LYCOMING

DIVISION - THE AVIATION CORPORATION

WILLIAMSPORT, PA.

REPORT NO. X-661

LYCOMING X-7 ENGINE MOUNT DESIGN STUDY

FORM 874 20M-12-43

LYCOMING
DIVISION--THE AVIATION CORPORATION

ENGINEERING MEMORANDUM REPORT NO. X-661

Subject: Lycoming X-7 Engine Mount Design Study
Project: E-1682 - Contract No. W-33-038-AC-564, Item IV
Date of Report: June 9, 1944
Reported by: B. B. Chew B.B.C.
Distribution: Messrs. Hoffman, Carpenter, Analytical File, Engineering Records, AAF Materiel Command (3)

OBJECT:

1. The object of this design study was to investigate several methods of mounting the X-7 engine and to determine which method appeared to be the most practical with regard to size, weight, assembly, serviceability, space occupied by the mount, and vibration characteristics.

CONCLUSIONS:

2. The most desirable engine mount design for the X-7 engine, considering the items mentioned in paragraph 1, is a dynafocal type of mount attached to the engine at the blower housing.

3. This design as shown on LH-762 should satisfactorily withstand the design load factors of a heavy bomber. No major changes to increase the strength are anticipated.

RECOMMENDATIONS:

4. It is recommended that the dynafocal mount attached to the blower housing be used on the X-7 engine, and that the blower housing be built as designed.

5. It is recommended that an experimental stress analysis be made of the blower housing as soon as possible with the design loads being applied through the rubber mounts.

DESCRIPTION OF THE MOUNTS CONSIDERED:

6. When the problem of mounting the X-7 engine was first discussed it was thought desirable to mount the engine in a fashion comparable to the mounting of an in-line engine; that is with supports on the crankcase both front and rear.

Obviously, with this type of mounting the bending moment in the crankcase, due to the propeller overhang and the length of the engine itself, would be less than that which occurs with a cantilever mounting at the rear of the engine.

7. This type of mounting system demands an engine mount which is strong enough to take the weight of the engine and propeller at its outer end. Three general methods of accomplishing this were considered:

a) See Figure 1 on page 17. This mount consists of twin cantilever beams, one on each side of the engine made of tubing or fabricated from sheet metal. This system has the advantage of being simple to design and build and it applies virtually no moment to the crankcase; i.e., the reactions are vertical. Accessibility to the spark plugs, coolant manifolds, etc. is good, detachment of the engine from the mount is simple, and the vibration characteristics can be controlled in the same manner as those of an in-line engine.

   The deciding disadvantage in the case of this design is the fact that a cowl considerably wider than the diameter of the engine must be used to accommodate the two beams. Also, some form of brackets, front and rear, to reach from the crankcase to the beams are necessary, and the resulting mount is inclined to be heavy because of the concentrated loads which must be carried.

b) See Figure 2 on page 18. This design consists of a monocoque structure surrounding the engine, serving as both an engine cowl and a support for the engine. This design, like (a), "cradles" the engine and would probably be the lightest of all the mounts considered.

   The disadvantages are so serious, however, that this suggestion was quickly discarded. Since a stressed skin structure is not well suited to concentrated loads, attachment to the engine should be made at many places. This complicates the vibration isolation problem. Also, with the engine buttoned up in the sheet metal cowling, serviceability is very poor. To remove any substantial part of the cowling-mount would necessitate removal of a large number of body-fit bolts, requiring considerable time. This design might be practicable with a V type engine where removal of but one section would satisfy most service requirements. With a radial engine, however, at least two 180° sections would have to be removed for service, and they could obviously not both be off at the same time without some further method of holding the engine with the cowl off being provided. It is also seen that the process of mounting and demounting the engine, aside from service in the airplane, would be very lengthy.

c) See Figure 3 on page 19 and LH-790. This consists of tubing or forged structural shapes which pass between the cylinder banks to mounting points on the crankcase at the front and rear. The narrowness of the space available between banks results in a structure which is not complete without the crankcase. However, the maximum bending moment in the crankcase is only a small part of the bending moment obtained with a cantilever mounting at the rear of the engine. The mount structure of this design is entirely contained within the cam housing diameter and the dynafocal vibration suspension can be used. Several layouts were made of this scheme of mounting, utilizing various structural shapes and methods of fastening to the crankcase. In every case difficulty was experienced in fitting the struts of the mount between the cylinder banks and the manifolds. Because these struts are loaded as columns and there is no space to increase the radius of gyration of the sections, the sections must be of large area and consequently the struts are heavy. These struts to the front support points can take very little tangential loading, so the propeller torque must be carried through the rear support points. To transmit this torque to the rubber units and to complete the structure of the mount, radial struts or a plate are needed.

