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ELIMINATION OF GALLING OF PENDULUM-VIBRATION DAMPERS USED IN AIRCRAFT ENGINES

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ADVANCE RESTRICTED REPORT

ELIMINATION OF GALLING OF PENDULUM-VIBRATION DAMPERS USED IN AIRCRAFT ENGINES

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SUMMARY

An analysis of the contact stresses existing in a damper installation of an aircraft engine indicated that galling could be eliminated by increasing the diameters of damper pins and holes. Tests were conducted on a damper assembly of a 12-cylinder V-type aircraft engine having pins enlarged 33 percent and holes sufficiently enlarged to maintain proper tuning. Whereas a standard damper was galled after $63\frac{3}{4}$ hours' operation, the modified damper disk was in perfect condition after $131\frac{1}{2}$ hours' operation. Torsional-vibration surveys indicated that the damping properties of the modified damper disk had not been impaired.

INTRODUCTION

Torsional-vibration dampers of the pendulum type have been very successful in minimizing the vibrations at critical speeds in the operating range of aircraft engines. In many installations, however, serious wearing problems have been encountered in both radial and in-line military engines. Gray and Jenny (reference 1) describe attempts to eliminate damper galling by grit-blasting and lead-plating the damper pins; this process improved the life and the wearing qualities of the dampers but it did not completely or permanently eliminate the excessive wear.

The wear problem was theoretically analyzed at the NACA Cleveland laboratory and a method was developed for reducing the contact stresses of the bifilar-type damper. A damper modified in accordance with the theory was subjected to severe running conditions to determine the effects of the changes on galling and also on damping properties of the system. The resulting wear is shown by photographs.

CALCULATIONS OF CONTACT STRESSES

Analysis and Symbols

The type of damper construction analyzed is illustrated in figure 1. In this bifilar assembly two damper pins are placed in damper holes of larger diameter than the diameter of the pins. The resulting contact area is very small and consequently the stresses involved are high. The stress equation for the static case of an external cylinder contacting an internal cylinder under a load as theoretically developed by Hertz and given in reference 2 is

$$S_{c_{max}} = 0.798\sqrt{\frac{p\,\dfrac{D_1 - D_2}{D_1 D_2}}{\left(\dfrac{1-\gamma_1^{\,2}}{E_1} + \dfrac{1-\gamma_2^{\,2}}{E_2}\right)}} \qquad (1)$$

where

$S_{c_{max}}$ maximum compressive stress, pounds per square inch

p load per linear inch, pounds per inch

D_1 inside diameter of damper hole or twice radius of curvature of track that makes contact with damper pin, inches

D_2 diameter of damper pin, inches

E_1, E_2 modulus of elasticity of damper carrier or bushing and pin materials, respectively, pounds per square inch

γ_1, γ_2 Poisson's ratio for damper carrier or bushing and pin materials, respectively

Two assumptions are made when equation (1) is used: (1) that the damper weight remains motionless relative to the carrier, and (2) that the galling is caused by compressive loading of the surface material of the damper parts.

The load p acting on the damper parts is mainly composed of the centrifugal force on the damper weight. The effects of gravity and windage are so small compared with the centrifugal force that they may be neglected.

Then

$$p = \frac{Wr}{gl}\left(\frac{2\pi N}{60}\right)^2 \tag{2}$$

where

W damper weight that is suspended by two damper pins, pounds (In order to calculate the stress for the carrier, the weight of the pins must be added.)

r distance from center of gravity of damper weight to axis of rotation of carrier, inches

g gravitational acceleration, inches per second per second

l contact length of part for which stress is being determined, inches (For bifilar-type construction, the effective contact length involves contact length of two damper pins.)

