MEASURING DEAD LOAD STRESS OF EYEBARS IN STEEL RAILROAD BRIDGES

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ABSTRACT

Many steel truss railroad bridges use tension members consisting of eyebar groups. After years of service, these eyebars may exhibit significant wear at the pin connections, leading to the development of unequal tension stresses among eyebars within the same group. To restore bridge performance, eyebar tensions may be equalized through flame-shortening. The tension in any eyebar during this shortening process is commonly estimated by measuring the fundamental flexural natural frequency of the eyebar about its minor axis, and then converting this frequency to the corresponding tension by using a chart provided in the AREMA Manual for Railway Engineering. This chart is based on a simple analytical model that assumes the ends of an eyebar to be pinned; however, for transverse vibration about the minor axis, a significant degree of rotational constraint will often exist, potentially resulting in very inaccurate tension estimates.

This paper presents the results of research to improve the prediction of eyebar tension from the observed transverse natural frequency. This includes consideration of analytical and laboratory studies, where eyebar behavior for various end conditions is considered. Alternate charts for fixed-fixed and pinned-fixed conditions are presented, and using measurements from actual railroad trusses, the effectiveness of these charts to predict dead load tension stresses in eyebars is examined. Besides supporting the flame-shortening process, the potential to achieve more reliable tension measurements using the tools presented also provides for possible applications in bridge rating and performance assessments.
INTRODUCTION

Pin-connected trusses are often used in older steel railroad bridges, where tension members may consist of some number of pairs of eyebars. For optimal performance, the tension stresses should be equal among the entire group of eyebars that comprise a particular member. After years of service, it is common for sufficient wear to occur at the pin connections to result in significantly differing tensions among the eyebars for a given member. In extreme cases of wear and when no live load is present on the bridge, some eyebars may become so loose that they rattle when shaken, indicating that virtually no portion of the dead load is being carried. To restore bridge performance, stresses among eyebars might be equalized through flame shortening by following the procedure given in AREMA (2). As part of this procedure, a method is provided for estimating dead load tension stress in any eyebar by converting the observed transverse fundamental flexural natural frequency about the minor axis of the eyebar to the corresponding stress. This conversion is based on an exact mathematical relation between tension stress and the fundamental natural frequency as developed by Timoshenko (7) and assumes that the ends of the eyebar are ideal (frictionless) pin connections. For steel eyebars, AREMA presents this relation in the following form (2):

\[ F = 0.00293 N^2L^2 - 24,700,000 \left( \frac{T}{L} \right)^2 \]  

(1)

where \( F = \) axial tension stress of eyebar, psi

\( N = \) number of complete oscillations per second (i.e. natural frequency)

\( L = \) length of eyebar, in.

\( T = \) thickness of eyebar, in.

Based on this relation, AREMA also gives a chart solution for certain combinations of eyebar length and thickness.

The flame-shortening procedure given by AREMA is based on a report prepared by AREA Committee 15 (3). When introducing Eq. (1), this report recognized that since the mode of vibration being observed is about the minor axis (perpendicular to the axis of the pin), a much higher degree of end fixity could potentially exist than that represented by the modeled pin-ended condition. Unfortunately, similar exact mathematical solutions do not exist for other end conditions, and the limited computational tools at the time evidently precluded a general numerical analysis. As an alternative, “Seebeck’s Second Approximation” (based on the vibration characteristics of a tensioned wire) was provided to approximate
the behavior of fixed-ended eyebars. However, it was also acknowledged that this approximation does not provide satisfactory results for low stress or low slenderness ratio \( (L/T) \) situations.

The length to be used in Eq. (1) is referred to by Reference (3) as the “effective” length, but no guidance is given with regard to what this length should be. Since the distance between pin centers is readily available through construction drawings, it is believed that this length is what has generally been used in the application of Eq. (1). In this paper, other effective lengths to be considered among the various analytical models will include the clear distance between overlapping head edges, and the distance from the pin center at one end to the overlapping head edge at the other end. Figure 1 illustrates these lengths and the notation that will be used to identify them throughout.

This paper will consider analytical models, and corresponding chart solutions obtained through numerical methods, that enable the vibration-based determination of stresses in eyebars of various end conditions. The results of laboratory tests will then be presented that were conducted to validate the analytical models and to observe the general physical response of eyebars with various end conditions. Eyebar behavior in actual bridges will then be examined through two case studies. Finally, application of the analytical tools presented in this paper will be discussed.

