of the elastomer.

After 700,000 cycles, the lower stainless plate was well polished (figure 9). The plate shows signs of rotation of the disc, but the condition of the disc indicates that it is only a small partial rotation. The upper surface of the disc shows very little sign of wear, as can be seen in figure 10. The lower surface (figure 11) is considerably blackened, caused by dust from wear in the center pin. The scratches made during initial machining of the disc are still visible, as are numbers cast into the upper surface. This indicates that the polished stainless steel used as facing was effective in eliminating the abrasion noted in the preliminary tests.

The weights of the disc before and after the test were as follows:

Start	7.81 kg.
Finish	7.71
Weight loss	0.10 kg.

In figures 10 and 11, a circumferential crack is evident on both surfaces of the disc, extending about 120° on the high pressure side. This crack was noted at about 100,000 cycles, and grew throughout the test. It is apparently caused by tensile stresses in the disc surfaces which are due to the expansion of the disc as it is compressed. Friction retains the center portion of the surface, while there is no such restraint at the recessed edge. This crack is shown in section figure 12, which is cut along line 1 (see figure 11), and is approximately 3/8 inch deep. It is not expected that the crack will grow deeper, and the crack is not considered detrimental.

SLIDE BEARING TESTS

Wear rates of virgin and glass filled teflon from manufacturer's literature indicated that the life of these materials is insufficient for use on the Manhattan Bridge. Polyphenylene sulphide (PPS) filled teflon showed promise as a suitable material, exhibiting the lowest wear rate of all commercial teflon materials found, and therefore was selected as the material to be tested. Table 2 shows the wear rates of these materials published by the manufacturer of the PPS filled teflon, Chemplast, Inc.

TABLE 2
Wear Rates of Teflon Materials

Material	Wear Factor, K*
Virgin	$5,000 \times 10^{-10}$
15% glass filled	7×10^{-10}
Polyphenylene Sulphide filled	1×10^{-10}

* discussion of the wear factor is given at the end of this section.

Two pads, 12-1/4 inches and 16 inches in diameter respectively and 3/16 inch thick, were tested in order to determine the effect of pressure on the wear rate.

The arrangement of the test is shown in figure 13. The two specimens bear on opposite sides of a stainless steel clad sliding plate, which is water cooled to carry off excess heat. The slide plate is driven laterally in a reciprocating motion by a horizontal hydraulic jack, thus testing both bearings simultaneously. A nest of jacks below the bottom bearing provides a constant compressive force. An earlier test showed that too rapid a test rate caused the pads to overheat despite cooling in the sliding plate and led to failure of the adhesive, scorching of the pads, a high coefficient of friction and rapid wear. Based upon the results of this previous test, the speed of testing and the loading were selected to maintain the temperature at a tolerable level. A force of 295 kips is used, giving pressures of 2503 psi and 1467 psi on the two pads, respectively. For comparison, the expected mean pressure on the pads in the bridge bearings will be 1000 psi and the maximum pressure will be about 3000 psi.

The speed of the test was chosen as approximately six cycles per minute at a four inch amplitude, or 48 inches per minute, greater than the estimated 12 inches per minute occurring on the bridge. At this speed a temperature of 38 to 42 degrees centigrade in the supporting steel plates just behind the pads was maintained.

The bearing plates with the teflon faces were removed from the machine at intervals of approximately 50,000 cycles and the wear at 9 points on the surface of the pad was measured. Measurements were made with a depth micrometer with .001 inch divisions. The shaft of the micrometer was inserted into holes in a steel plate supported on three studs which fitted into holes in the bearing plate, thus providing a means of exactly locating the micrometer during each reading (figure 14). Measurements, which were estimated to 1/10,000 inch, were made twice at each point and averaged for the nine points. Figure 15 shows the surface of the 16 inch diameter disc and its mating stainless steel plate after 500,000 cycles. As can be seen, there was a transfer of teflon to the stainless steel surface.

