hours		
11 Feb			75 000	30 seconds		
11 Feb			75 000	3 hours		
12 Feb	EMC 16-184-A	Tension	34 000	30 seconds	4	none
12 Feb			50 000	30 seconds		
13 Feb			50 000	3 3/4 hours		
13 Feb			75 000	30 seconds		
13 Feb			75 000	3 1/2 hours		
17 Feb	EMC 16-184-A	Compression	20 000	30 seconds	3	none
17 Feb			30 000	30 seconds		
18 Feb			40 000	30 seconds		
18 Feb			50 000	30 seconds		
18 Feb			60 000	30 seconds		
18 Feb			70 000	30 seconds		
18 Feb			70 000	3 1/2 hours		
27 Feb			80 000	30 seconds		
27 Feb			90 000	30 seconds		
27 Feb			100 000	30 seconds		
27 Feb			110 000	30 seconds		
27 Feb			120 000	30 seconds		
27 Feb			130 000	30 seconds		
27 Feb			140 000	30 seconds		
27 Feb			150 000	30 seconds		
2 Mar			150 000	30 seconds		
2 Mar			10 000-pound increments to 300 000	each 30 seconds		none: compression pedestal failed
4 Mar	EMC 16-184-A	Tension	80 000	30 seconds	1	none
4 Mar			10 000-pound increments to 170 000	30 seconds each		initial crack patterns on bearing carrier at 90,000 pounds load
4 Mar			130 000	30 seconds		half patterns
4 Mar			170 000	30 seconds		full local patterns - exhaust housing cracked
23 Mar			157 000	30 seconds		second exhaust housing cracked

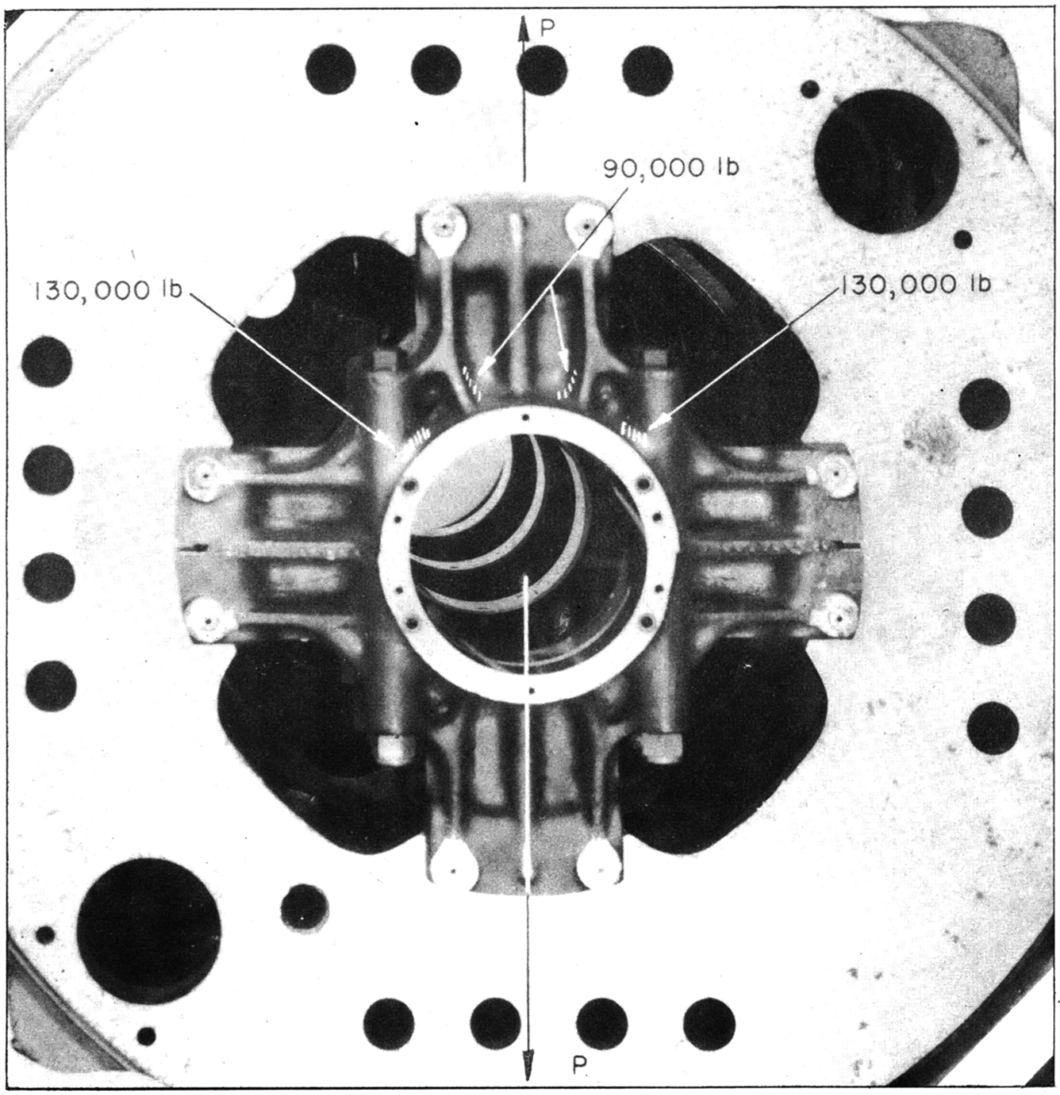

Figure 10 - Stresscoat Strain Patterns on Main Bearing Carrier in Bulkhead 5, Crankcase

The Stresscoat strain patterns were similar to those shown but were actually on the inner side of the bearing carrier, opposite the positions indicated. A tension load was applied in a vertical direction as indicated by the force arrows. The loads at which the strain patterns first appeared are marked on the photograph. The patterns which appeared at lower loads became accentuated as the load was increased.

inside surface of the bearing carrier. On the outer surface of the carrier the bending stress was compressive and neutralized the tensile stress, so that no strain patterns were produced.

The nature of the loading is shown more clearly in the sketch, Figure 11. There the sections of Main Bearing Carriers 4 and 5 with the adjoining crankcase bulkheads and the tension-rod eye are shown. The points of application of the test load also are shown. The section of Bearing Carrier 5 is unsymmetrical in relation to the