N carrier speed, rpm

When substitutions are made for p, g, E_1, E_2, γ_1, and γ_2 in equation (1) and it is assumed that the damper parts are made of steel, which is true for all extensively used pendulum dampers, the equation becomes

$$S_{c_{max}} = 17.25N\sqrt{\frac{Wr}{l}\left(\frac{D_1 - D_2}{D_1D_2}\right)} \tag{3}$$

Contact Stress in Standard Damper Design

Equation (3) can be used to work out an example of an existing standard damper installation used in a 12-cylinder V-type engine; the damper disk (carrier) of this system is shown in figure 2. In this particular case, the total damping weight consists of six weights, each independently suspended by two pins. Three of the weights are used to suppress $4\frac{1}{2}$-order torsional vibrations and the other three to suppress $7\frac{1}{2}$-order vibrations.

The values as used in equation (3) for the $4\frac{1}{2}$- and the $7\frac{1}{2}$-order dampers are:

	$4\frac{1}{2}$-order damper	$7\frac{1}{2}$-order damper
W, pounds	1.239	0.932
r, inches	4.040	4.178
D_1, inch	.564	.448
D_2, inch	.375	.375
l, inch	2 × 0.281	2 × 0.281

At an engine speed of 3000 rpm, or take-off speed, the maximum stress in the $4\frac{1}{2}$-order holes of the damper disk is

$$S_{c_{max}} = 17.25 \times 3000 \sqrt{\frac{1.239 \times 4.040}{0.562}\left(\frac{0.564 - 0.375}{0.564 \times 0.375}\right)}$$

$$= 146,000 \text{ pounds per square inch}$$

For the $7\frac{1}{2}$-order damper-disk holes at take-off speed of the engine

$$S_{c_{max}} = 17.25 \times 3000 \sqrt{\frac{0.932 \times 4.178}{0.562}\left(\frac{0.448 - 0.375}{0.448 \times 0.375}\right)}$$

$$= 90,500 \text{ pounds per square inch}$$

The $4\frac{1}{2}$-order holes were seriously galled in three damper failures inspected at the Cleveland laboratory but in all cases the $7\frac{1}{2}$-order holes were entirely free from galling. This condition would indicate that the stress in the $7\frac{1}{2}$-order holes were well within safe limits for the materials used and that the failure in the $4\frac{1}{2}$-order holes could be prevented by reducing the stresses.

Reduction of Contact Stresses

The tuning equation for pendulum dampers (reference 3, p. 517) states

$$n = \sqrt{\frac{R}{L}}$$

where

n order number (multiple of engine speed) of harmonic exciting torque to be dampened by pendulum

R distance from center of oscillation of damper weight to axis of rotation of carrier, inches

L length of pendulum arm, inches

The length of the pendulum arm L is equal to the difference in damper hole and pin diameters $D_1 - D_2$. The distance from the center of oscillation to the axis of rotation R is equal to the distance from the center of gravity of the damper weight to the axis of rotation r minus the difference in hole and pin diameters $D_1 - D_2$. Therefore

$$n = \sqrt{\frac{r - (D_1 - D_2)}{(D_1 - D_2)}}$$

If the relation $D_1 - D_2$ and the value of r remain constant, the damper-pin and the damper-hole diameters may be varied at will without affecting the tuning. If the values of W, r, and l in equation (3) are relatively unchanged for the modified design, the maximum stress is inversely proportional to the square root of the product of the damper-pin diameter and the damper-hole diameter. That is

$$S_{c_{max}} \propto \sqrt{\frac{1}{D_1 D_2}}$$

From this proportionality it is obvious that increases in D_1 and D_2 will result in a reduction of the maximum compressive stress.

If the hole diameter D_1 is increased to 0.688 inch and the pin diameter D_2 is increased to 0.497 inch, the stress for the $4\frac{1}{2}$-order damper becomes

$$S_{c_{max}} = 17.25 \times 3000 \sqrt{\frac{1.309 \times 4.080}{0.562}\left(\frac{0.688 - 0.497}{0.688 \times 0.497}\right)}$$

$$= 119{,}100 \text{ pounds per square inch}$$

The weight was slightly increased (from 1.239 to 1.309 lb) because of the increased pin size and the value of r was increased 0.040 inch to allow relocation of the holes to permit remachining.