![Figure 1. Effective Lengths for Eyebars](image)

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ANALYTICAL MODELS

The various analytical models considered are shown in Figure 2. Each assumes the eyebar to be prismatic, homogeneous, and linear-elastic, and does not include the effects of shear and rotary inertia. Case (a) illustrates the pin-ended model of the current AREMA procedure. Using the general equation of motion as developed by Timoshenko (7) and applying the appropriate boundary conditions, numerical solutions were completed for cases (b) through (d). The details of the solution process and its development for the fixed-ended case are given in Reference (5). The outcome of cases (b) through (d) will now be examined.

Figure 2. Eyebar Analytical Models
The Fixed-Ended Model

From the solution of the fixed-ended model, the chart shown in Figure 3 was developed that relates tension stress to the frequency of vibration for the fundamental flexural mode about the minor axis of the eyebar. In this figure, the same eyebar lengths and thicknesses are used as those included in the chart for the pin-ended case provided by the current AREMA procedure (2). While it is not possible to obtain an exact equation for the fixed-ended case, there is a consistent stress relation between this and the pin-ended case for a given slenderness ratio. This relation, illustrated by Figure 4, demonstrates that the differences in predicted stresses between the two models can be significant, especially for eyebars of relatively low slenderness. Also shown is the “equal stress limit”, or the line where the predicted fixed-ended stress is the same as the pin-ended stress; this limit is approached by eyebars having particularly large slenderness ratios. Figure 4 is also a useful tool; through this chart, stresses first determined using Eq. (1) for the pin-ended model can then be easily converted to stresses based on the fixed-ended model.

The Pin-Fixed-Ended Model

Developed in the same manner as the previous case, Figure 5 provides the tension-frequency relations for an eyebar with one end pinned and the other end fixed, and Figure 6 shows the stress relations between this pin-fixed case and the pin-ended case. The same general trends observed in the previous case are also evident here; more specifically, Figure 6 shows that the differences between the pin-fixed and pin-ended cases become more pronounced as eyebar slenderness ratios decrease. As expected, however, these differences are considerably less than that observed for the fixed-ended case.

The Variable-Rotational-End-Stiffness Model

In order to study eyebars having intermediate degrees of end constraint, a pin-ended model with rotational springs of variable stiffness was developed in the same manner as the previous cases. It is difficult to generalize this model with simple charts as was done for these other cases. However, it does serve as a valuable research tool that enables the fine-tuning of analysis to very closely match actual in-situ eyebar behavior, providing a means to better assess the effectiveness of the simpler models in predicting eyebar stresses. Results from this model will be used to support the case studies examined later in this paper.
Figure 3. Eyebar Stress Based on the Fixed-Ended Model

Figure 4. Eyebar Stress Relations for Both Ends Fixed vs. Both Ends Pinned
Figure 5. Eyebar Stress Based on the Pin-Fixed-Ended Model

Figure 6. Eyebar Stress Relations for One End Pinned and One End Fixed vs. Both Ends Pinned

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LABORATORY TESTING

Experiments with scaled-down eyebars were conducted in a laboratory to enable validation of the fixed-ended analytical model as well as to observe the physical response of eyebars with significant pin-hole wear and little lateral restraint of the heads. Sized to fit within the universal testing machine used, each eyebar was fabricated from A36 steel, had a cross section of 1 in. × 0.1875 in., a length between pin centers of 43 in., head radii of 1.125 in., and pin holes bored to accommodate 0.875-in. pins. Further details of the experimental methods and initial tests are presented in Reference (6). To allow comparisons of the pin-fixed-ended analytical model, additional tests were subsequently conducted that focused on eyebar response with one head highly restrained and the other head unrestrained. At the restrained end, overlapping eyebar heads were emulated by using steel washers of 1-in. outer radius, slightly (0.125 in.) smaller than the head radii of the eyebar. The head and washers were tightly clamped between support blocks at this end. This restraint condition will be referred to as a “packed” joint. At the other end, the eyebar head had no lateral restraint of any kind. To reflect heavy wear, the pin hole was also oversized approximately 3% relative to the pin diameter. For comparison, AREMA (1) limits pin clearances to about 0.4% to 0.6%, depending upon pin diameter. This condition will be referred to as an “unpacked” joint.

