Dynamic tests on mill cranes, 1941

I. E. Madsen
DYNAMIC TESTS ON MILL CRANES
by I. E. Madsen*

1. SYNOPSIS

This report describes the results of impact and lateral load tests on three Electric Overhead Traveling Cranes at the Bethlehem Plant of the Bethlehem Steel Corporation and on the E.O.T. Crane in the Fritz Engineering Laboratory of Lehigh University. In these tests, the stresses in the girders due to acceleration, braking, and impact, were measured with scratch gages. It was found that the maximum lateral stress was due to braking and was about ten per cent of the vertical live and dead load for the cranes tested. The measured impact values were quite variable. Jerk impact was found to vary from 9 to 33 per cent of the live load. Secondly, when the crane was run over wedges to simulate bad runway joints, the impact was found to vary from 56 to 100 per cent of the live load.

2. INTRODUCTION

These tests were made in order to determine whether specification requirements for dynamic stresses in mill cranes were in accordance with those actually present in the girders of such cranes.

The dynamic stresses in cranes are of two types. One type of dynamic stress is the lateral stress due to acceleration and braking of the crane. This stress may be very large when the crane is of long span and the girders are relatively narrow. The braking stresses are usually larger than the acceleration stresses. The ratio of the braking load to the vertical load on the braked wheels may approach the value of the coefficient of friction between the crane wheel and runway rail. This maximum value may be modified by the timing of the swing of the load on the crane. If the instant of maximum swing should occur with the maximum braking force, the ratio of the maximum lateral force to the vertical force may exceed the coefficient of friction. However, in the tests made, the swing of the pendulum lagged behind the braking force. Lateral forces may also be due to other causes than those above enumerated. One such case would be the use of cranes to spot railroad cars. However, even in such cases, the maximum ratio of lateral load to vertical load should not exceed the coefficient of friction, since at this point the crane would slip on the runway rail.

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Impact stresses may be due to many causes. In mill practice running the crane over bad runway joints is probably the most important cause for such stress. Impact stress will also occur when a load is jerked off the ground.

Four cranes were tested in this program. One series of tests was made on the crane located in the Fritz Engineering Laboratory. This, a 10-ton Niles Crane No. 10097, has a span of 47 ft., and is a riveted fish-belly box girder. The bridge brakes are foot-operated mechanical brakes. The braking tests were made with no load on the crane due to limited clearance in the laboratory. A number of impact tests were also made on this crane. The jerk impact tests were made in the following manner. Slack was let out of the hoist. Then the slack was taken up so that the hook was moving at the maximum hoisting speed when the load was jerked off the floor. A tank of water weighing 8250 lb. was used for the load. To simulate bad joint impact, the crane was run over 3/4-in. oak wedges placed under both ends of one bridge girder. Under the weight of the crane, the wedges flattened down to 9/16 in. and this was assumed to be the drop of the crane. The 8250 lb. load was also used in the wedge tests and the crane was tested with 21-ft. and 14-ft. distances between the hook and hoisting drum.

Three cranes were tested at the Bethlehem plant of the Bethlehem Steel Corporation. The first crane, Bethlehem No. 430, was a new 10-ton riveted, fish-belly, box girder of 69 ft. span with hydraulic bridge brakes. Since the shop was under construction, the full length of the runway was available for test.

In the lateral tests, the load was picked up with the crane stationary. The crane was then fully accelerated until maximum speed was attained, and then the brakes were fully applied. These tests were made with a 31,000-lb. load of billets hung first as high as possible and then just high enough to clear the floor. Similar tests were made with a load of 15,500 lb. The clearance from the floor to the trolley rail of this crane was 23 ft. 9 in.

Jerk impact tests were also made on this crane. The load was raised with a jerk, and it was stopped with a jerk while being lowered.

The second crane tested at the Bethlehem plant was Bethlehem No. 410. This crane, located in the electrical shop, is a 5-ton hand-operated I-beam crane of 37 ft. 4 in. span. Lateral load tests were made on this crane using a magnet weighing 9100 lb. for the vertical load.
The third crane tested was Bethlehem No. 4. This is a 30-ton riveted, fish-belly box girder crane of 75 ft. span which is used as a skullcracker. This crane was tested to determine what impact is present as the ball is released from the magnet. A ball weighing 19,500 lb. was used in the tests and the magnet weighed 8800 lb.

The stresses in the cranes were measured by means of the De Forest Scratch Recording Strain Gage*. Since the stresses in the cranes were too small to use the gage as manufactured, a special holder was made to increase the gage length to 20 in. Pictures of the strains were taken with a microscope using a magnification of 450X. Fig. 1 shows such a picture. The gages were all calibrated against a 20-in. Whittemore strain gage. This was done in the laboratory by measuring the strains on a beam under a static bending load. In the crane tests, the gages were mounted on both edges of the top and bottom flanges of the crane girder at the span center. The trolley was also located at the span center during the tests. Occasional difficulty was encountered with some of the gages sticking so that no scratch was recorded, but generally enough gages worked so that the lateral stresses could be determined. In addition each test was repeated to insure the availability of enough records to determine the dynamic stresses.