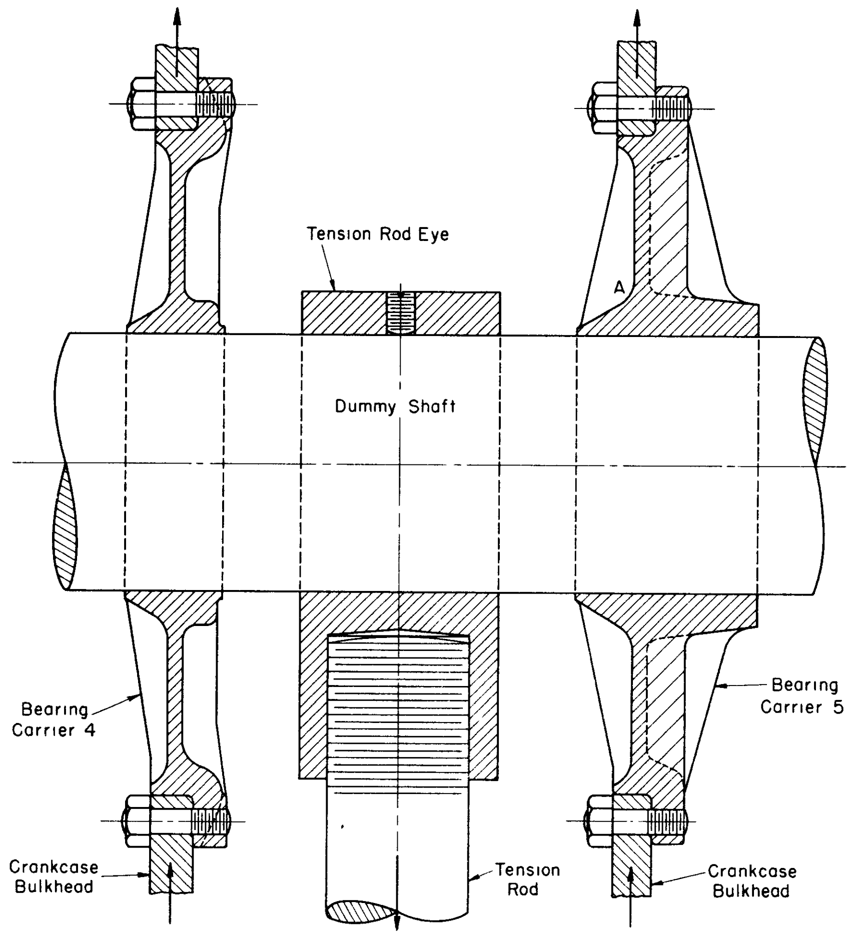

Figure 11 - Section through Main Bearing Carriers 4 and 5

The schematic drawing shows the sections of the bearing carriers, the adjoining crankcase bulkheads, and the tension-rod eye in relation to each other during the testing. The points of load application also are shown. Tension forces acting through the upper sides of the bulkheads and also in the tension-rod caused bending of the dummy shaft, which in turn caused bending of the carriers. This bending action was revealed by Stresscoat lacquer crack patterns only at the position indicated at A. Here the bending stress was tensile, which added to the tensile membrane stress in the carrier, whereas in the corresponding area on the lower part of the carrier the bending stress was compressive and added to the compressive membrane stress. However, no Stresscoat patterns were formed here as the duration of the load was too short for the Stresscoat to indicate compression.

bulkhead; where that of Bearing Carrier 4 is comparatively symmetrical with respect to the section of the bulkhead. This lack of symmetry could well have accounted for some of the bending action in Carrier 5.

There was additional cause for stress concentration on the inside members of Main Bearing Carrier 5, in that the loaded cylinder was at the end of the crankcase and without adjacent structure, such as is found in service conditions,* to absorb

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* In service conditions, the right end of the crankshaft, shown in Figure 11 as a dummy shaft, is connected to a propeller shaft by bevel gears and operates with its axis normal to the axis of the propeller shaft. Therefore, in service, the reaction of the propeller shaft through the bevel gears and crankshaft on Bearing Carrier 5, may well offer a more balanced condition of loading than did the test loading. Hence, it would show less tendency to bend the carrier than did the test loading.

some of the load. In the case of Bearing Carrier 4, adjacent to the loaded cylinder, which had considerably less cross-sectional area than Bearing Carrier 5, the stress concentrations were not sufficient to produce strain crack patterns. It is believed that the distribution of load on Bearing Carrier 4 more nearly simulated service conditions than did those on Bearing Carrier 5.

The strain patterns on the inner surfaces of the airbox walls, Figures 12a and 12b, were to be expected as the area of the horizontal cross section at the lower corners of the handholes was less than that of any other section between the points of load application, with the possible exception of the bearing carriers. The full strain patterns at the corners of the airbox walls at their junction with the bulkhead and the deck plate at the end of the crankcase probably were due to the combination of welding stresses near the corner and the stresses set up by loading, and to the absence of adjacent cylinder structure which would have absorbed some of the load.

It is apparent from Figures 13 and 14 that the failure of the exhaust housings in the webs above the stud bossings is to be expected as the stud connections overhang and the housing is subjected to bending in this region. In addition, because of greater spacing, the two end studs in the overhanging flange assume more load than the others.

COMPRESSION TEST

In the compression test, as shown in Figures 3 and 4, pages 4 and 5, the load was applied through the compression rod to the dummy shaft and to the main bearing carriers of one of the inner cylinders. The load was then transmitted through the two adjacent bulkheads, which have a heavy section, to the inner crankcase wall, which has a heavier section than the outer and airbox walls. The compression pedestal took the load from the inner wall at points close to the junction of the inner wall and the heavy bulkhead.

The compression load requested by the Bureau of Ships was 10,000 pounds. The compression pedestal was designed for a working load of *double* this amount, with a reasonable factor of safety.

The crankcase was not stressed sufficiently in any part during the compression test, up to the designated load and to double that load, to form any strain patterns. The load was increased progressively up to a maximum applied load of 300,000 pounds, thirty times the designated load, at which failure of the pedestal fixture occurred. No strain patterns in the brittle lacquer appeared even at this extreme load.

TORSION TESTS

The torsion test, illustrated in Figure 9, page 10, was carried to a maximum torque of 160,000 pound-inches. It was estimated that the approximate stress in

(Text continued on page 18)

Figure 12a - Valve-Pushrod Side

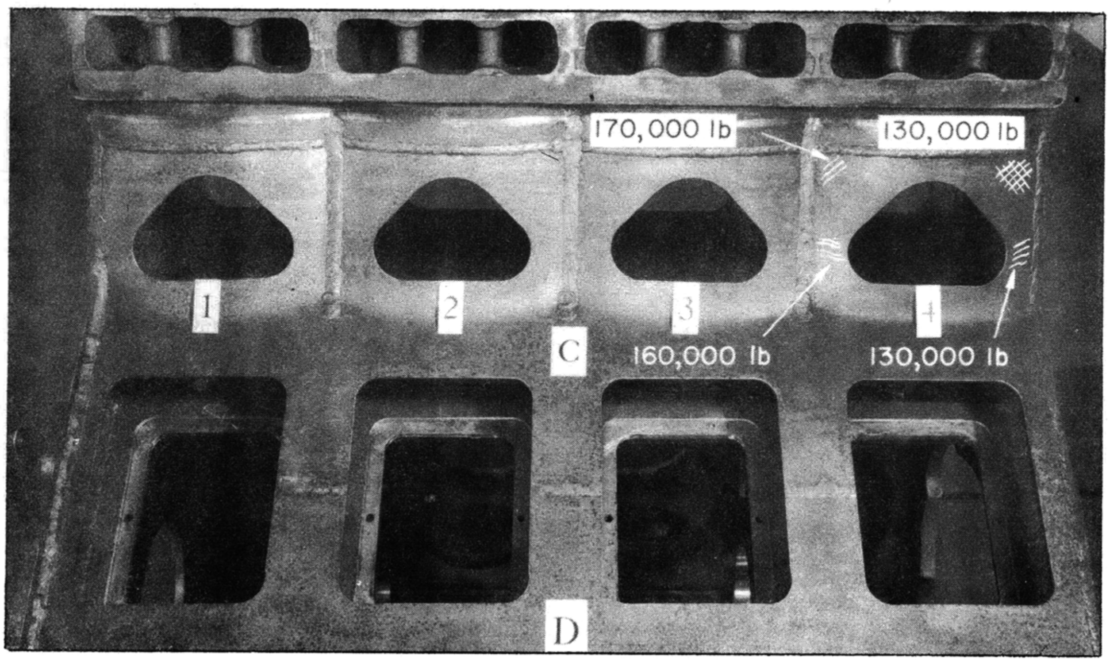


Figure 12b - Valve-Port Side

Figure 12 - Stresscoat Strain Patterns on Crankcase EMC 16-184-A after Tension Test

The Stresscoat strain patterns are represented by the hachure lines on the outside of the airbox wall, but the patterns actually appeared at the corresponding positions on the inside of the wall.