Assumptions

As stated earlier, two main assumptions were made when deriving the stress equations: (1) that the damper weight remains motionless relative to the carrier, and (2) that the galling is caused by compressive loading of the surface material of the damper parts.

From an inspection of the galled surfaces shown in figure 3, it is difficult to determine whether the galling resulted from a subsurface shear of the softer metal beneath the hard case or from a compressive failure of the case itself. The subsurface stresses are linearly related to the contact compressive stresses and lowering the compressive stresses would reduce the subsurface stresses. When the damper pins and holes are enlarged, the galling will therefore be alleviated regardless of where it begins.

When the damper is revolving in the crankcase at speeds other than the critical speed, the speed at which the exciting frequency equals the natural frequency of the system, the weights are at rest or are oscillating at very small amplitudes relative to the carrier. Under these conditions the existing stresses should closely agree with the stresses calculated by using the static equation (3). The critical speed causes the damper weights to oscillate. This oscillation causes the pin to roll at high speeds on the surface of the damper hole. The effects of the high rolling velocities on the actual stresses are unknown and therefore could not be evaluated for this investigation. In order to estimate the stresses in oscillating damper parts, Wilson (reference 3, pp. 594-596) used an empirical formula derived for gear-tooth contact stresses that takes into account the gear speed. The sliding motion of gear teeth and the rolling motion of damper pins differ greatly. Regardless of the effects of rolling velocities, the stress is a function of the damper pin and bushing or hole diameters and can be reduced by increasing these diameters.

RESULTS AND DISCUSSION

The bifilar-type torsional-vibration damper that is widely used in radial and in-line engines consists of a weight suspended from a rotating carrier by two damper pins. High stresses are inherent in this design because the centrifugal force acting on the damper weight is supported by a very small effective contact area. The small contact area results from the considerable difference in the diameter of the pin and the diameter of the damper hole. Trouble often develops in the form of excessive wear or galling of the damper holes and pins as a result of the high stresses. Many methods have been tried to reduce the excessive wear but no published report is available on the attempts to increase the effective contact area and thus decrease the high stresses that cause wear. Contact areas can be increased without affecting damper tuning by enlarging the pin and hole diameters exactly the same dimensional quantity. In most applications the required pin and hole sizes may be obtained in existing assemblies without a complicated redesign.

Figures 2 and 3 show a standard damper disk that was removed from a 12-cylinder V-type engine after $63\frac{3}{4}$ hours' running time following the last overhaul and inspection. Galling in the $4\frac{1}{2}$-order holes to the extent shown causes the damper to be out of tune and prevents the pin from rolling in the hole. The damper is therefore ineffective in eliminating torsional vibrations. Two other damper failures that occurred in different engines of the same model indicate that the galling was not caused by something peculiar to the one engine. Actually the galling in these two failures was more severe than in the damper shown in figure 3 after a much shorter time of operation. The maximum theoretical stress in the galled holes is 146,000 pounds per square inch as calculated in the analysis section of the report. By enlarging the pin from 0.375 inch to 0.497 inch and increasing the hole diameter in the existing damper parts to maintain proper tuning, the maximum stress can be reduced to 119,100 pounds per square inch. A damper was modified according to this theory and was installed in an engine. The results are shown by figures 4 and 5. The galling was completely eliminated although the disk shown had been in operation for $131\frac{1}{2}$ hours under severe conditions. The standard and the modified damper disks were run in the same model engine with the same engine mounts, coupling, and dynamometer and under the same test conditions.

Torsional-vibration surveys were conducted on the multicylinder engines with the standard and the modified damper assemblies to determine the effect of the modifications on the damping characteristics. The locations of the critical speeds were unchanged and the amplitudes of the existing torsional vibrations were alike for all orders. The weight of the redesigned system is relatively unaffected because the weight removed by enlarging the holes is replaced with larger pins.

SUMMARY OF RESULTS

From a theoretical analysis and tests of one set of damper pins that were enlarged 33 percent and damper holes that were increased to give correct tuning with the larger pins, the following results were obtained:

1. Excessive wear or galling of pendulum-damper parts was eliminated by increasing the damper-hole size in the existing parts and replacing the damper pin with one of a larger diameter.