Validation of the Fixed-Ended Model

Figure 7 shows eyebar stress as a function of the fundamental flexural natural frequency, and provides experimental results along with several analytical model predictions. For a fixed-ended eyebar, it was reasoned that the fixed condition should begin close to the overlapping head edge. If true, the appropriate effective eyebar length should be $L_{HE}$, or the clear distance between overlapping heads (i.e., washers), making the slenderness ratio $L/T = 219$. This hypothesis is supported by Figure 7, where excellent correlation is apparent between the results for an eyebar with both joints packed and the fixed-ended analytical model when using $L_{HE}$.

Evaluation of the Pin-Fixed-Ended Model

For tests of an eyebar with one joint packed and the other unpacked and worn, the results in Figure 7 compare very well with the pin-fixed analytical model when using $L_{HE}$, particularly at low stress levels. Achieving this consistency through the use of $L_{HE}$ is an interesting and unanticipated result, and implies that an equivalent hinge condition develops close to where the head edge would be. At higher stresses, the
Experimental results deviate increasingly (in a conservative way) from the predicted pin-fixed stress, and begin to drift toward the fixed-ended analytical model. Evidently, this reflects a rise in rotational resistance at the unpacked joint as the contact stresses between head and pin increase with elevated load.

**Comparison to the Current AREMA Model**

As expected from the analytical model comparisons discussed earlier, Figure 7 demonstrates that the conventional pin-ended model using $L_{PC}$, or the distance between pin centers, deviates considerably from the actual response of both eyebars.

![Graph showing comparison of experimental eyebars stresses with analytical models](image)

**Figure 7. Comparison of Experimental Eyebars Stresses with Analytical Models**

**CASE STUDIES**

Eyebar behavior as observed on two operational railroad bridges will now be considered. For each eyebar tested, stations were marked at the eighth points relative to length $L_{PC}$ and the dynamic response was measured by attaching accelerometers at these stations. Auto and cross-spectral records of 0.0625-cycle/sec. resolution were collected and processed using a spectrum analyzer. Vibrations were induced by either a single impact from a rubber mallet or by manually shaking and releasing the eyebar. Natural frequencies were obtained by peak-picking, modal amplitudes determined for a particular natural frequency by scaling auto-spectra to a reference station, and cross-spectra used to confirm phase relations.
CSX Canton Bridge

The first study involved Bridge 1C of the former Baltimore & Ohio Railroad’s Philadelphia Branch in Canton, MD, now owned by CSX Transportation. Built in 1885, the bridge utilizes a skewed, pin-connected, 12-panel Whipple through truss. Designed to accommodate two tracks, the bridge now carries a single track that has been shifted to the centerline of the bridge. Tests were conducted on the south truss and included lower chord member L9-L10, consisting of four eyebars of 6-in. by 1-7/16-in. cross section and lengths $L_{PC} = 228$ in. and $L_{HE} = 213.5$ in. (Further details of the testing and initial results are reported in Reference (5)). The pin connections appeared to be very tight and the faces of the overlapping eyebar heads in firm contact with each other, and the joints were well-restrained by floorbeams against translation and out-of-plane rotation. With an $L/T$ of approximately 150 to 160 (depending upon the effective length used), these eyebars are in the lower range of those considered in Figure 4, where a greater difference in predicted stress between the pinned and fixed-ended models would be expected.

Fundamental flexural mode shape results for the third eyebar (counting from the deck side of member L9-L10) are given in Figure 8, with the horizontal axis set up in terms of $L_{PC}$ as shown. When using $L_{PC}$, it is apparent that the fixed-ended analytical model is much closer to the measured response than the pin-ended model. This figure also shows that using $L_{HE}$ in the fixed-ended model provides even better agreement with the measured response. The behavior of the other three eyebars (not shown) was also very consistent with that shown here for eyebar 3.

![Figure 8. Mode 1 for Member L9-L10, Eyebar 3, Referenced to $L_{PC}$](image)
Dead load stresses as predicted by several analytical models are presented in Table 1, illustrating the wide range that can occur among the models and effective lengths. As noted earlier, a 0.0625-cycle/sec. frequency resolution was used and actual frequencies determined by peak-picking; due to this aspect, the error in the predicted stresses shown is less than 2%. In addition, the table shows the stress calculated using the design dead load as given in the original construction drawings (and does assume two tracks). For the best-fitting model from Figure 8 (fixed ends, using $L_{HE}$), the average stress (4.60 ksi) is also the closest of the models to the design dead load stress (4.38 ksi). The table further shows the predicted stress obtained using the variable-rotational-end stiffness model (4.93 ksi), which was fine-tuned to achieve an even better fit with the measured response. The fixed-ended model using $L_{HE}$ compares well with this model, but does deviate in a slightly non-conservative way, indicating that a very small degree of rotation occurs at the head edge. Nonetheless, these results show that the fixed-ended model with $L_{HE}$ provides good stress estimates when joints are tight and well-restrained, such as observed for this bridge.