The tension loads at which the patterns first became visible are indicated. The patterns developed further as the load was increased. The tension load was applied along the centerline of Cylinder 4.

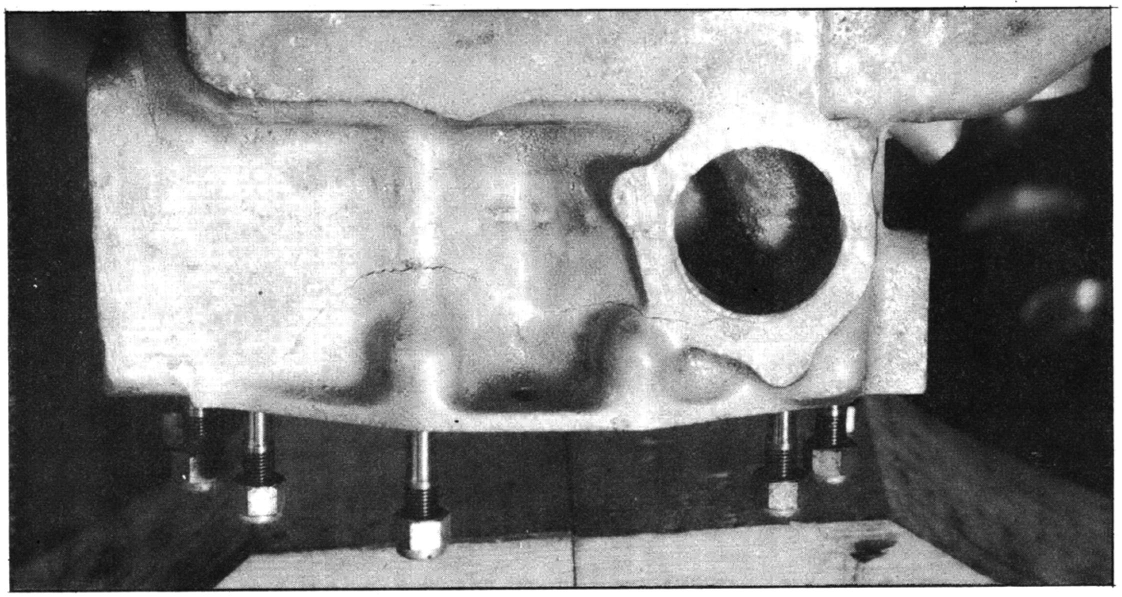


Figure 13 - Exhaust Housing 1 showing Failure Cracks after Tension Test

Two cracks appeared at a tensile load of 170,000 pounds.

One crack appeared across the web above the left stud nub; the other crack appeared above the right stud nub and crossed the rim of the cooling-water port.

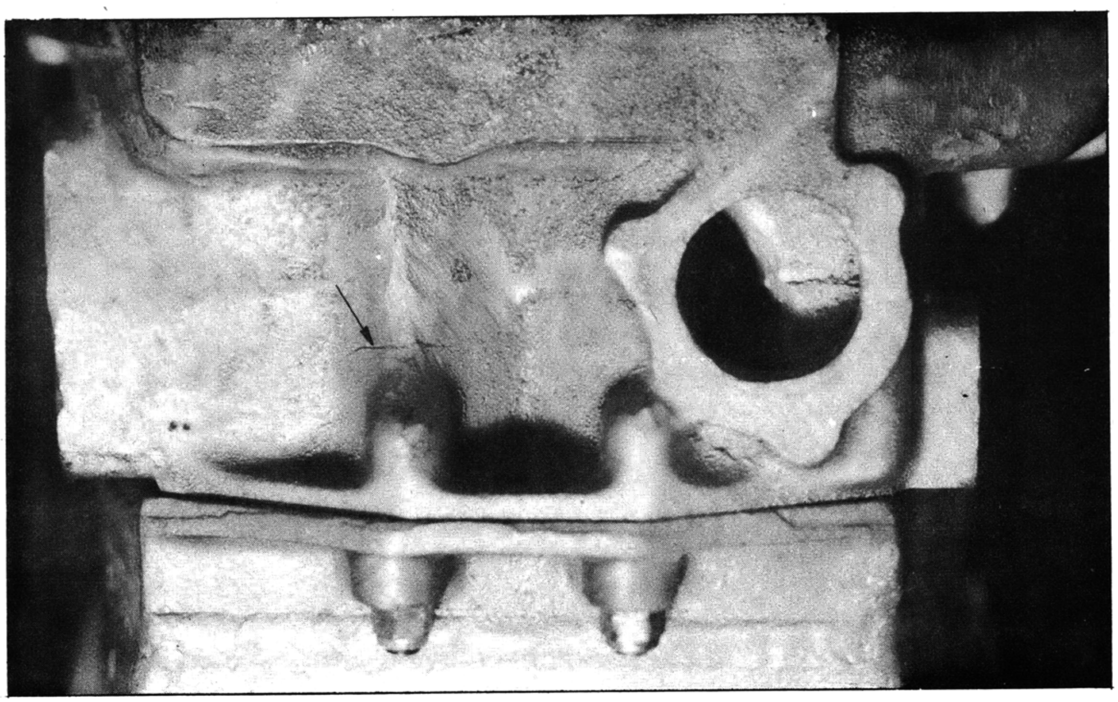


Figure 14 - Exhaust Housing 2 showing Failure Cracks after Tension Test

The crack indicated by the arrow appeared across the web just above the left stud nub at a tensile load of 157,000 pounds.

the outer members of the crankcase was 650 pounds per square inch at the highest loading. This stress value was insufficient to produce strain patterns in the Stresscoat lacquer.

Although the test equipment proved inadequate for the production of high stresses during the torsion and the compression tests, these tests were taken to a sufficiently high load to show that the crankcase possessed static strength considerably in excess of load requirements.

CONCLUSIONS

1. The static strengths of the Electro-Motive Corporation diesel Crankcase EMC 16-184-A and of the General Motors Crankcase GMC 16-184 are far in excess of the working loads.

2. The present tests throw no light on the endurance strength of the crankcases under service loading.

REFERENCES

(1) Bureau of Ships letter NOs-78926 (Dq) of 12 May 1941 to Inspector of Machinery, USN, c/o Electro-Motive Corporation, La Grange, Illinois, copy to TMB.