The acceleration and deceleration were determined from stop-watch readings measuring the time to one-tenth of a second. Distances were marked on the runway rail and the time at each mark was found from the stop-watch. From these measurements, the speed and acceleration can be easily determined. Fig. 2 gives a picture of the acceleration and deceleration of Bethlehem, No. 4. This method is not as accurate as it should be and in future tests, it is suggested that a moving picture camera be used to photograph the stop-watch and distance marks along the rail at the same time. The acceleration and deceleration, as measured, apply only to the bridge of the crane and do not necessarily apply to the load since the motion of the load is modified by the swing.

3. TEST RESULTS

About one hundred pictorial records of dynamic stresses were taken in this investigation. Typical records are illustrated in this report and a summary of the pertinent data is given in Tables I and II.

* Bulletin No. 153, Baldwin Southwark Division, Baldwin Locomotive Works
Table I summarizes the results of the lateral load tests on the several cranes. The columns of computed and measured static stress apply to the vertical live load. The static stresses in the tables are the arithmetical average of the tension and compression flanges. The acceleration, deceleration, and speed in the table are taken from curves similar to Fig. 2. The measured lateral stresses are the arithmetical average of the readings on all the flanges, and are due to the live load and the dead load. The column of computed lateral stress is found by assuming that ten per cent of the live load trolley and dead load acts laterally. The next to the last column (lateral load from measured braking) is obtained by dividing the observed deceleration by \( g \), the acceleration of gravity, and multiplying by one hundred to get the percentage.

With the exception of the I-beam crane, the observed deceleration is about ten per cent of gravity as shown in the last column of Table I. However, the measured stresses are somewhat less than those which correspond to the observed braking force except for the laboratory crane.

The acceleration and deceleration stresses for the I-beam crane were somewhat larger on the top flange than on the bottom flange. The average value is given in the table.

Table II gives the results of the impact tests on the various cranes. The static stress referred to in this table is that due to the live load. For the jerk impact test, this is the only force which causes an impact stress. However, when the crane is run over a bad joint or wedges as in the case of the laboratory crane, there is an impact due to the weight of the trolley and girders, in addition to the live load impact. For this reason, two columns are given for the per cent of impact, the first column is the per cent impact based on the live load, and the other is the per cent impact based on all the loads causing impact. In specifications most impact factors are based only on the live load.

The variation in the impact factor due to changing the length of the hoist is shown in tests 3 and 4 on the laboratory crane, where the impact decreased from 100 to 56 per cent of the live load with an increase in the length of the hoist.

The jerk impacts varied from 9 to 33 per cent.

Bethlehem No. 4 is a skullcracker crane, on which the impact was measured as the ball was released. The measured impact stress was 37 per cent of the weight of the ball and 25 per cent of the weight of the ball and magnet,
as found from the scratch records. The center deflection was also measured in this crane with a 0.001-in. Ames dial and the impact measured from the dial was twenty per cent of the ball and magnet. This value may be a little low due to inertia of the dial. The jerk impact (not given in the table) was twelve per cent of the ball and magnet as found from the deflection.

Fig. 1 shows the scratch record for one edge of the compression flange in one of the wedge impact tests. All distances in this picture are measured from the centerline of the scratch. The portion AD shows the whipping action of the hoist as the cable tightens. In BC the load is being transferred to the crane. Between C and D the jerk impact can be seen. The distance between the bottom line which is the line of zero stress and the top line DE represents the static stress. At E the crane dropped off the wedges, and the reduction in stress, while the crane is dropping is shown in this portion. Before this stress has been completely released, however, the crane hits the runway rail and point G gives the sum of the static stress plus maximum impact stress. The portion of the curve after H shows the vibrations in the crane after impact.

Some other scratch curves are given for illustration in Fig. 3, 4, 5, and 6.

In Fig. 3 is shown the static stress and vibration which occur when the load is picked up quickly. The variation of the vibrations about their neutral position is a measure of the jerk impact.

In Fig. 4, are shown the acceleration and deceleration stresses for Bethlehem No. 430 in one of the lateral load tests.

Fig. 5 is a low-power magnification of the scratch record of Bethlehem No. 430. The left portion shows the stress as the load is lifted. The center shows the acceleration and braking stresses, while the right end shows the release of stress when the load is taken off the crane.

Fig. 6 shows the lateral stresses of Fig. 5 under a higher magnification.