2. The damping properties of the pendulum-vibration damper were not affected by the changes of the damper-pin and hole diameters.

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3. Wilson, W. Ker: Practical Solution of Torsional Vibration Problems. Vol. II. John Wiley & Sons, Inc., 2d ed., 1941, pp. 517, 594-596.

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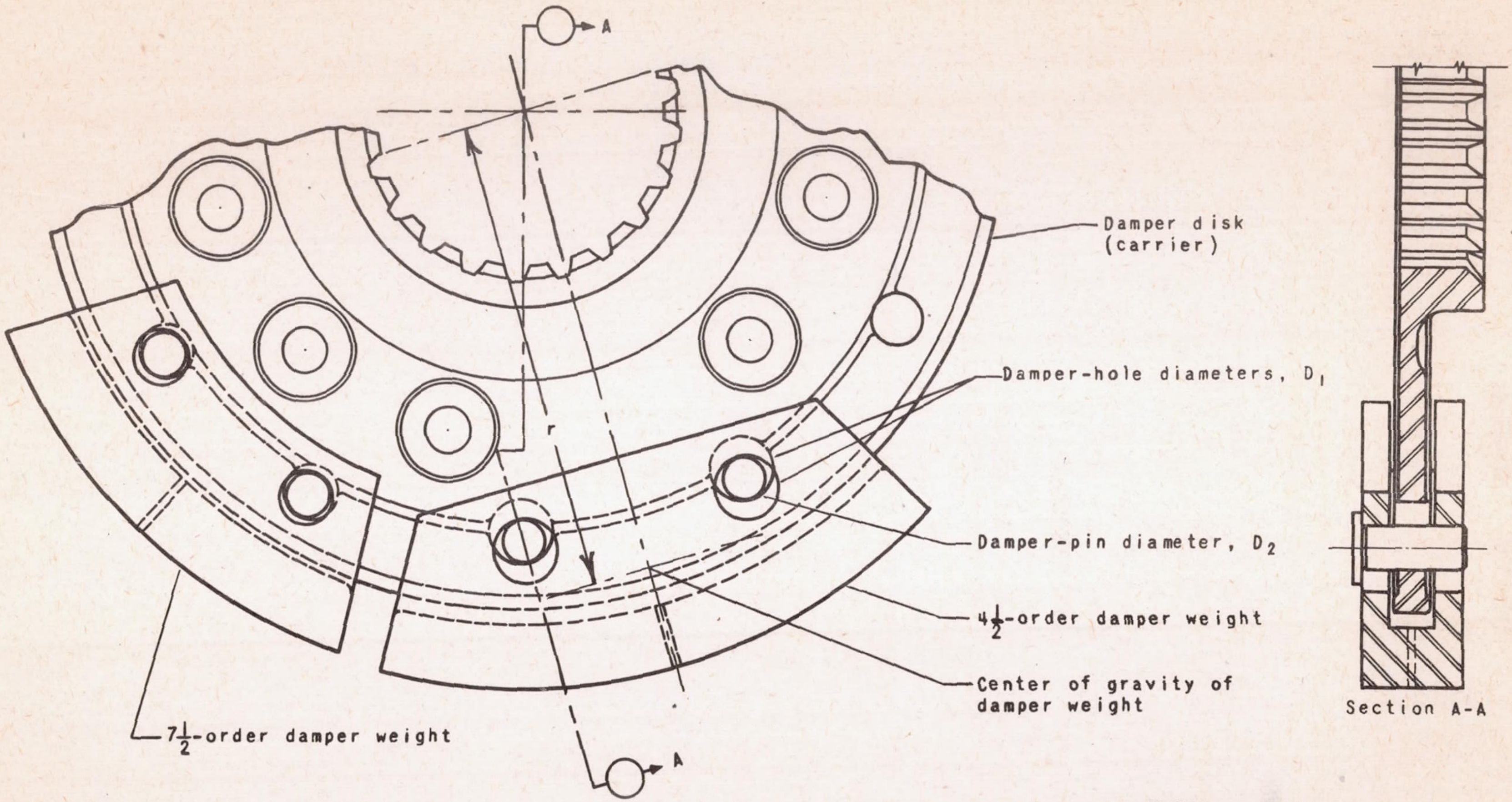

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Figure 1. - The principal parts of a pendulum-type torsional-vibration damper.