(2) "Preliminary Report on Stress Analysis of Crankcase for 16-184 Diesel Engine Built by the Electro-Motive Corporation," 10 June 1941. Approved by Lt. D.C. MacMillan, USN, Inspector of Machinery; see Appendix 2.

(3) Taylor Model Basin letter S41-5-(1) of 6 June 1941 to Bureau of Ships.

(4) Bureau of Ships letter NOs-78926 (3643) of 5 August 1941 to TMB.

(5) "Bearing Load Analysis of 16-184-A, 90° X Engine," Report ME-6-68, Project 26, 18 July 1941, Research Laboratories Division, General Motors Corporation, Detroit, Michigan; see Appendix 3.

(6) "Life Test on a Section of the Welded Steel Crankcase for the 16-184 Radial Diesel Engine," Report ME-3-9, Project 26, 18 November 1938, Research Laboratories Division, General Motors Corporation, Detroit, Michigan; see Appendix 4.

APPENDIX 1

STRUCTURAL DETAILS OF CRANKCASES

In the following views the main structural features of both crankcases are shown, and the principal differences in construction between Crankcase GMC 16-184 and Crankcase EMC 16-184-A are indicated. The GMC was an experimental crankcase whereas the EMC crankcase represented the production model at the time this static test was being made.

Figures 15 to 17 inclusive and Figures 19 and 20 show a crankcase at various stages of partial assembly. Figures 21 to 23 inclusive show some of the structural differences between the GMC and EMC crankcases as well as a few of the details of the inner parts of the crankcases.

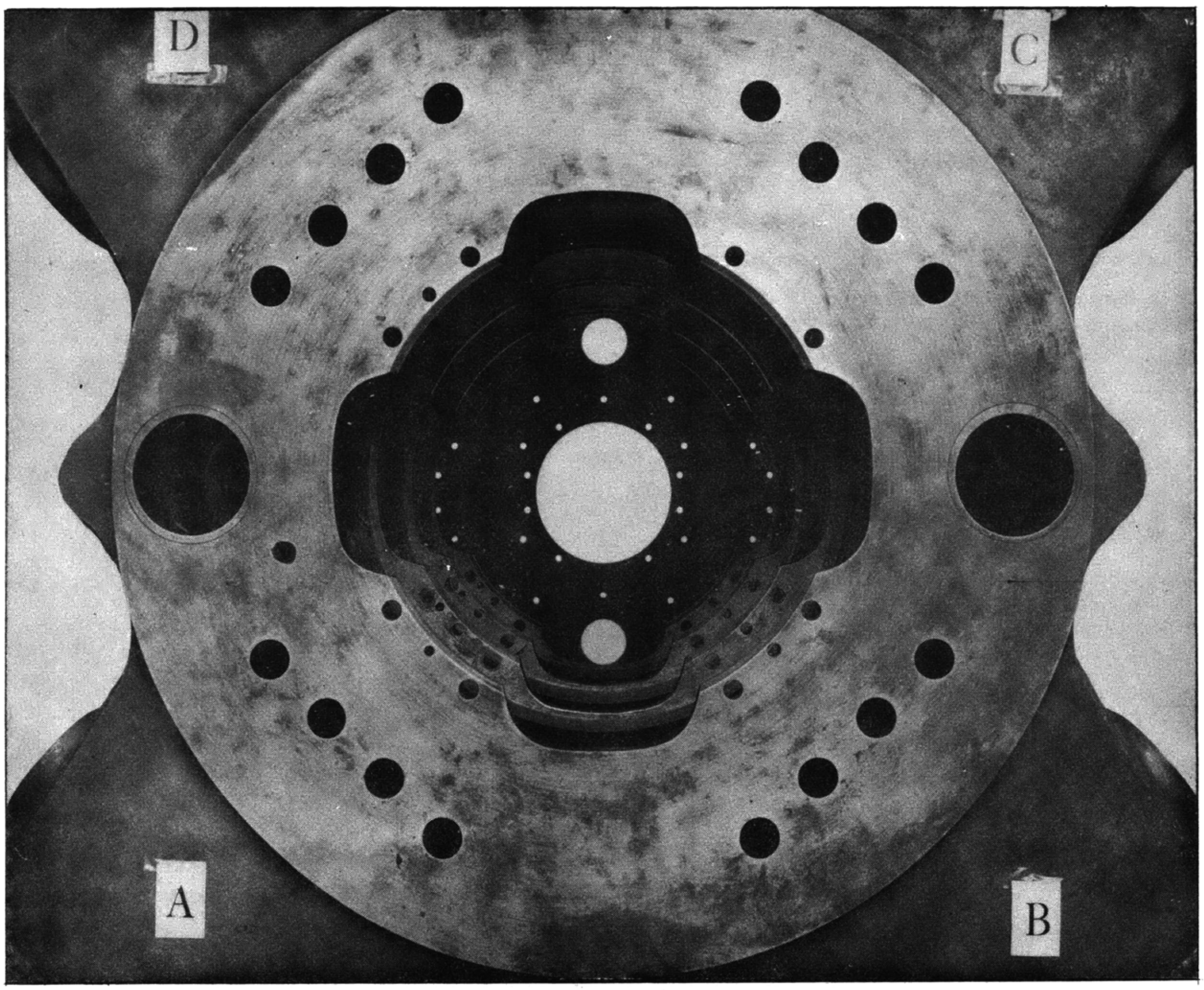


Figure 15 - Crankcase EMC 16-184-A

View looking toward the top end bulkhead, showing a detail of the four main bearing carrier lugs drilled for receiving the cross-shaped main bearing carriers.

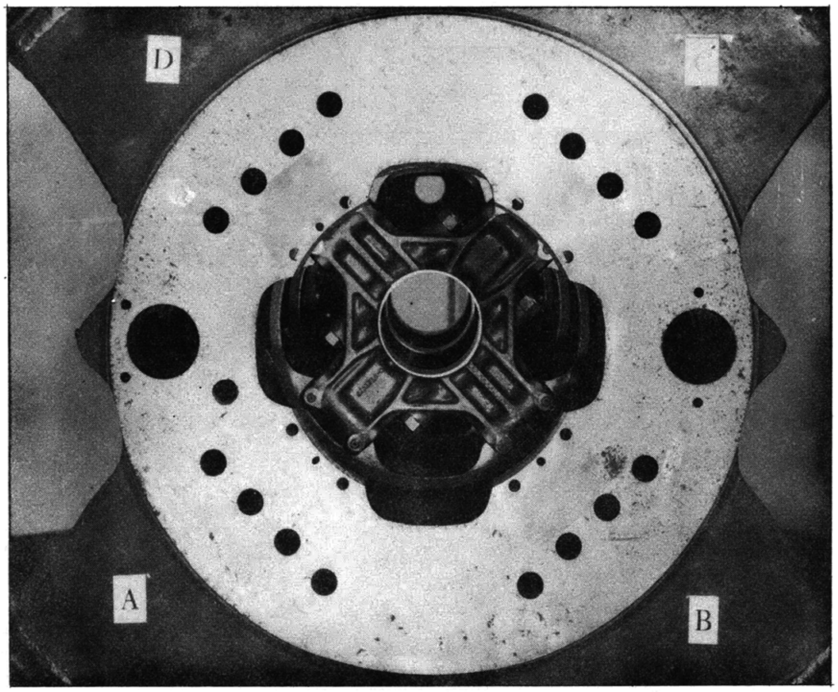


Figure 16 - Crankcase GMC 16-184

View showing the crankcase with the intermediate main bearing carriers installed.