The close spacing between banks requires that the mount be fastened to the support points on the crankcase before the manifolds are assembled. Accessibility to the engine as installed in the airplane is generally good, except for the coolant manifolds. This mount has a relatively large number of parts which leads to extra weight and increased difficulty in manufacture and assembly.

As an improvement which would reduce weight and increase serviceability, it was suggested that the intake manifolds serve as the struts to the front support points. This would really amount to increasing the sectional moment of inertia of the crankcase. However, investigation revealed that it would be difficult to secure the front ends of the manifolds to the forged crankcase with sufficient rigidity to accomplish the desired purpose, therefore, the idea was abandoned.

8. The alternative to mounting the engine front and rear in in-line engine fashion is to mount it from the rear as a cantilever, such as is done with single and two row radial engines. This scheme demands that the engine itself be strong enough as a beam to carry its own weight and that of the propeller without

excessive deflection which might damage the crankshaft or the bearings. Two general types of rear mounting were considered:

a) See Figure 4 on page 20 and LH-727. This consists of a system of short struts or links at the rear of the engine which carry a large part of the engine weight through the cylinder heads and the propeller torque through the crankcase. This design relieves the crankcase of the maximum bending moment and because of the large diameter on which the cylinder heads are located an effective vibration isolating mount can be achieved with small connecting members. This mount does not block acess to the usual service-in-plane items and is entirely within the outline of the engine. However, its use prevents accessories from being mounted at the rear ends of the camshafts and causes concentrated loads to be applied to the cylinder heads, which must therefore be strengthened. Also, the distribution of load along the cylinder heads is somewhat uncertain, depending upon machining accuracy, hold-down stud tension, temperature differentials between cylinder head and crankcase, and the angle at which the cylinder bank is set. To insure that the cylinder heads do not carry any torque, a sliding joint must be provided between the heads and the links.

b) See Figure 5 on page 21 and LH-762. This might be termed a standard radial engine type of mounting, although there are some detail differences. Nine rubber vibration isolating units are located between the blower housing and the forged mounting ring. The mounting pads on the blower housing are located between the super-charger outlets to the manifolds and consist of circular pilots into which the ball forgings are fitted. The ball joints of these mountings replace the tapered roller bearings of the older dynafocal mounts in the design which the Lord Manufacturing Company has suggested for use with the X-7 engine.

This mount permits accessibility comparable to that of other radial engines and, since it is entirely within the frontal outline of the cylinders, the cowl outline of the installation does not have to be increased because of the mount. Since the blower housing is a rigid part of the engine which can be locally strengthened and because the mounting ring can be located fairly close to the housing, it is possible to use short, light connecting members between the two. This results in an installation with the maximum space available for air ducts, coolant pipes, and accessories at the rear end of the cam housings, etc. The standard radial engine dynafocal suspension can easily be used with this mounting scheme and it has the twin advantages of giving good vibration isolation and being well known to aircraft manufacturers.

A disadvantage of this type of mounting is that the rearmost section of the crankcase and the blower housing must withstand the full bending moment of the engine and propeller. As shown in the section titled "Analysis", however, the stresses produced are not excessive and under normal flight conditions the crankcase deflections should cause no trouble. The sections of the blower housing must be increased in thickness when this scheme of mounting is used, but while increasing the engine weight somewhat the outside dimensions remain unchanged except for the mounting pads. The installed weight of engine and mount may not be much less than with some of the other methods of mounting considered, but the space occupied by this mount is by far the least.

$$M = \frac{6g}{g}\left[W_p(A + 18.2) + W_e(18.2)\right]$$

153500
106760
126020
520480

$$M = F(d) = \frac{W}{g}(d)2g$$

(See Fig. 6. pag 22.) $$M = \frac{2}{\cancel{g}}\left[W_p(A + 18.2) + W_e(18.2)\right]\cdot \cancel{g}$$

g = acceleration gravity.
W_p = weight prop.
W_e ~ weight engine.