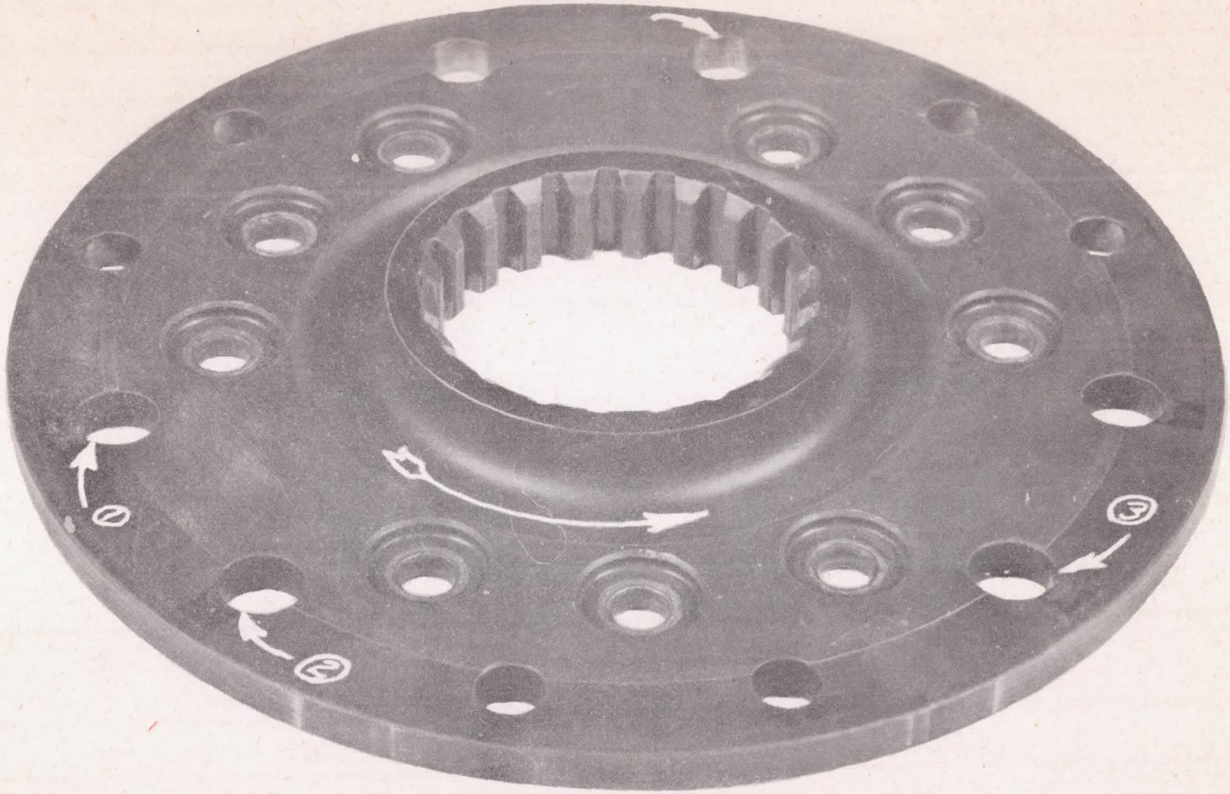

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Figure 2. - Standard damper disk. The 12 outside holes are the damper holes; 6 larger holes for $4\frac{1}{2}$-order weights and 6 smaller holes for $7\frac{1}{2}$-order damper weights. Numbers (1), (2), and (3) indicate galled holes shown enlarged in figure 3.