Note here, and in Figure 21, that each main bearing carrier is in two parts, held together by four long bolts. The parting line is serrated to hold the two halves accurately in position.

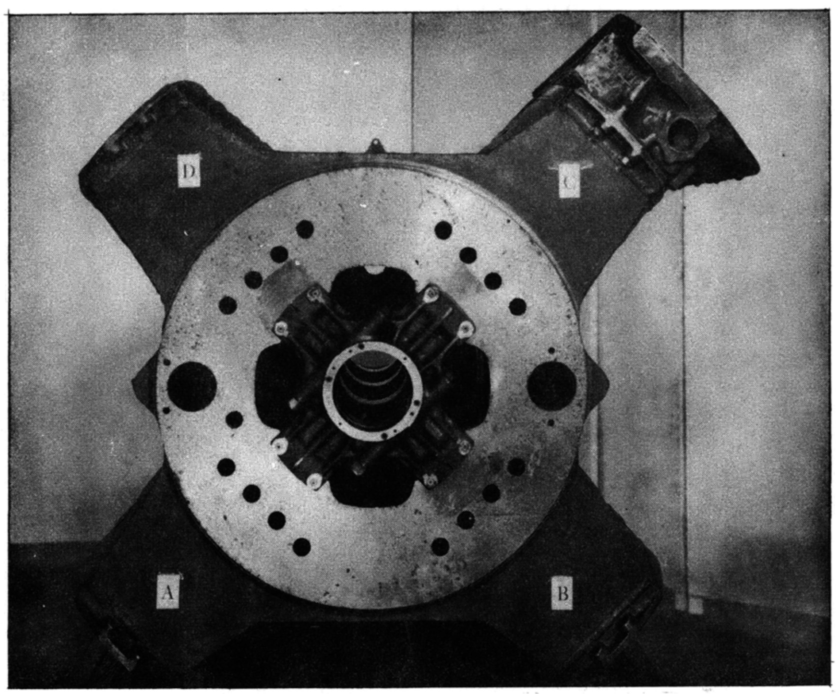
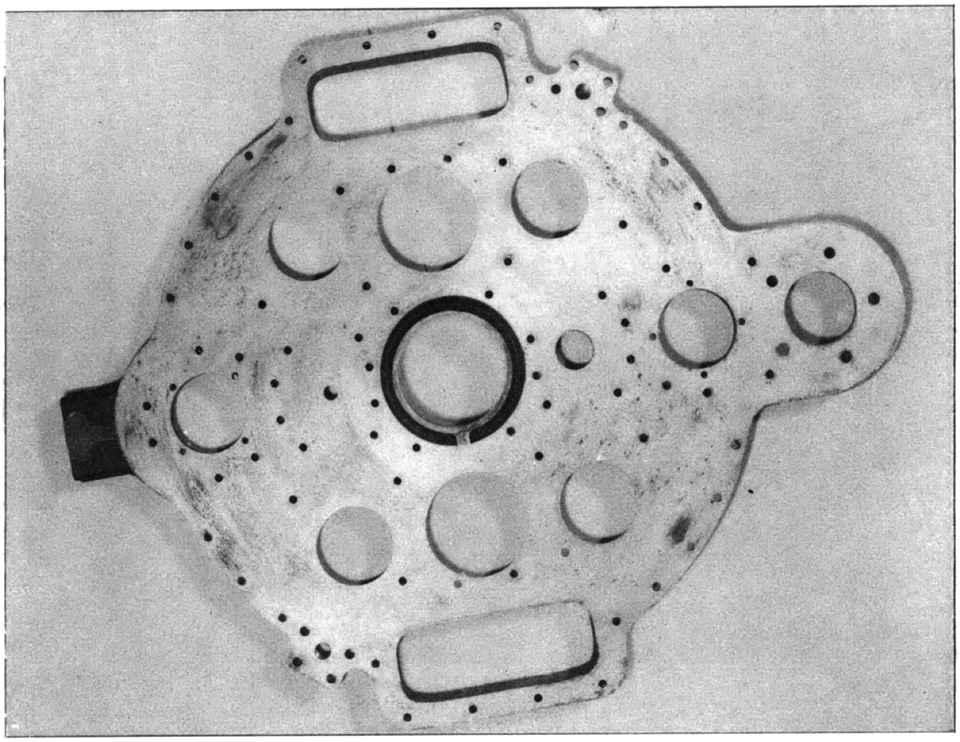


Figure 17 - Crankcase GMC 16-184

This view shows the crankcase with the intermediate and lower main bearing carriers installed. An exhaust housing is shown mounted on Cylinder Bank C.



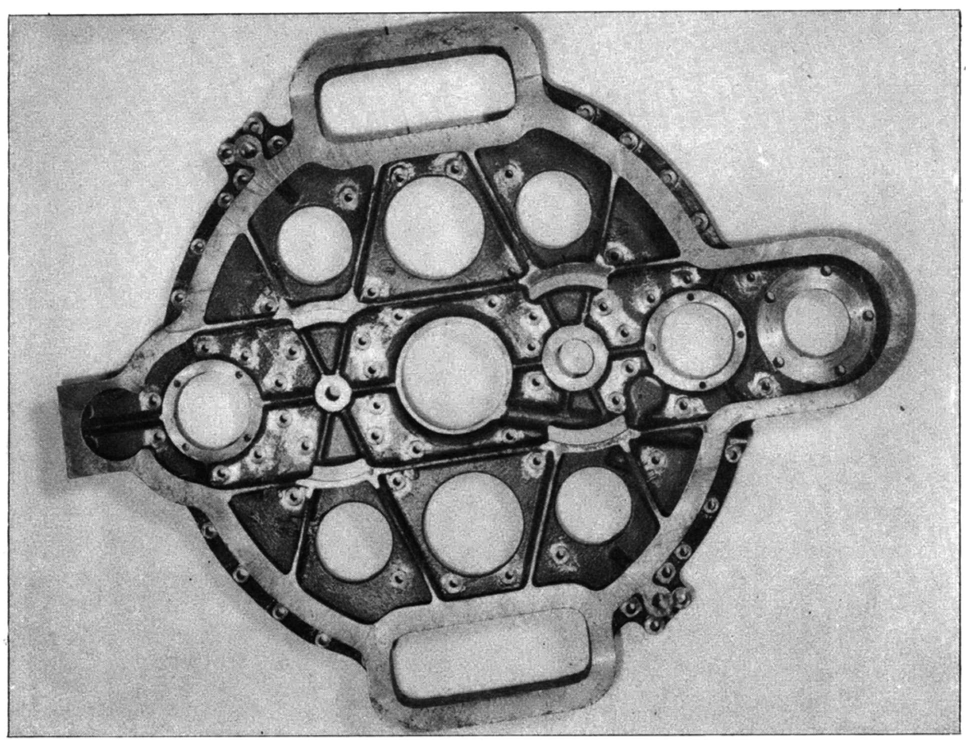


Figure 18 - Timing Gear Housing for Crankcases GMC 16-184 and EMC 16-184-A

The upper view shows the machined face of the timing gear housing which fits against the top side of the crankcase. The lower view shows the outer side of the gear housing.