The wear vs. cycle plot in figure 16 shows a rapid break-in (two to four thousandths of an inch at about 20,000 cycles) followed by a uniform rate of wear. The 16" diameter pad has a high rate of wear at about 400,000 cycles which occurred on a particularly humid day when excess cooling of the sliding plate caused very heavy condensation. The tests were performed in a fabrication shop where considerable grinding and cutting of steel expansion joints were continuously performed; this generated airborne debris which was quite abrasive and was apparently pulled into the sliding surface by the water. It is interesting to note that the lower pad, where the stainless was above the teflon (the proper positioning of slide pads), shows no such effect even though it was also subjected to heavy condensation.

The solid and dashed straight lines are fitted to data between 24,000 and 500,000 cycles by linear regression; the dash-dot lines are fitted to the 5 peak values of wear. The extremely high wear at 400,000 cycles has been eliminated from the data and replaced with estimated wear values for the purpose of determining the wear rates.

The rate of wear, which is the slope of the wear vs. cycles curve, for each of the above lines was plotted on the wear rate vs. pressure diagram on figure 16. The points lie near two straight lines, one for average wear and the other for a more conservative peak fit. This indicates that wear is not only linearly proportional to total travel, but also to the pressure applied.

Chemplast, Inc., gives a formula for wear as follows:

$$\text{Wear} = \text{pressure} \times \text{velocity} \times \text{time} \times K$$

Where pressure is in pounds per square inch, velocity is in feet per minute and time is in hours. The resulting wear is in inches, and the unit for the wear factor K, is found to be inches cubed-minutes per pound - foot- hour. This type of formula is convenient for running bearings as used in mechanical applications, but can be more conveniently expressed for bridge bearing applications as:

Wear = pressure x distance traveled x K/720

Where wear is in inches, pressure is in pounds per square inch and distance traveled is in inches. The constant K is identical to that in the first formula, being divided by the constant 720 to convert hours to minutes, and feet to inches.

The travel during each cycle of the tests is 8 inches (4 inches in each direction), and, substituting the test data into this formula results in a value of K of 0.9 for the average fit and 1.7 for the peak fit. Both are in good agreement with the value of 1 given by the manufacturer.

Figure 16 also shows the relationship of the coefficient of friction of the teflon-stainless steel combination vs. speed of travel. The bearings on the bridge will translate at a velocity of about 12 inches per minute, indicating a coefficient of friction of 7.5%.

ESTIMATE OF LIFE OF BEARINGS ON BRIDGE

Disc Bearings

From observation of the disc and bearing plates after 700,000 cycles, it is evident that the life of the bearing will be not less than 1,500,000 cycles and could well be greater. The average rotation on the bridge will be only half that in the test, and the average force is only 83% of that in the test. If the pressure in the bearing on the high pressure side is taken as a measure of damage, then one cycle on the bridge represents 0.67 cycles of test, calculated as follows:

Average force on bridge	= 414 kips
Moment to rotate .005 rad.	= 49.5 x 12 in-k
Area of Disc	= 241 sq in.
Section Modulus	= 590 in. cubed
Pressure on bridge	= 414/241 + 49.5 x 12/590
	= 2.72 ksi
Pressure in test	= 499/241 + 99 x 12/590
	= 4.08 ksi
Bridge Cycles	= 2.72/4.08 = 0.67 test cycles

Therefore, the minimum life is estimated as follows:

Life = Est test cycle life/0.67 x bridge cycles per yr

= 1,500,000/(0.67 x 166,000)

= 13.5 years

Teflon Pads

The pad is considered to be worn out when its thickness has been reduced by 1/32 inch, or 0.031 inches. Using a conservative estimate of 0.01 inches for break-in

wear, and a wear factor of 1.7 x 10^{-10}, the life of the teflon pads is estimated as follows:

Area of Teflon Pads	= 415 sq x in
Avg. Pressure	= 414 k/415 sq x in
	= 1 ksi
Dist. Traveled	= 3.68 x 166,000
	= 611,000 in/yr
Wear per year	= pressure x distance x K/720
	= 1000 x 611,000 x 1.7 x 10^{-10}/720
	= 0.000144 inches/yr
Estimated life	= (.031 - .010)/.000144
	= 145 years

CONCLUSIONS

The testing program was a fatigue study. The static strength of the bearings was never in question; teflon slide bearings and a variety of elastomeric bearings have been used on bridges for many years. However, the use of these elements where there is frequent and relatively rapid motion is uncommon, and there existed a clear need to study the performance of the bearings before committing to the full-scale installation.