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**Amtrak Windsor Locks Bridge**

The second study examined a bridge of the former New York, New Haven & Hartford Railroad, now owned by Amtrak, which spans the Connecticut River at Windsor Locks, CT. The specific span considered is a circa 1904 structure that utilizes a skewed, pin-connected, 7-panel subdivided Pratt through truss. Tests were conducted on the north truss and focused on diagonal member M5-L6, consisting of four eyebars of 8-in. by 1-1/4-in. cross section and lengths $L_{PC} = 255-3/16$ in., $L_{HE} = 236-7/16$ in., and $L_{PC,HE} = 245-13/16$ in. (Further details of the testing and initial results are reported in Reference (6)). The first eyebar (relative to
the deck) was obviously loose and under very little tension; when vibrated, its head was visibly rocking about the pin at mid-panel point M5. The other three eyebars appeared to have higher degrees of tension and rotational resistance at this joint. At joint L6, all four eyebars displayed firm contact between overlapping heads, with the joint being tight and well-restrained by a floorbeam. With an $L/T$ of approximately 190 to 200 (depending upon the effective length used), these eyebars are in the low to middle range of those considered in Figures 4 and 6, where moderate differences in predicted stress between the pinned and other analytical models would be expected.

The results for the first three eyebars of member M5-L6 are shown in Figures 9, 10, and 11, where the horizontal axis is set up in terms of $L_{PC-HE}$. (Not included is eyebar 4, whose behavior was very similar to eyebar 3.) Because of access limitations, measurements could only be acquired from joint M5 to the midpoint of each eyebar. Treating joint L6 as fixed, the variable-rotational-end-stiffness model (labeled as “Partial-Fix” in these figures) was used to adjust the analysis to closely match the measured response by changing the spring stiffness at joint M5. Among the effective lengths considered, this fitting was best achieved using $L_{PC-HE}$. In Figure 9, this model shows that very little rotational resistance exists at joint M5 (i.e., the left end of the chart) and the predicted tension is zero, which is consistent with behavior observed in the field. For eyebar 2 (Figure 10), a small amount of tension occurs and the rotational resistance is a little higher, while both parameters increase dramatically for eyebar 3 (Figure 11).

Figures 9 through 11 also show the pin-fixed-ended model predictions using the effective lengths $L_{PC-HE}$ and $L_{HE}$. Between these two lengths, the predicted stress results obtained using $L_{HE}$ matches more closely with the variable-rotational-end-stiffness (or “partial-fix”) model. This agreement is particularly good for the first two eyebars, suggesting that an equivalent pin support exists very near the head edge when little rotational resistance exists; note that this result is very consistent with the laboratory studies presented earlier for an eyebar with one end unrestrained. Naturally, the mode shape for the pin-fixed model using $L_{HE}$ increasingly deviates at the left end from the best-fit partial-fix mode shape, since their modeled supports are at different positions. However, the better consistency in stresses seems to reflect that the pin support of this pin-fixed model is closer to the inflection point of the partial-fix model. As expected, when rotational resistance becomes more substantial, the accuracy of both pin-fixed models decreases (Figure 11), but does so in a conservative manner.
Figure 9. Mode 1 for Member M5-L6, Eyebar 1, Referenced to $L_{PC-HE}$

Figure 10. Mode 1 for Member M5-L6, Eyebar 2, Referenced to $L_{PC-HE}$

Figure 11. Mode 1 for Member M5-L6, Eyebar 3, Referenced to $L_{PC-HE}$
Upon application of the current AREMA pin-ended model (using $L_{PC}$), the following stress predictions were obtained for eyebars 1 through 3, respectively: 1.42 ksi, 3.16 ksi, and 7.78 ksi. Comparing these with the results of Figures 9 through 11, the pin-fixed model with $L_{HE}$ provides a much better estimation of actual stress, while appearing to remain conservative for all eyebars considered here.