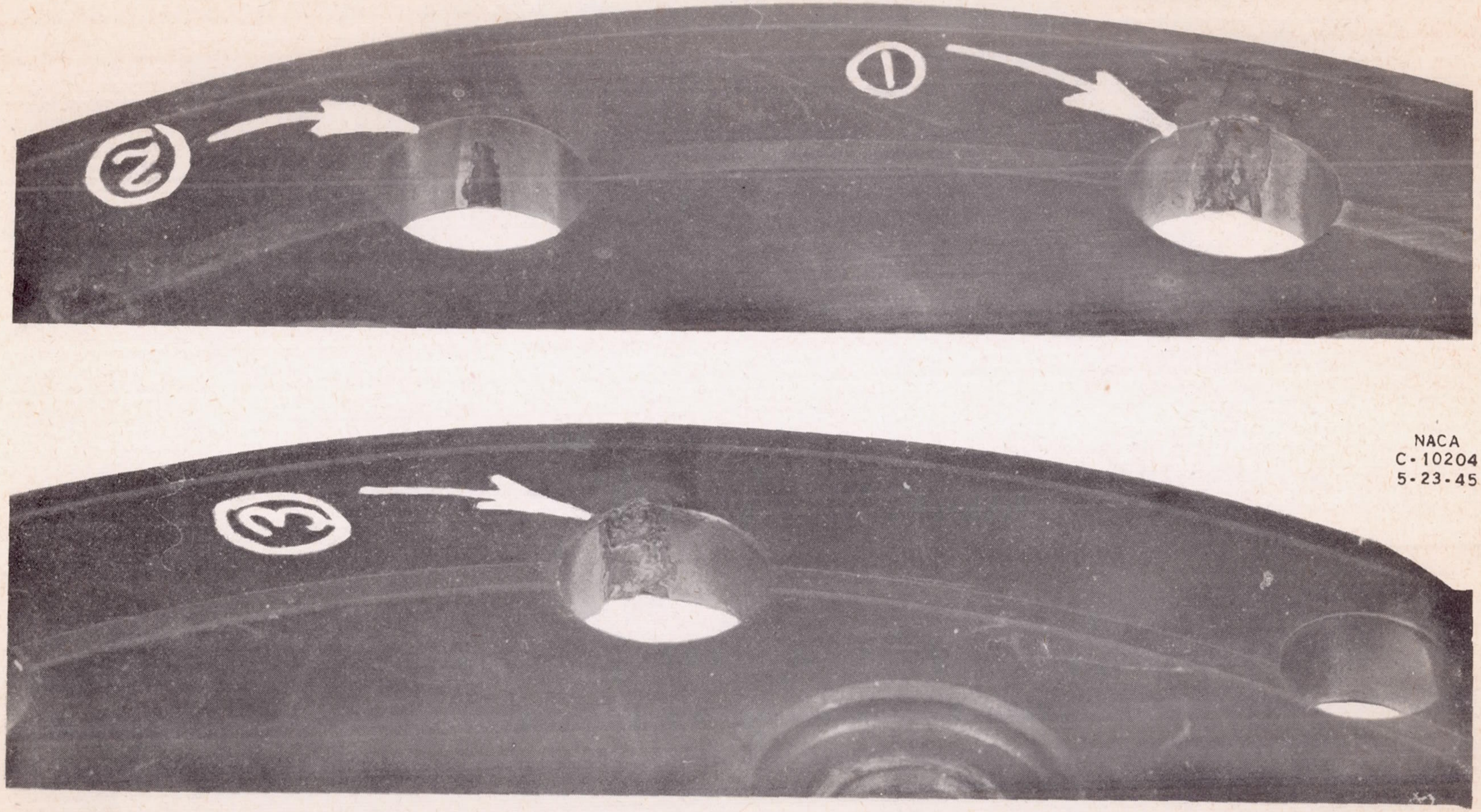

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Figure 3. - Standard damper disk (fig. 2) showing galled holes. Total running time since last inspection, $63\frac{3}{4}$ hours. X2.