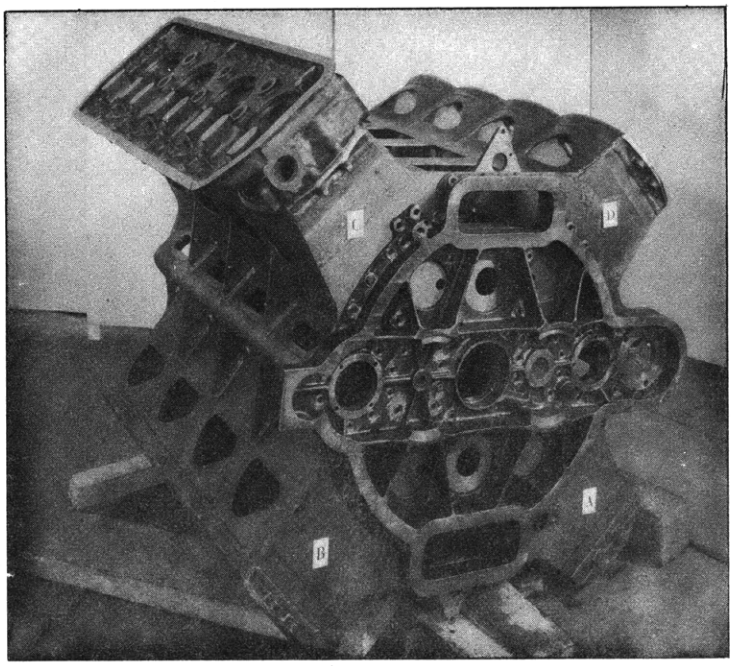


Figure 19 - Crankcase GMC 16-184

View of the crankcase showing the timing gear housing mounted in place. The exhaust housing mounted on Cylinder Bank C shows the inner detail of the housing.

The unflanged triangular air-box handholes are shown.

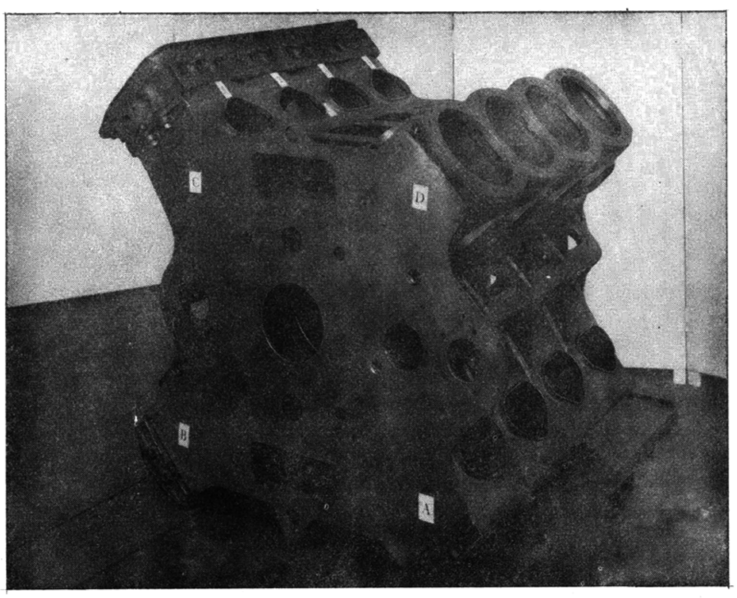


Figure 20 - Crankcase EMC 16-184-A

A view of the crankcase showing the machined topside bulkhead. The timing gear housing is secured to this bulkhead.

It will be observed that the triangular handholes in the airbox walls which are shown in Cylinder Bank C, below the cylinder numbers, are flanged. In crankcase GMC 16-184 these holes have unflanged edges.

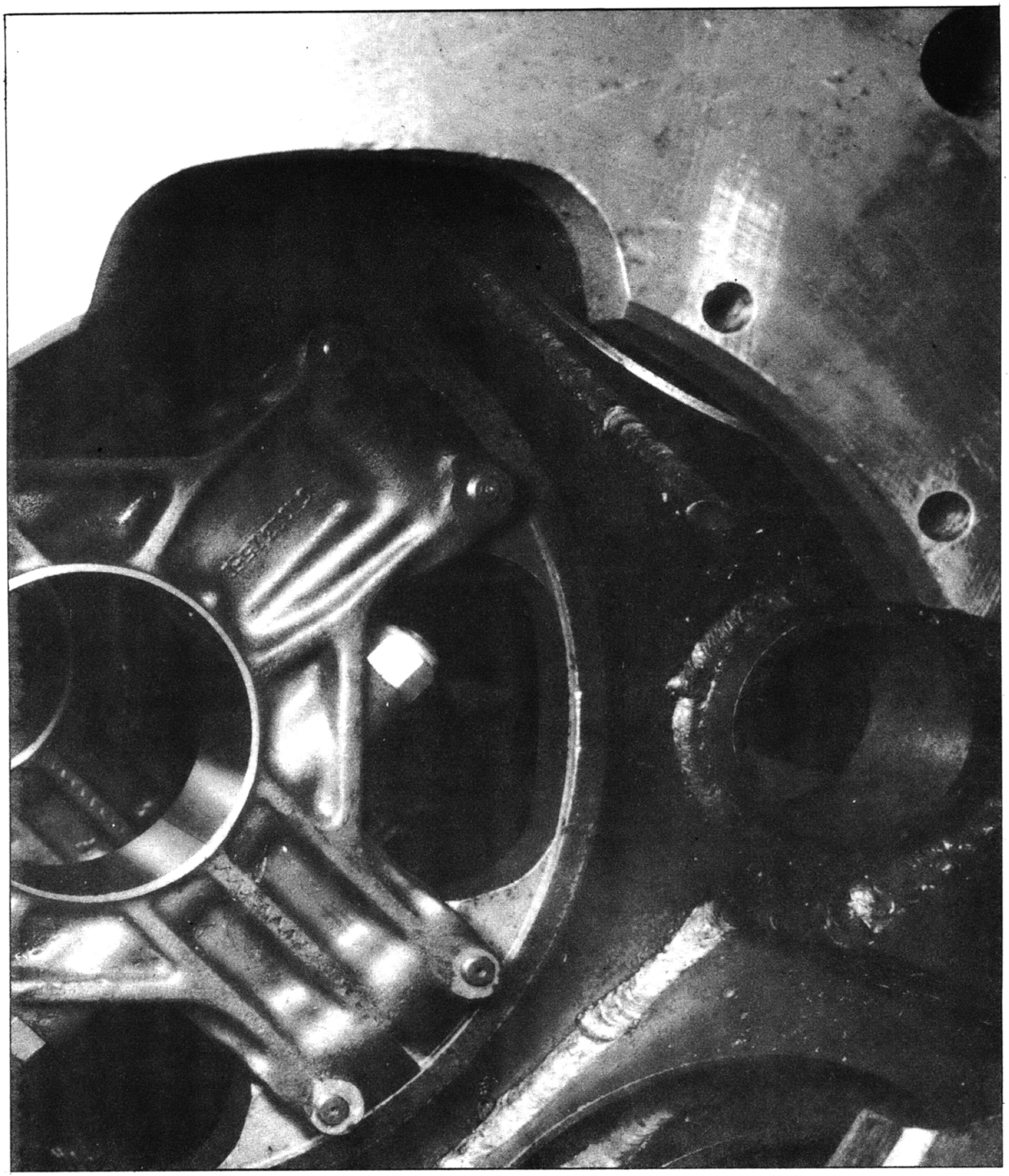


Figure 21 - Detail of Inner Walls and Bulkhead and Intermediate Main Bearing Carrier of Crankcase GMC 16-184