**APPLICATION OF ANALYTICAL MODELS**

For the flame-shortening procedure given by AREMA, the primary purpose of the pin-ended analytical model is to provide a means to easily establish when eyebar tensions are approximately equal, and for this purpose it is adequate. This paper has shown, however, that this model is not effective in determining actual tension stresses, especially for eyebars of small $L/T$ ratios. It has also been demonstrated that for eyebars with tight and well-restrained joints (such as commonly occurs with lower chord members) the fixed-end model using $L_{HE}$ does provide good accuracy. It has further been shown that for eyebars with one end loose and the other well-restrained (as may be prone to occur with diagonal members) the pin-fixed model using $L_{HE}$ provides a much better estimation of actual stress than the pin-ended model. This ability to more confidently predict eyebar tensions, especially in fixed-ended situations, may have a number of applications in support of rating and bridge performance assessments.

To implement the models presented here, the frequency of an eyebar can be easily measured using a magnetically mounted accelerometer wired to a data acquisition system. While a spectrum analyzer would provide better results, acceptable measurements might be obtained using a lesser expensive portable oscilloscope. The instrumentation must be able to accommodate the relatively low frequencies (possibly as low as 1 cycle/sec.) characteristic of eyebars. Vibrations can be induced by manually shaking the eyebar and acquiring a measurement upon release. Using Eq. (1), the corresponding pin-ended stress can be determined from the measured fundamental natural frequency. Using the charts of Figure 4 or 6, this pin-ended stress can be converted to either the fixed-ended or pin-fixed-ended case as appropriate. Of course, the same effective length should be used in the equation that is also to be used in the chart.

The methods presented here are not well suited for measuring stresses associated with moving loads, as it is difficult to isolate the fluctuating eyebar frequencies. However, these methods would be appropriate to assess the distribution of train loads that are stationary on a structure.
To obtain meaningful results, actual eyebar characteristics must not deviate significantly from the assumptions behind the models. Therefore, eyebars must be of uniform width and thickness, have no significant bends or distortions, and contain no tensioning devices. Each must be completely free to vibrate, having no interferences along the entire body of the eyebar; thus, any clamps or clips that are sometimes used to tie together groups of eyebars would have to be removed.

**Example**

The following example is provided to further demonstrate the application of the methods proposed here. It is based on an American Bridge Company 200-ft through truss as given by Ketchum (4).

**Data:**
- Member L3-L4: 4 bars, 8 in. × 1-15/16 in. × 28 ft 6-27/32 in. \((L_{PC})\)
- Design dead load: 205 kips (or 3.31 ksi)
- Pin diameter: 6-1/2 in.
- Head diameter: 18 in.

\(N\) (obtained using design dead load, fixed-ended model, and \(L_{HE}\)): 5.26 cycles/sec.

**Analysis:** Assume the behavior reflects fixed-ended conditions.

\[L_{HE} = L_{PC} - 2(\text{head radius}) = 342.84 \text{ in.} - 2(9 \text{ in.}) = 324.84 \text{ in.}\]

From Eq. (1), the dead load stress based on pinned ends is

\[F = 0.00293 N^2 L_{HE}^2 - 24,700,000 \left(\frac{T}{L_{HE}}\right)^2\]

\[F = 0.00293 (5.26)^2 (324.84)^2 - 24,700,000 (1.9375/324.84)^2\]

\[F = 7680 \text{ psi, or 7.68 ksi}\]

Convert this stress to the fixed-ended case using Figure 4 and \(L_{HE}/T = 168\):

\[F = 3.3 \text{ ksi}\]

(Note that this result checks with the given design dead load stress and assumed \(N\).)

**CONCLUSIONS**

The limitations associated with AREMA’s conventional vibration-based method for estimating dead load stress in eyebars have been examined. Based on a pin-ended analytical model, this method can result in significantly overestimated stresses when fixed-end conditions exist, particularly for eyebars of relatively low slenderness. An alternative fixed-ended analytical model has been presented; laboratory and field tests
have shown this model to give good estimates of eyebar stress when both of the joints are tight and well-restrained by relatively rigid structural elements, such as a floorbeam. A pin-fixed analytical model was also considered for situations where one end of an eyebar exhibits looseness. While it too is more effective than the conventional method, it remains approximate in nature, with the accuracy of the estimated stress dependent upon the actual degree of rotational restraint, tension level, and effective length used. In addition to supporting AREMA’s procedure for flame-shortening of eyebars, the new analytical models presented here may have useful applications for rating and bridge performance studies.