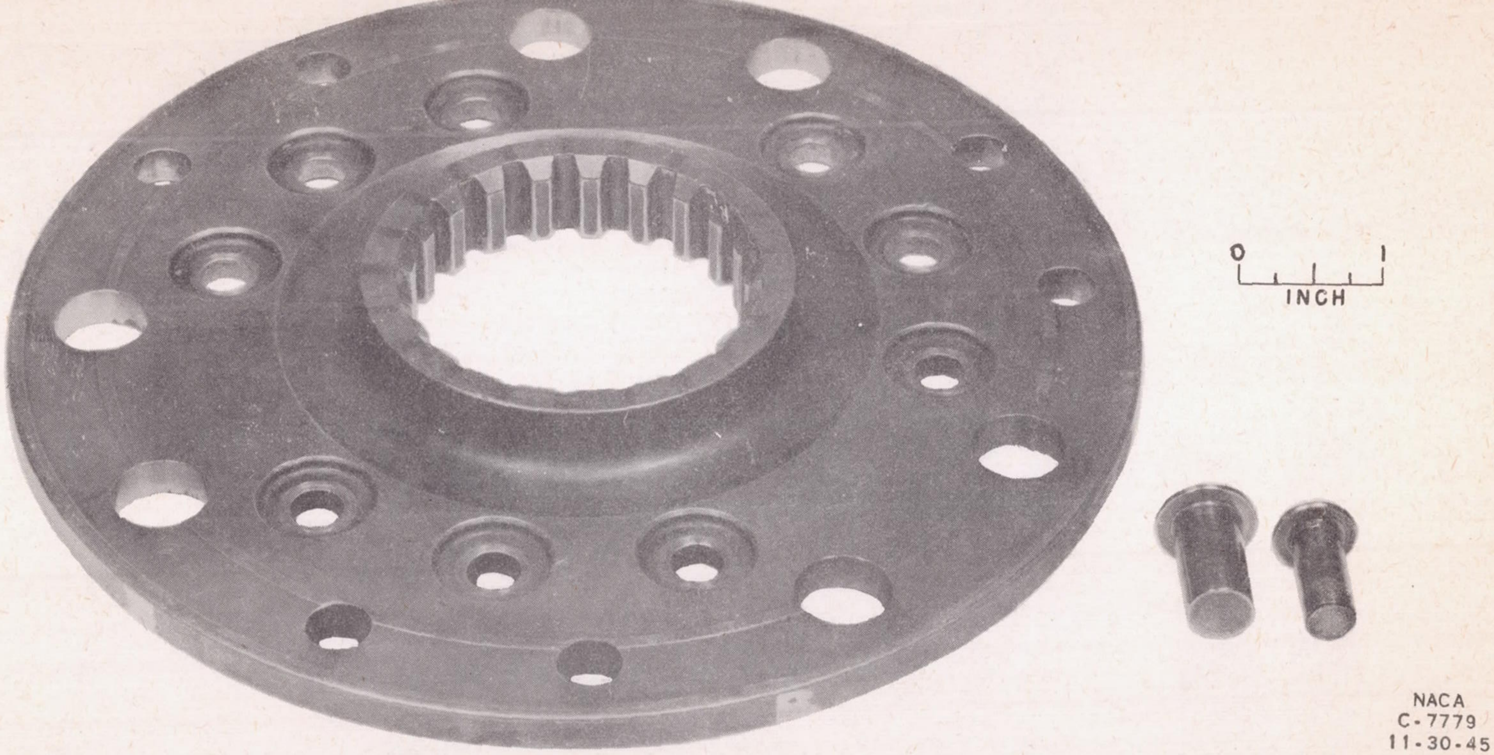

Figure 4. - Modified damper disk with enlarged $4\frac{1}{2}$-order holes and enlarged damper pin compared with standard damper pin.