$\frac{1}{sec} \times \frac{1}{sec}$

ANALYSIS:

9. In this section an analysis of the blower housing mounting is made since this design appears to be the most practical for mounting the X-7 engine. In this case, as seen from Figure 5, the engine is supported as a cantilever beam and the crankcase section adjacent to the rear flange must carry the full bending moment.

10. There are two maximum conditions of loading which the engine and mount must withstand. These are:

a) For an engine with dual rotation propeller and a 6 g acceleration factor upward.

b) For an engine with a single rotation propeller having 1 rad./sec. angular velocity and a 2 g upward acceleration factor.

11. The bending moment at the rear of the crankcase is found for condition (a) to be: (Refer to the loading diagram on page 22.)

278000

$$M = 6\left[2200(57.5 + 18.2) + 6150(18.2)\right] = 6 \times 278{,}800 \quad ✓$$

$$= 1{,}673{,}000 \text{ lb.-in.}$$

1,673,800

12. The bending moment for condition (b) is composed of two components at right angles, gyroscopic and linear acceleration:

The gyroscopic moment is, assuming the propeller rpm = 500,

$$M = I\,\omega\,\Omega \quad ✓$$

$$= (27{,}000)(500)\left(\frac{6.28}{60}\right)(1) = 1{,}413{,}000 \text{ lb.-in.} \quad ✓$$

and the moment due to the acceleration is

(67.95) 255 500

$$M = 2\left[2200(49.75 + 18.2) + 5850(18.2)\right] = 2 \times 256{,}000 \quad ✓$$

$$= 511{,}000 \text{ lb.-in.}$$ (NOT FROM THEIR FIGURE)

512,000 lb. in.

The resultant of these two moments is

$$\sqrt{1{,}413{,}000 + 511{,}000} = 1{,}\cancel{506{,}000} \text{ lb.-in.}$$ 1,380 lb in

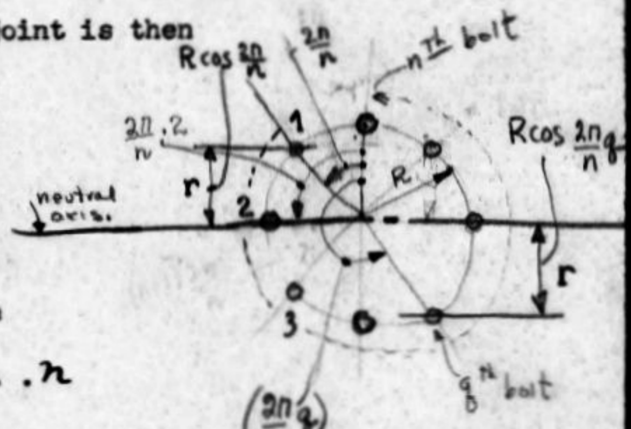
or. $\sqrt{1{,}413{,}000^2 + 511{,}000^2} = 1{,}506{,}000$ lb. in ✓

13. Condition (a) is seen to be most severe and is, therefore, used in the design calculations.

14. The assumption will be made that the average area of crankcase material taking the preload is equal to 1.5 times the minimum section area at the cylinder centerlines. This is determined by planimeter from the layout (p. 23) to be 1.13 square inches. The cross-sectional area of the bolt at the neck diameter of .70 is .385 square inches, making a total area of 1.5 x 1.13 + .385 = 2.08 sq. in.

15. Using the assumptions of the flexure formula, the load carried by any one bolt and section is P = kr, where r is the moment arm of the bolt centerline about the neutral axis. The moment supplied by this bolt and section is then kr^2 and the total resisting moment at the joint is then

$$M = k \sum r^2$$ ✓



for all n bolts. But

$$r = R \cos q \frac{2\pi}{n}$$ ✓

where R = radius to bolt centerline

$$q = 1, 2, 3, \ldots \ldots n$$

so

$$M = kR^2 \sum_{q=1}^{q=n} \cos^2 q \frac{2\pi}{n} = kR^2 \left(\frac{n}{2}\right)$$ ✓

There are n bolts in 2π radians of bolt circle.