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REFERENCES


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Figure 2. Eyebar Analytical Models
Figure 3. Eyebar Stress Based on the Fixed-Ended Model
Figure 4. Eyebar Stress Relations for Both Ends Fixed vs. Both Ends Pinned
Figure 5. Eyebar Stress Based on the Pin-Fixed-Ended Model
Figure 6. Eyebar Stress Relations for One End Pinned and One End Fixed vs. Both Ends Pinned
Figure 7. Comparison of Experimental Eyebar Stresses with Analytical Models
Figure 8. Mode 1 for Member L9-L10, Eyebar 3, Referenced to $L_{PC}$
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Measuring Dead Load Stress of Eyebars in Steel Railroad Bridges

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Introduction

Eyebars in a steel railroad bridge
Introduction

• Due to joint wear, unequal tensions may develop among the eyebars of a truss member.

• Various methods might be used to adjust tensions

• For flame-shortening approach, AREMA provides a simple procedure for estimating dead load stress from the observed natural frequency
AREMA Method for Estimating Eyebars Tension

• Basis is a closed-form relation between axial tension stress ($F$) and observed natural frequency ($N$) in uniform bars of length $L$ and thickness $T$:

$$F = 0.00293 N^2 L^2 - 24,700,000 \left(\frac{T}{L}\right)^2$$

• Relation assumes pinned ends.
• For minor-axis vibration, actual degree of end restraint in many cases is anticipated to be more consistent with fixed-ends.
• A similar closed-form solution for fixed ends does not exist.
Figure 15-8-4
What is Eyebbar Effective Length?

$L_{PC}$, or length between "pin centers" 

$L_{HE}$, or length between "head edges"

$L_{PC\cdot HE}$, or length between "pin center" and "head edge"
Eyebar Analytical Models and 1st Modes

- **Pin-Ended Model**
- **Fixed-Ended Model**
- **Pin-Fixed-Ended Model**
- **Variable-Rotational-End-Stiffness Model**
Eyebar Stress based on Fixed-Ended Model

\[ F = \text{Stress in Eyebar, ksi} \]

\[ N = \text{Frequency of Vibration, cycles/sec.} \]

- \( L = 50 \text{ ft} \)
- \( L = 30 \text{ ft} \)
- \( L = 40 \text{ ft} \)
- \( L = 20 \text{ ft} \)

- \( T = 1 \text{ in.} \)
- \( T = 1.5 \text{ in.} \)
- \( T = 2 \text{ in.} \)
Fixed vs. Pin-Ended Stress Relations
Eyebar Stress based on Pin-Fixed Model

\[ F = \text{Stress in Eyebar, ksi} \]

\[ N = \text{Frequency of Vibration, cycles/sec.} \]

- \( L = 50 \text{ ft} \)
- \( L = 30 \text{ ft} \)
- \( L = 40 \text{ ft} \)
- \( L = 20 \text{ ft} \)

- \( T = 1 \text{ in.} \)
- \( T = 1.5 \text{ in.} \)
- \( T = 2 \text{ in.} \)
Pin-Fixed vs. Pin-Ended Stress Relations
Laboratory Testing

- Eyebars sized to fit universal testing machine
- Fabricated from 3/16-in.-thick ASTM A36 steel
- $L_{PC}/T \approx 230$ (well within AREMA range of 120 – 600)

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$L_{PC}/T \approx 230$ (well within AREMA range of 120 – 600)
Laboratory Testing

Eyebars:
Laboratory Testing

“Loose” pin; hole oversized about 3%:
Laboratory Testing

Test apparatus:
Laboratory Testing

“Unpacked” joint:
Laboratory Testing

“Packed” joint:
Laboratory Testing

Comparison of experimental and analytical results:

Graph showing the comparison of stresses in eyebar against frequency, cycles/sec., with data points for Tight Pins, Packed Joints, Loose Pins, One Joint Unpacked, Fixed-Ended Model, L(HE), Pin-Fixed-Ended Model, L(HE), and Pin-Ended Model, L(PC).
Case Study 1 – CSX Canton Bridge

• Canton, MD bridge of former Baltimore & Ohio Railroad, now owned and operated by CSX Transportation.
• Skewed 12-panel Whipple through truss, built in 1885.
• Designed for two tracks, now carries a single track centered longitudinally on bridge.
• Focus of testing on south truss, member L9-L10, consisting of 4 eyebars ($L/T \approx 150$).
• It appeared both joints were well-restrained, with firm contact between overlapping eyebar heads.
CSX Canton Bridge
CSX Canton Bridge
CSX Canton Bridge