The view shows the greater part of an intermediate main bearing carrier, and the mode of securing it to the supporting bulkhead. The serrated joint between the two halves of the bearing carrier, and the bolts holding the two halves together, can be clearly seen. The camshaft bearing sleeve is shown at the right of the view. Part of a cylinder opening in the double walls of the coolant chamber is shown in the lower right corner.

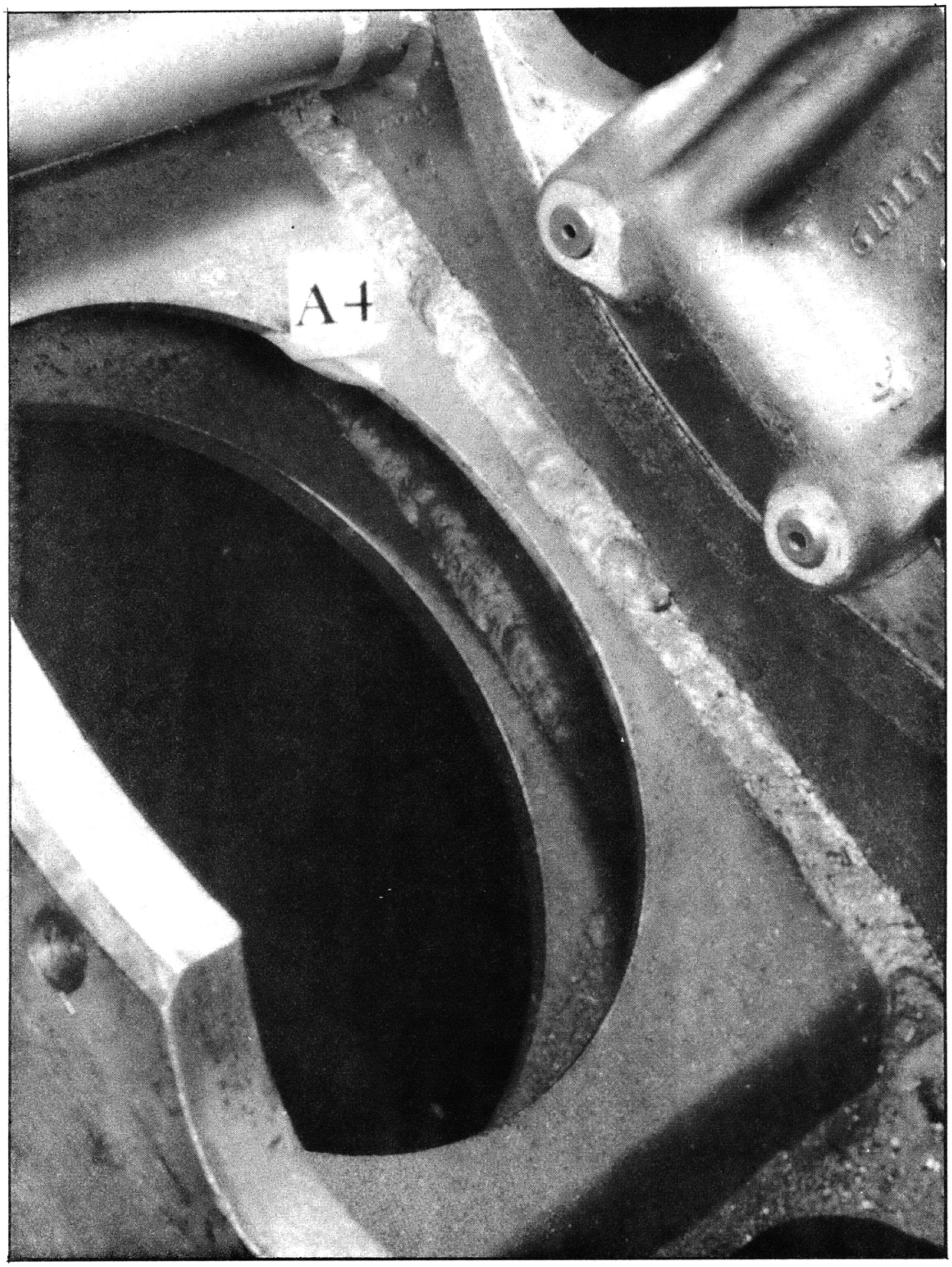


Figure 22 - Cylinder Opening in Crankcase GMC 16-184

The designation A4 appears at the edge of a typical cylinder opening in the inner walls of the crankcase. A cylinder sleeve is screwed into this opening, whereas it is welded in place in the EMC 16-184-A crankcase.