For the bolt at the maximum distance, R, from the neutral axis

$$k = \frac{P}{R}$$

so that

$$P_{max.} = \frac{M}{R\left(\frac{n}{2}\right)}$$

$$\frac{kR^2(\frac{n}{2})}{R(n/2)} = kR$$ ✓

$$= \frac{1{,}673{,}000}{11.625 \times 4.50} = 31{,}980 \text{ lb.}$$

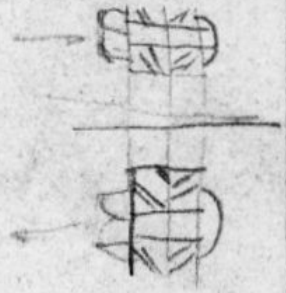
16. With the assumptions given before as to the cross-sectional area of the crankcase which is effective in taking the pre-load, the pre-load required in the bolt to prevent separation of the crankcase is:

$$\frac{1.70}{2.08} \times 31{,}980 = 26{,}100 \text{ lb.}$$

The average tension stress in the bolt becomes

$$\sigma_{av.} = \frac{31{,}980}{.385} = 83{,}100 \text{ psi}$$

For a section on the opposite side of the crankcase, in compression, the maximum load taken by the section surrounding the bolt is



actual load + preload

$$2 \; \frac{1.70}{2.08} \times 31{,}980 = -52{,}200 \text{ lb.}$$

in the crank case

and the average compressive stress at the smallest section is

$$\sigma_{av.} = - \frac{52{,}200}{1.13} = -46{,}100 \text{ psi}$$

crankcase area

17. The total shear area of the nine bolts at the joints is $9 \times \frac{3.14}{4} \times .812^2 = 4.66$ square inches. The average shear stress with a 6 g load becomes therefore:

$$\tau_{av.} = \frac{6\ (2200 + 6150)}{4.66} = 10{,}730 \text{ psi}$$

18. The stresses computed above are not excessive for the steel bolts and the steel crankcase; they represent ultimate conditions of loading and are not repetitive. It may, therefore, be concluded that the crankcase is strong enough to take care of a cantilever mounting.

19. Because of the great amount of stiffening afforded by the monobloc cylinder heads and the irregular shapes of the crankcase and the heads, a reasonably accurate calculation of the crankcase deflection is a somewhat difficult process, even when simplifying assumptions are made. Then, too, the

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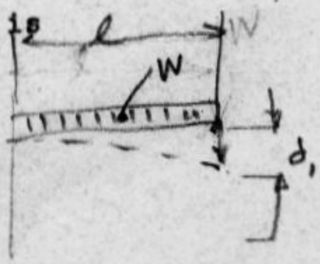
exact amount of deflection which is permissible is an unknown quantity which can only be determined by test. A qualitative estimate of the deflection can be obtained, however, in the following manner.

20. Consider the crankcase without the cylinders to be loaded uniformly by the engine weight along its length of approximately 35 inches. Then the deflection of the front end of the crankcase due to bending is



$$\delta_1 = \frac{Wl^3}{8EI}$$

where $I = AR^2 \sum_{q=1}^{q=n} \cos^2 q \frac{2\pi}{n}$ average

$$= (2.08)(11.63)^2(4.50) = 1265 \text{ in.}^4$$

so that $$\delta_1 = \frac{6150 \times 35^3}{8 \times 29.5 \times 10^6 \times 1265} = .000885 \text{ in.}$$

The deflection due to shear is

$$\delta_2 = \frac{Wl}{2A_sG}$$

where A_s is taken as $(9 \times 2.08)\ 2 = 37.5$ sq. in. to allow for the thickening of the sections between the cylinders.

then $$\delta_2 = \frac{6150 \times 35}{2 \times 37.5 \times 12 \times 10^6} = .000239 \text{ in.}$$

The propeller weight of 2200 lb. at approximately 40 in. ahead of the front face of the crankcase produces deflections in addition.