Results for Member L9-L10, Eyebar 3, Mode 1:

- **Modal Amplitude**

  - **Pin-Ended Model**
  - **Fixed-Ended Model**
  - **Fixed-Ended Model, using L(HE)**
  - **Measured, N = 9.125 Hz**
### Dead load stress results for Member L9-L10:

<table>
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<tr>
<th>Eyebars</th>
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<td>11.97</td>
<td>6.55</td>
</tr>
</tbody>
</table>

Total: 4.38
Case Study 2 – Amtrak Windsor Locks Bridge

- Windsor Locks, CT bridge of former New York, New Haven, and Hartford Railroad, now owned and operated by Amtrak.
- Skewed 7-panel sub-divided Pratt truss, circa 1904.
- Focus of testing on north truss, member M5-L6, consisting of 4 eyebars ($L/T \approx 200$).
- Apparent looseness of Eyebar 1 was observed at Joint M5.
- It appeared that Joint L6 was well-restrained, with firm contact between overlapping eyebars.

$L/T$ represents the ratio of the length ($L$) to the height ($T$) of the truss member.
Amtrak Windsor Locks Bridge
Amtrak Windsor Locks Bridge
Amtrak Windsor Locks Bridge
Amtrak Windsor Locks Bridge
Amtrak Windsor Locks Bridge
Amtrak Windsor Locks Bridge

Results for Member M5-L6, Eyebar 1, Mode 1:

- Pin-ended model, using $L_{PC}$: stress = 1.42 ksi
Amtrak Windsor Locks Bridge

Results for Member M5-L6, Eyebar 2, Mode 1:

Pin-ended model, using $L_{PC}$: stress = 3.16 ksi
Amtrak Windsor Locks Bridge

Results for Member M5-L6, Eyebars 3, Mode 1:

Pin-ended model, using $L_{PC}$: stress = 7.78 ksi
Application of Analytical Models

• Conventional pin-ended model does indicate when eyebar tensions are equal, but can be very inaccurate regarding specific tension stresses.

• New models permit more reliable tension estimates, especially when joints are well restrained. This may provide for applications beyond flame-shortening.

• Frequencies can be easily measured with various instrumentation options.

• Besides dead load, method can be used to measure stresses associated with stationary live loads.

• In applying models, eyebar characteristics must be reasonably consistent with modeling assumptions.
Application of Analytical Models

Example: American Bridge Co. 200-ft through truss as given by Ketchum, Member L3-L4

Data: 4 bars, 8” 1-15/16” 28’ 6-27/32” \( (L_{PC}) \)
Design dead load = 205 kips (or 3.31 ksi)
Pin diameter = 6-1/2”
Head diameter = 18”
\( N = 5.26 \) cycles/sec. (using design dead load, fixed-ended model, and \( L_{HE} \))
Analysis: Assume fixed-ended conditions

\[ L_{HE} = L_{PC} - 2(\text{head radius}) \]
\[ = 342.84" - 2(9") \]
\[ = 324.84" \]

\[ L_{HE}/T = \frac{324.84}{1.9375} = 167.66 \]

\[ F_{pin} = 0.00293 \times N^2 \times L_{HE}^2 - 24,700,000 \times \left( \frac{T}{L_{HE}} \right)^2 \]
\[ = 0.00293 \times (5.26)^2 \times (324.84)^2 \]
\[ - 24,700,000 \times \left( \frac{1}{167.66} \right)^2 \]
\[ = 7680 \text{ psi, or 7.68 ksi} \]
Application of Analytical Models

![Graph showing stress reflecting fixed-ended conditions vs. stress reflecting pin-ended conditions with a line for equal stress limit and a notation L/T = 168]
Conclusions

• The conventional pin-ended model can significantly over-estimate eyebar stress, depending upon $L/T$.

• For tightly-packed joints where firm contact between the faces of eyebar heads is observed and the pin itself is well-restrained, eyebar behavior is very consistent with the fixed-ended model.

• When one joint is loose, the pin-fixed model provides much better estimations of stress than the pin-ended model.

• Besides supporting eyebar shortening procedures, the ability to easily and confidently estimate stresses may also benefit bridge rating and inspection efforts.
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