4. DISCUSSION OF RESULTS

The column for maximum force of deceleration of Table I shows that a value of ten per cent of the vertical load gives a fair approximation of the lateral load. However, the measured stresses for the Bethlehem Cranes are somewhat less than the computed strains. One reason for this is that the computations neglected the walkway and end fixity. Since there was a walkway and end ties on these
cranes, these factors probably reduced the lateral stresses. This would be particularly true for Bethlehem No. 430 which was a new crane with a tight end connection. The ratio between the observed lateral stresses and those computed for the bridge girder as a simple beam, based on the observed maximum braking force, vary from 53 to 73 per cent.

The laboratory crane which is old and has practically no end connection would be expected to show no end fixity and in this crane the measured and computed stresses were in fair agreement.

A value of ten per cent for the lateral load would correspond to a coefficient of friction of 0.20 if one-half the bridge wheels were braked. If all the wheels were braked the lateral force would be twenty per cent. The value of 0.20 for the coefficient of friction is one which would not be at all unlikely under actual operating conditions.

The lateral end fixity probably can not be taken into account in design, since the end connections on a crane would loosen in service, and reduce the fixity.

On the other hand, some fixity will probably be present at all times, no matter how much the end connection loosens. Any end fixity present however, will provide an additional safety factor if the crane is designed as a simple beam for the lateral load.

The fact that the maximum swing of the load and the maximum acceleration may not have occurred at the same time may also have reduced the observed stress. This can be seen in Fig. 2 where it will be noted that there are two peaks in the deceleration curve. These peaks can also be seen in the acceleration and braking stresses in Fig. 4. This effect is due to the fact that when the tangent of the angle of swing is greater than the coefficient of friction, the load is being decelerated more than the bridge, and when this angle is less than the coefficient of friction the load is being decelerated less than the bridge. In actual practice, however, a good crane operator will operate his brake in such a manner as to keep the swing down to a minimum.

The impact stresses caused by running the crane off wedges under both ends may represent an unduly harsh condition, if this test is to simulate bad runway joints; because bad joints under both ends of the girder are unlikely to occur at the same time. Further tests, with wedges under only one end of the bridge should be made. In the tests made, this type of impact was the most severe, and it is quite likely that it is the type of impact which should govern in design. The wedge impact tests showed also that this impact is much more severe when the load is close to the
hoist drum. Since this is the position in which the load is usually placed as the crane moves down the runway, it should govern in design. Impact due to bad joints is also particularly severe since the weight of the trolley and girders also contribute to the impact stresses, which is not the case in other types of impact. Whenever, the length of hoist is long, the impact will be reduced because of the cushioning effect of the cables.

The jerk impact was a smaller factor than that due to dropping the crane off wedges. If this, then is always the smaller factor, the design impact factor should not be a function of the hoist speed of the crane.

One interesting effect was noted in the impact test on the skullcracker. After the switch was cut to release the ball, it took an average of 1.2 seconds before the ball left the magnet. This was apparently due to residual magnetism in the magnet even though the switch was of such a type that a reverse current was applied.

5. CONCLUSIONS

The dynamic tests discussed in this report indicate that for the cranes tested:

1. The maximum lateral force on a crane was due to braking and was approximately ten per cent.

2. Some end fixity was present which reduced the stresses due to the lateral force. However, this fixity probably can not be counted on in design and the full value of ten per cent for the lateral load should be used in design.

3. Jerk impact, as measured, varied from 9 to 33 per cent.

4. The impact due to running the crane off wedges to simulate bad joints was quite large. For the tests made, values of 56 and 100 per cent of the live load were measured. More tests on this factor are needed.

5. The closer that the load is to the hoist drum, the greater will be the impact.

6. ACKNOWLEDGMENTS

These tests are a part of a program sponsored and financed by the Association of Iron and Steel Engineers at the Fritz Engineering Laboratory of Lehigh University. The work is being carried out in direct cooperation with the Crane Specifications Committee of the A.I. & S.E. under the chairmanship of Frank W. Cramer. Special acknowledgment is due to Brent Wiley, Managing Director of the Association. The general program is under the direction of Bruce Johnston, Associate Director of Fritz Laboratory. Many thanks are also due D.M. Petty, L.V. Black, and C. Eichenberg for their help and cooperation in making the tests at the Bethlehem Plant of the Bethlehem Steel Corporation.
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Fig. 1 - Scratch Record of Impact Stresses in Laboratory Crane
SPEED-VELOCITY CURVES

BETHLEHEM CRANE #410

Fig. 2
Fig. 3 - Scratch Record - Jerk Impact - Bethlehem No. 430, 31,000 lb. load

Fig. 4 - Scratch Record - Acceleration and Braking Stresses - Bethlehem No. 430 31,000 lb. Vertical Load
Fig. 5 - Scratch Record - Low Magnification
Lateral Stresses - Bethlehem No. 430
31,000 lb. Vertical Load

Fig. 6 - Scratch Record - High Magnification
Lateral Stresses - Bethlehem No. 430
31,000 lb. Vertical Load