83×10^6 85

Bending:

$54 \times 10^6 + 3190 \times 10^4$

$$\delta_3 = \left[\frac{2200 \times 40 \times 35^2}{2} + \frac{2200 \times 35^3}{3}\right] \frac{1}{29.5 \times 10^6 \times 1265} = .00228 \text{ in.}$$

Shear:

$$\delta_4 = \frac{2200 \times 35}{37.5 \times 12 \times 10^6} = .000171 \text{ in.}$$

The total deflection of the crankcase alone, loaded with the weight of the engine and propeller, is then

$$.000885 + .000239 + .00228 + .000171 = .003575 \text{ in.}$$

or, say .004 in

21. It has been reported that the addition of the cylinder blocks to the Allison engine crankcase reduced the deflection to one-fifth of that obtained with the crankcase alone. On this basis it would seem likely that the cylinder blocks of the X-7 will reduce the crankcase deflection to somewhere in the neighborhood of 1/5 to 1/10 of the deflection with no cylinder blocks. If we take the first figure, then the 1 g deflection of the front end of the crankcase is .0008 and the 6 g deflection will be .0048. The estimated 1 g deflection is of the same order of magnitude as the usual manufacturing tolerances and is, therefore, not likely to cause trouble. The 6 g deflection will occur infrequently and can only be of short duration. Then, even assuming a 100% error in the conservative estimate made, the resulting maximum deflection of .010" in 35" does not seem likely to cause failure of the crankshaft or bearings.

22. Applying the same equations as before, the maximum load in the 27 studs holding the blower housing to the crankcase is

$$\frac{278,800}{11.0 \times 13.5} = 1872 \text{ lb.}$$

The tension stress in the stud is then

$$\sigma = \frac{1872}{.785 \times .406^2} = 14,440 \text{ psi} \qquad (1\ g)$$

or

$$6 \times 14,440 = 86,660 \text{ psi} \qquad (6\ g)$$

23. The shearing load at the joint between the crankcase and the blower housing is taken by nine dowels around the crankcase through bolts. With an O.D. of 1.19 and an I.D. of .83, the cross-sectional area of each is $.785(1.19^2 - .83^2) = .57$ square inches. The average shearing stress for a 6 g loading is

$$\tau_{av} = \frac{(6150 + 2200)\ 6}{9 \times .57} = 9770 \text{ psi}$$

24. The Lord Manufacturing Company made a preliminary analysis of a dynafocal type mounting for the Lycoming X-7 engine, the results of which are given in their report No. 160.

25. The maximum design condition for which they figured the mount loads was 6 g plus take-off torque and thrust. These loads are tabulated on page 7 of their report. Since their report was prepared, the estimated weight of the propeller has been increased by nearly 1000 lb. Also, the support points on the engine are at a slightly smaller radius. Taking these changes into account it is estimated that the maximum loads will now be as follows:

F_a (torsional direction) = 4500 lb.

F_r (focal direction) = 23,400 lb.

F_s (perpendicular to above) = 1152 lb.

(Link angle = 22-1/2°)

These maximum loads are not located at the same mounting point, but for simplicity in the analysis it will be assumed that they are.

26. In the following discussion, refer to Figure 8 on page 24.

27. The peg projecting from the forged bracket, which is piloted to the blower housing, is loaded transversely by a force

$$\sqrt{23{,}400^2 + 4{,}500^2} = 23{,}800 \text{ lb.} \quad = \underline{27900}$$

4500

and longitudinally by a force of 1152 lb.

The maximum bending stress at the base of the peg is

$$\sigma = \frac{23,800 \times .75}{.0982 \times 1.125^3} + \frac{1152}{.785 \times 1.125^2}$$

$$= 12,770 + 1160 = 13,930 \text{ psi}$$

The maximum shearing stress at the base of the peg

$$\tau_{max} = \frac{4}{3}\left(\frac{23,800}{.785 \times 1.125^2}\right) = 31,950 \text{ psi}$$

The maximum shearing stress tending to tear the peg from the plate is

$$\tau_{max} = \frac{M}{t\pi r^2} = \frac{23,800 \times 1.0}{.438 \times 3.14 \times .56^2} = 55,100 \text{ psi}$$