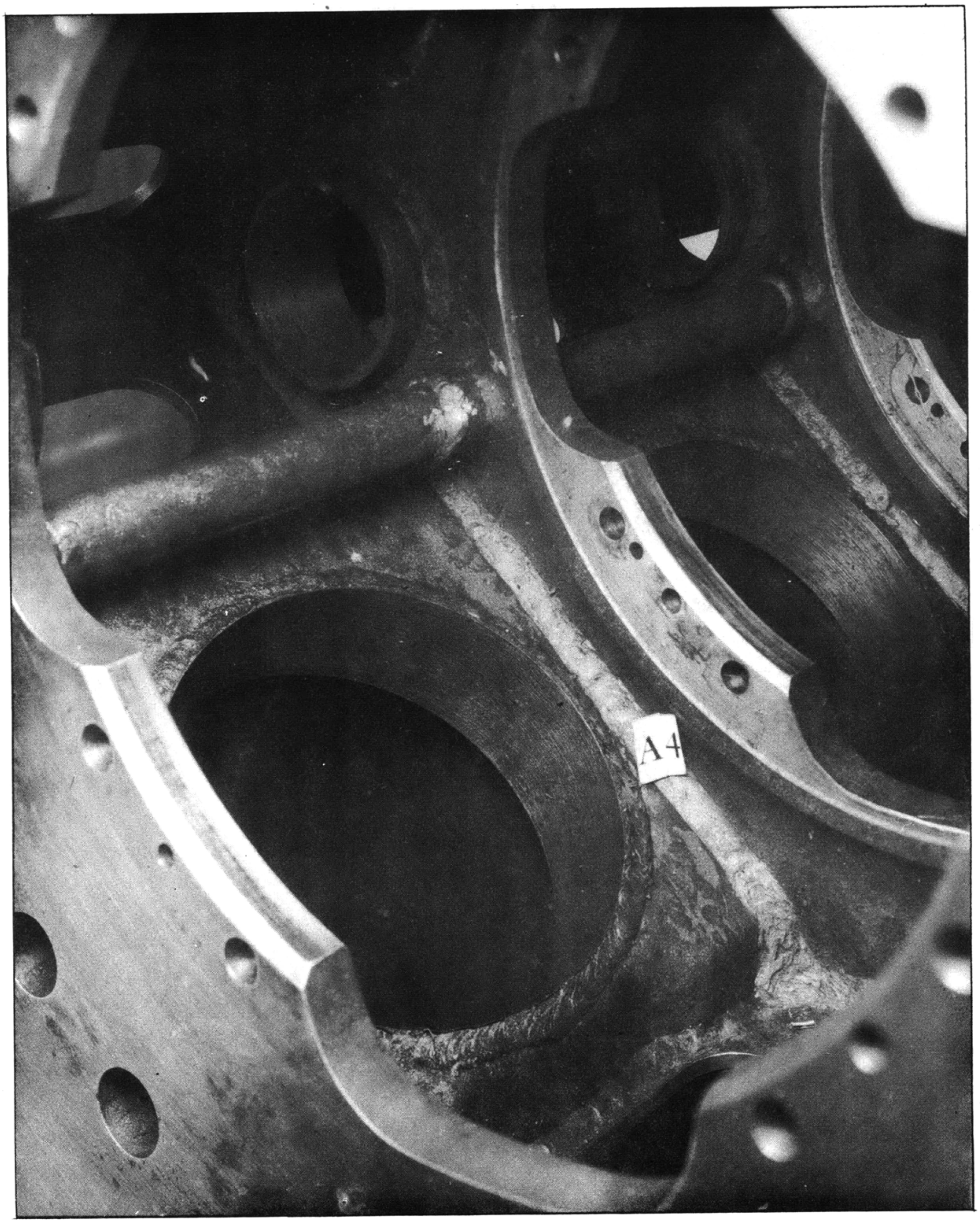


Figure 23 - Detail of Cylinder Opening in Crankcase EMC 16-184-A

The designation A4 appears at the edge of the cylinder opening. The cylinder sleeve is here shown welded in place. The tube shown to the rear of the cylinder opening is an oil duct. Behind the duct are shown the camshaft bearing sleeves which are welded in the bulkheads. The main bearing carriers are not in position.

APPENDIX 2

This is a synopsis of the "Preliminary Report on Stress Analysis of Crankcase for 16-184 Diesel Engine built by the Electro-Motive Corporation," dated 10 June 1941, Reference (2).

A preliminary stress analysis of the crankcase for the Type 16-184 diesel engine was made by the Bureau of Ships to determine the magnitude and direction of the forces transmitted through the head and the main bearings to the crankcase. This analysis was made to furnish a guide for the static tests at the Taylor Model Basin.

No indicator diagrams for the engine were available, so that assumptions of cylinder conditions had to be made and an indicator card constructed on these assumptions. A polar diagram of one cylinder was plotted from the indicator diagram. A gas pressure-load diagram for the crankpin was developed by use of the polar diagram and superpositions in the correct phase relations. The constant inertia and centrifugal forces were added algebraically to the crankpin diagrams. The crankpin diagrams were then combined in such a manner as to develop diagrams showing the magnitude and direction of the forces acting on each main bearing. The main bearing diagrams were then simplified by plotting diagrams of maximum main bearing loads.

It was found to be impossible to duplicate the conditions shown on the final bearing force diagrams. In view of this it was recommended that a static tensile test of each section of the crankcase at 34,000 pounds load be made, using Stresscoat lacquer for locating stress concentrations, and employing strain gages in the section under test and in the adjacent section to determine strains, as supplementary to the Stresscoat observations. Particular care was recommended in the detection of stress concentrations in the outer members of the crankcases.

To determine the behavior of the main bearing carriers when loaded as a beam it was recommended that load be applied 45 degrees to the serrated joint between the carrier forgings. A load limit of 10,000 pounds was specified for the test.

A torsional loading test was recommended and authorized by the Bureau of Ships to determine the points of high stress concentration that would be set up by the piston side thrust as well as by the torsional loading of the crankcase.

APPENDIX 3

This is a synopsis of the report "Bearing Load Analysis of 16-184-A, 90° X Engine," Report ME-6-68, Project 26, 18 July 1941, Research Laboratories Division, General Motors Corporation, Detroit, Michigan; Reference (5).

A bearing load analysis of the 16-184-A diesel engine as built by the Electro-Motive Corporation was made by the General Motors Research Laboratories. The bearing load diagrams were developed from full-load indicator diagrams and the stresses in the trunnion bolts and slipper retaining rings were determined at calculated loads. The bearing-load diagrams were developed for speeds of 1000 RPM, 1800 RPM, and 2250 RPM. The resultant main bearing loads were obtained by combining

(a) half the combined gas and inertia forces acting on adjacent crankthrows,

(b) centrifugal forces due to adjacent counterweights,

(c) half the centrifugal force due to rotation of the adjacent crankthrow masses,

and in the case of Main Bearing 5 there is

(d) an additional load due to the bevel-gear drive.

The maximum bearing load was indicated by the resultant load curves to be 17,000 pounds.

APPENDIX 4

This is a synopsis of the report "Life Test on a Section of the Welded Steel Crankcase for the 16-184 Radial Diesel Engine," Report ME-3-9; Project 26, 18 November 1938, Research Laboratories Division, General Motors Corporation, Detroit, Michigan; Reference (6).

In early development work a life test was made by the General Motors Research Laboratories in 1938 at Detroit, Michigan, on a section of a welded steel crankcase for the Type 16-184 radial diesel engine. The crankcase for this test was similar to the crankcases of the present test with four banks of cylinders 90 degrees apart but with only two cylinders in each bank.

The construction of this test model provided a means for developing a welding technique for the construction of the complete 16-cylinder crankcases used in the present test. The strength of the welded structure, the bearing carriers, and the carrier lugs was tested under main bearing loads similar to those developed in full-load operation. The load was applied by two eccentric weights keyed to a straight shaft which rotated in the main bearings. A constant rotating force of 30,000 pounds, produced by the rotation of the eccentric weights, acted on the main bearings. The eccentric weights produced higher mean loading on the crankcase than occur in service but did not give the rapid rises of load.

After 204 hours of operation, which produced 20 million load cycles, there was one crack in the outer rib of the top main bearing carrier. The crack was thought to be a casting crack as it started at a small pit. Several welds of the air-box deck plate to the outer bulkheads and several of the welds of the air-box deck plate to the middle bulkhead were cracked.

As a result of this test the bearing carriers were re-designed and forgings were specified for greater strength. The air-box top deck was re-designed as a forging, because it takes the bending load from the weld which secures the deck plate to the air-box wall, placing the weld in tension. The welded steel crankcase construction was considered to be satisfactory as to rigidity and load-carrying capacity.