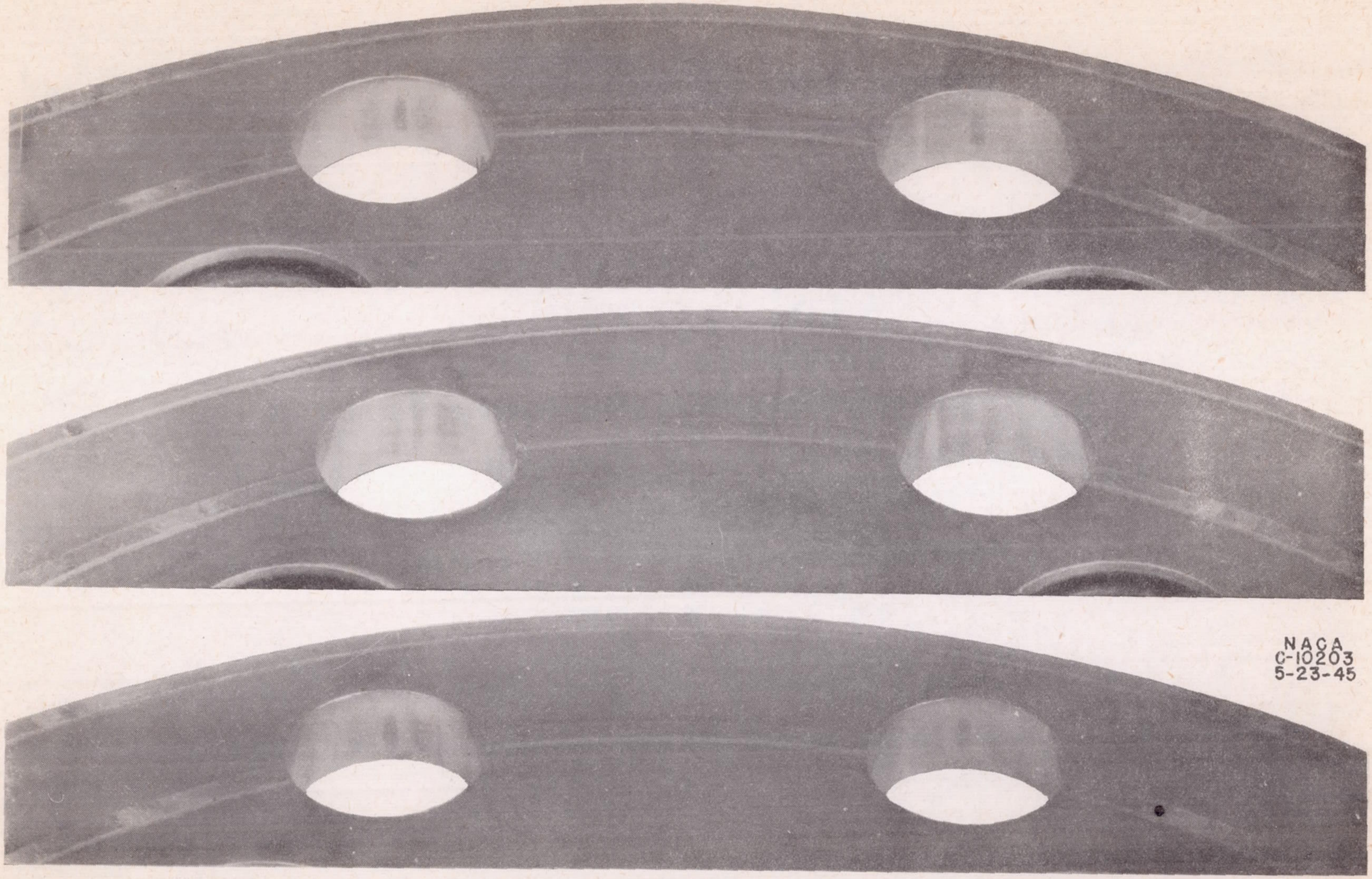

Figure 5. - Three sets of enlarged $4\frac{1}{2}$-order holes in modified damper disk (fig. 4) after $131\frac{1}{2}$ hours' running time.