The bending stress at the fillet

$$\sigma = \frac{23,800 \times 1.125 \times 2.00}{.0982(2.00^4 - 1.50^4)} + \frac{1152}{.785(2.00^2 - 1.50^2)} = 49,800 + 840$$

$$= 50,640 \text{ psi}$$

The bracket has clearance holes to take the studs in the mounting pad, so the studs carry no shear load. The maximum stress in the studs is

$$\sigma = \frac{23,800 \times 2.40}{2.69 \times 2 \times .785 \times .375^2} = 96,100 \text{ psi}$$

The average shearing stress in the pilot of the bracket

$$\tau_{av} = \frac{23,800}{.785(2.75^2 - 1.75^2)} = 6,750 \text{ psi}$$

28. A stress analysis of the blower housing can best be accomplished by experimental means, since its irregular shape is very awkward to fit into stress calculations. However, a few rough checks on the strength of the casting can be made by the use of simplifying assumptions.

29. The mounting pad appears to be marginal in tension on a plane perpendicular to the load, since the approximate cross-sectional area which takes most of the load is

$$2 \times .75 \times 1.00 = 1.50 \text{ sq. in.}$$

and the average tension stress on this area is

$$\sigma = \frac{23,800}{1.50} = 15,900 \text{ psi}$$

This is approximately the yield strength of the aluminum, but actually some of the load is taken in shear, so that the computed stress is higher than the actual.

30. The strength of the cylindrical portion of the blower housing is assured by the 27 steel studs. The aluminum bosses around the studs have cross-sectional areas of

$$1.225 - .196 = 1.029 \text{ sq. in.}$$

so that their stiffness is

$$\frac{1.029 \times 10}{.129 \times 30} = \frac{10.29}{3.87} = 2.65 \text{ times that of the studs.}$$

The required stud preload, to prevent the maximum load of 1872 lb. from separating the blower housing and crankcase is

$$\frac{10.29}{14.16} \times 1872 = \underline{1360 \text{ lb.}}$$

and the compression load in the aluminum which is obtained with this preload is $2 \times 1360 = \underline{3720 \text{ lb}}$.

and the corresponding stress is $\sigma = \frac{3720}{1.029} = \underline{3610 \text{ psi}}$

31. The section of the blower housing at which the mounting pads are placed is not a beam, since the sides are open. It is more like a three link structure in which the load is taken by compression and tension. This scheme of analysis is further justified when it is noticed that the centerlines of the

front and rear walls of the blower housing intersect at very nearly the center of the ball, where the load is applied.

32. The front wall makes an angle of 6° and the rear wall an angle of 33° with the peg centerline. To determine the loads in the front and rear walls due to F_r and F_s ,

$$.105\ P_R + .545\ P_F = F_r$$

$$.840\ P_F - .991\ P_R = F_s\ ;\quad P_F = \frac{.991}{.840} P_R + \frac{1}{.840} F_s$$

$$.105\ P_R + .545(1.18\ P_R + 1.19\ F_s) = F_r$$

$$P_R = \frac{F_r - .649\ F_s}{.738} = 1.355\ F_r - .879\ F_s$$

$$.840\ P_F - 1.343\ F_r + .870\ F_s = F_s$$

$$P_F = \frac{1.343\ F_r + .130\ F_s}{.840} = 1.60\ F_r + .155\ F_s$$

Then

$$P_R = 1.355 \times 23{,}400 - .879 \times 1152$$
$$= 30{,}700 - 1012 = \underline{29{,}688 \text{ lb.}}$$

$$P_F = 1.60 \times 23{,}400 + .155 \times 1152$$
$$= 37{,}400 + 179 = \underline{37{,}579 \text{ lb.}}$$

If a section of the walls 4" wide and 1/2" thick is assumed to be taking the load, then the larger stress, in the front wall, is

$$\sigma = \frac{37{,}579}{4 \times 1/2} = \underline{18{,}790 \text{ psi}}$$

33. If the wall is thought of as a column when loaded in compression, the radius of gyration of the section is .289 times the thickness. With a length of 4", the l/r ratio of the assumed column is 27.7. This is so low that the strength of the section can be said to be determined by the yield strength of

the material, even ignoring the stiffening afforded the section considered by the rest of the wall.