In the previous section, a comparison is made between test conditions and field conditions, which shows that the tests are a fair representation of actual loadings. Admittedly, there has been no attempt to simulate the myriad deleterious agents which the bearings must withstand, such as prolonged exposure to low temperature and infiltration of dirt and salt. These factors, though not duplicated, have been evaluated and it appears that for each of these factors the bearings either possess adequate resistance, or can be suitably protected.

The life of the disc bearing may well exceed 13.5 years, though the data gathered in testing is not sufficient to support a more generous estimate. Unlike the slide bearing, for which an observed wear rate allows an estimate of useful life, there is no progressive loss of either load or rotation capacity in the disc. The estimate of disc life is limited by a lack of data and not an extrapolation of observed wear.

The data gathered in the testing program indicate that the bearings can be used successfully on the Manhattan Bridge. Figures 17 and 18 show the cross-sections of the actual bearings which will be used on the bridge.

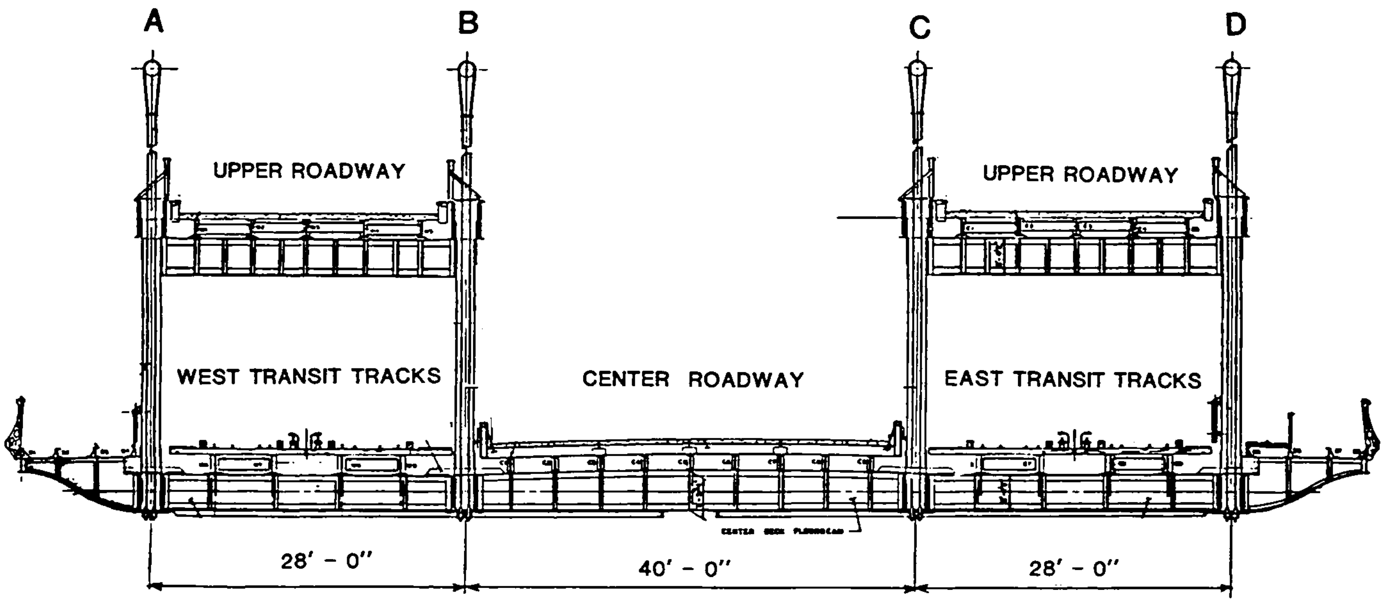

Fig. 1--Typical cross section of suspended span

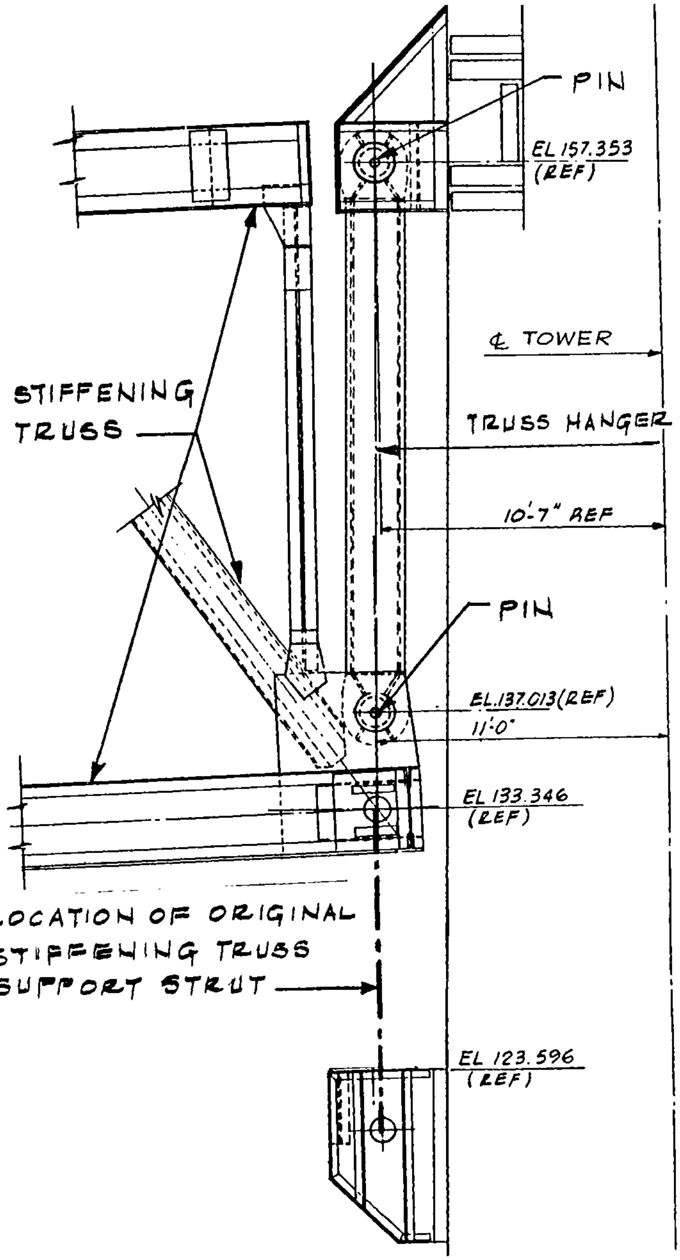

Fig. 2--Existing truss hanger

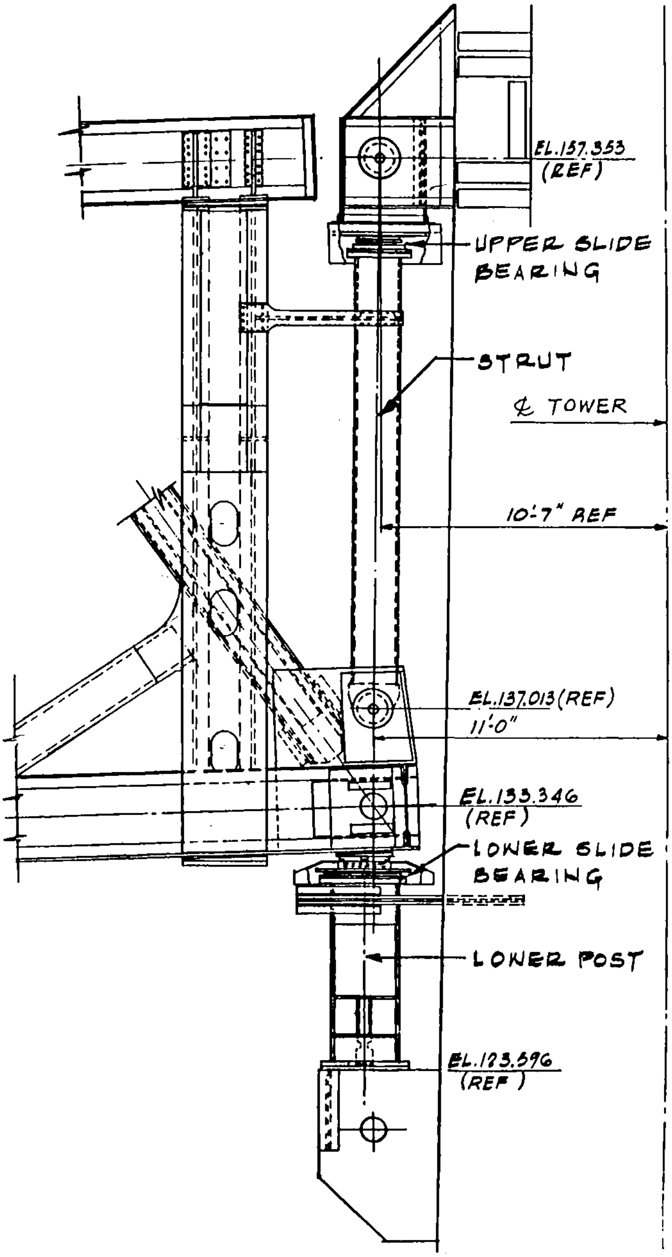

Fig. 3--Proposed new truss hanger

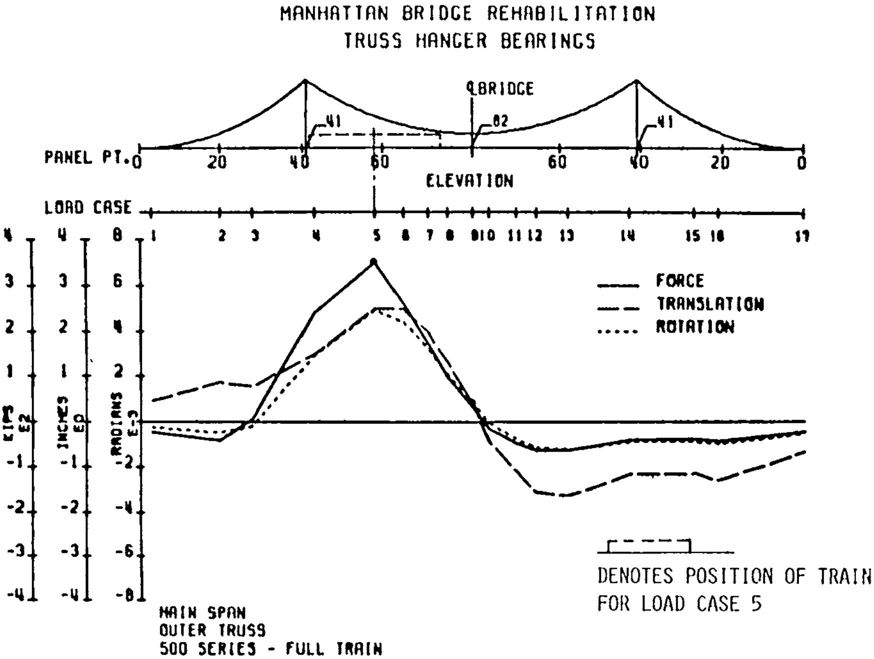

Fig. 4--Stiffening truss reactions and displacements due to one train on inner track