34. The maximum stress calculated above is somewhat greater than the average yield strength of the aluminum casting (15,000 psi) but since the load is actually distributed over a greater area than calculated, this value of stress will probably not be realized.

DISCUSSION:

35. The blower housing mounting is probably not as advantageous from the structural standpoint as are some of the other designs considered. The most important reasons for its use are the fact that there is no interference with the manifolds or accessories, making for simpler assembly and service, and the fact that it follows conventional practice in making the crankcase take the stresses and in using a nine point dynafocal rubber mounting.

36. The calculations which have been made to demonstrate the feasibility of this type of mounting are admittedly sketchy, but it is believed that they do show the inherent strength of the design. The static test of the mounting which has been recommended is not expected to show the need for anything but minor changes, such as adjustment of fillet radii or rearrangement of ribbing. This static test is, however, essential to prove the design and to eliminate any weak points. A calculated stress analysis on so irregular a shape as the blower housing should not be considered conclusive.

X-7 ENGINE MOUNT

PROPOSAL NO. 1

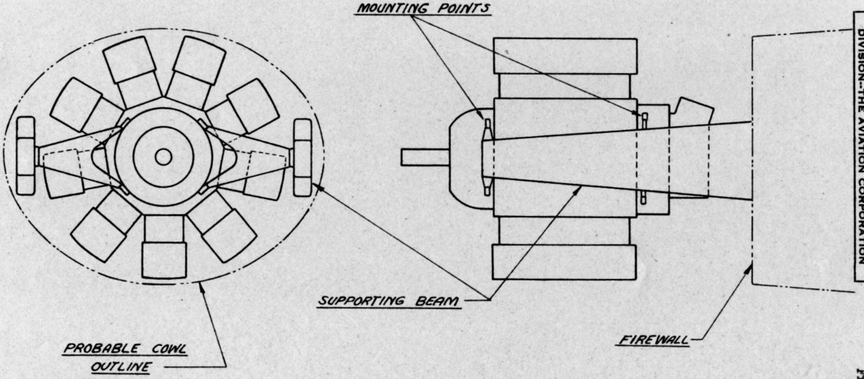


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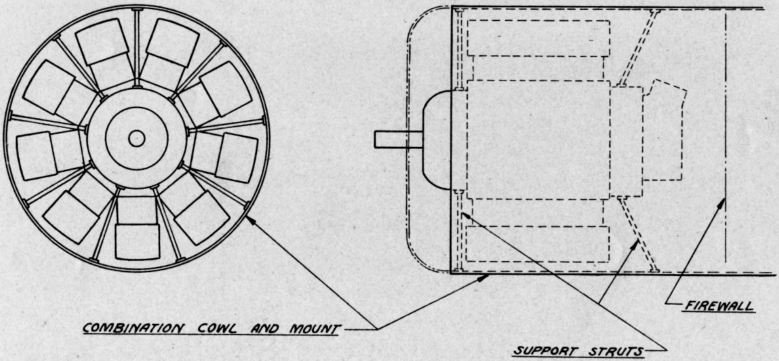
Figure No. 1

X-7 ENGINE MOUNT

PROPOSAL NO. 2



X-7 ENGINE MOUNT

PROPOSAL NO. 3

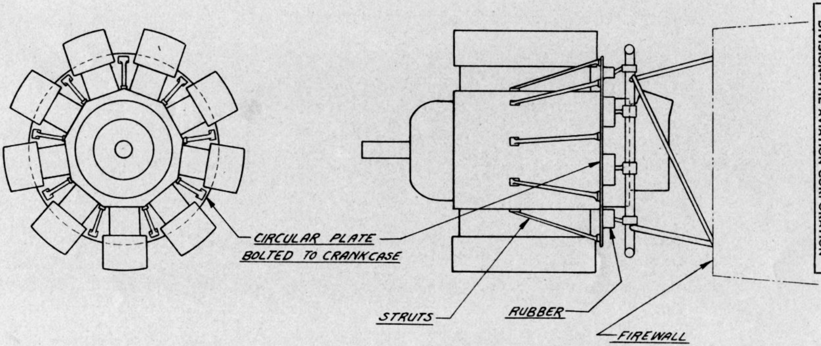


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Figure No. 3

X-7 ENGINE MOUNT

PROPOSAL NO. 4

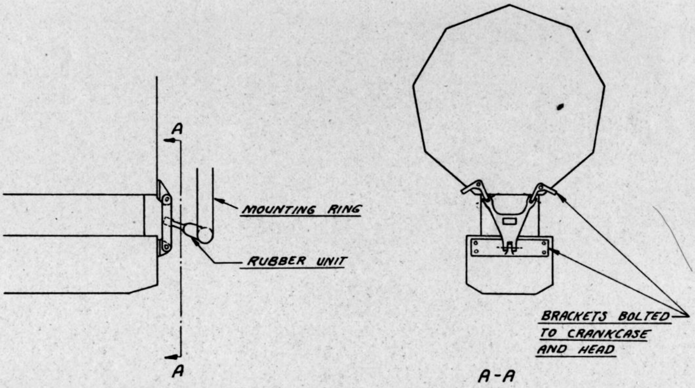

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Figure No. 4

X-7 ENGINE MOUNT

PROPOSAL NO. 5

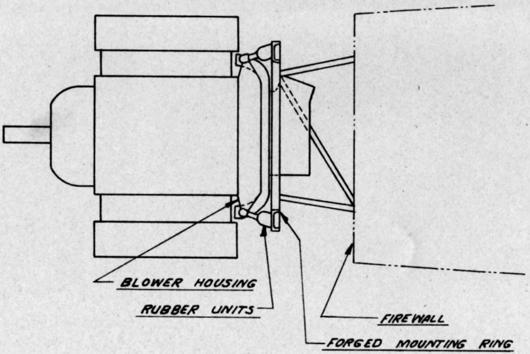
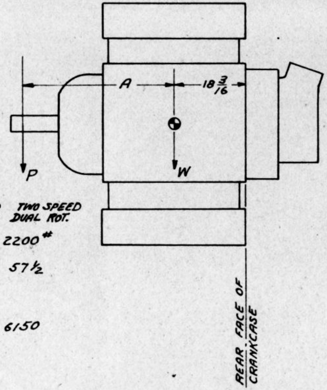


Figure No. 5

X-7 ENGINE MOUNT

DESIGN WEIGHTS



	TWO SPEED SINGLE ROT.	TWO SPEED DUAL ROT.
PROP. WT. 'P'	2200#	2200#
DIST 'A'	49¾	57½
ENGINE AND ACCESSORIES WT - 'W'	5850	6150

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Figure No. 6

FORM 674 20M-12-42

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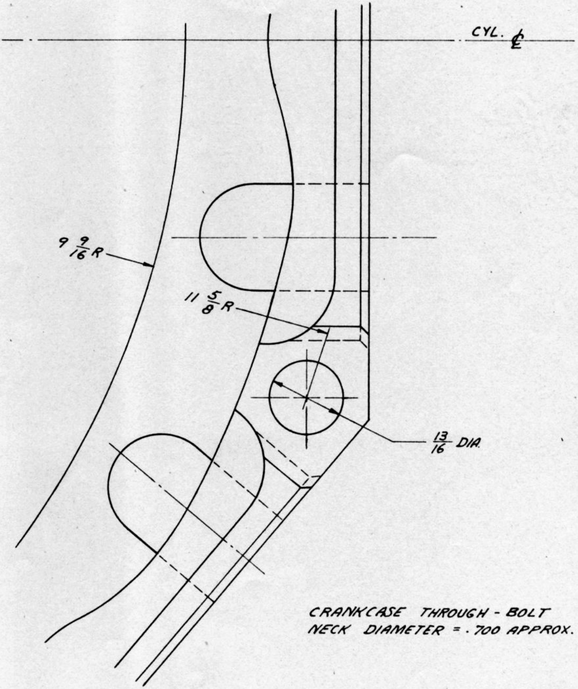
Figure No. 7

X-7 ENGINE CRANKCASE SECTION



CRANKCASE THROUGH - BOLT
NECK DIAMETER = .700 APPROX.