A COMPARISON OF RAILROAD AXLE STRESS RESULTS USING DIFFERENT DESIGN SIZES, LOADING CRITERIA AND ANALYSIS METHODS

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ABSTRACT

Although railroad axles do not often fail in North American freight service, they remain a critical component. This paper presents the results of stress analysis calculations that were performed for various different North American freight railroad axle designs. Loading criteria appropriate for the various axle designs were used, and different methods of axle stress analysis are compared and contrasted. Results are reviewed and discussed. Also, the authors propose that a standard axle stress analysis method be adopted by the North American freight railway industry for new axle designs, particularly for those axles to be used at increased gross rail loads (GRL).

INTRODUCTION

Standardized Association of American Railroads (AAR) railroad axle designs have certainly stood the test of time and continue to perform extremely well under demanding conditions. In today's severe North American freight railroad service environment axle failures are very rare, and axle failures between the wheelseats (not related to bearing issues) are almost non-existent. Figure 1 shows selected Federal Railroad Administration axle failure data for the last decade (FRA, 2001). Non-journal related freight car axle failures, coded as "Broken/bent axle between wheel seats," are displayed. Total axle failures, including bearing related failures, numbered between 59 and 68 for each year.

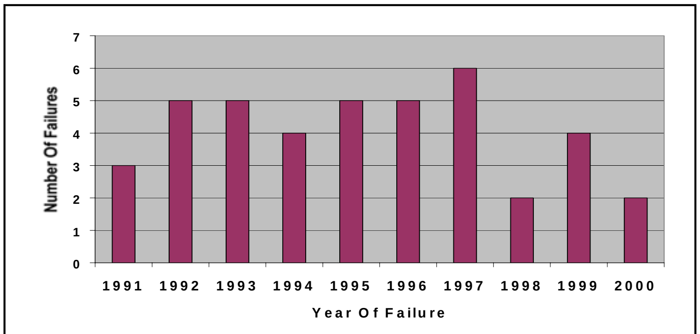

Figure 1. Selected FRA axle failure data, 1991-2000.

Axle fractures between the wheelseats, although not a common occurrence, normally are the result of a surface initiated fatigue crack that begins at surface damage on the axle, such as a dent, arc strike, or abrasion mark from rubbing brake rigging. Bent axles can occur as a result of derailment.

North American axles continue to perform admirably even though axle loads have increased, wheel/rail impact loads have increased and freight car utilization (thus the number of fatigue cycles experienced by the axles) has improved tremendously. Consider that a freight car could easily travel 50,000 loaded miles in a year. This means that the axles have experienced more than 28,000,000 loaded rotating bending fatigue cycles in one year.

Axle failures in certain other parts of the world are more common than in North America. Rose (1999) stated that axle breakage rates were considerably lower in countries outside the United Kingdom. The UK Rickerscote freight axle failure in March of 1996, which resulted in the derailment of a passenger train and a fatality, led to increased UK scrutiny for axles. A review of practices found that in the UK, specific axles are designed for a given application (Rose, 1999). In North America, car designers select service-proven, standardized AAR axle designs for use.

ENDURANCE LIMIT AND PAST FATIGUE TESTS

Traditionally, fatigue life is described using an "S-N curve" with stress plotted on the Y-axis against the number of cycles to failure on the X-axis. For steels, the S-N curve becomes horizontal at a limiting stress, known as the endurance limit. At stress levels below the endurance limit, the material can withstand an "infinite" number of stress cycles. Given the large number of fatigue cycles experienced annually by a railroad axle, the many years an axle can remain in service, and the need for very high reliability, infinite life is desired.

Byrne (1967) reviewed results for AAR full-scale axle fatigue tests of 184 axles (six designs) that began in 1938. The fatigue endurance limit for the body of raised wheelseat axles was found to be 17,500 psi. For the wheelseat area the endurance limit was found to be 14,000 psi. Additional fatigue testing to improve freight car axles involving 103 axles began in 1949. The testing established upper limits for design bending stress at four locations on the axle surface, as shown in Table 1 (Byrne, 1967).

Location On Axle Surface	Upper Limit Stress, psi
Central portion of axle body	17,500
Body at wheelseat-body 3” radius	16,800
Wheelseat at inner wheel hub face	12,500
Along journal	10,000

Table 1. Bending stress design limits from AAR tests.

Byrne (1967) noted that the axle testing (with results shown in Table 1) involved 100% loaded cycles, which is not completely realistic given the fact that a freight car can be empty about half the time. Also, the testing used a more severe loading condition than normally expected in service. However, until better data are available, a conservative approach towards axles is considered desirable.

INCREASED GROSS RAIL LOADS AND AXLES

In recent years North American railroads adopted 286,000 pounds gross rail load (GRL) as the maximum allowable load for 100-ton service, and a large number of freight cars currently operate at this level. A smaller percentage of cars now operate at 315,000 pounds GRL (125-ton service). The allowable 100-ton load was previously 263,000 pounds GRL. Although wheel service problems have increased in recent years, and arguably much of this increase can be attributed to higher GRLs, axle problems have not been an issue.

Recent discussions within the railroad industry to increase the allowable GRL further to 315,000 pounds (125-ton service) led the authors to write this paper. New design 7 x 9 axles (and older design 7 x 12 axles now used in 125-ton service) will carry the load at 315,000 pounds GRL. If the railroad industry continues to increase the allowable GRL, axles could, at some point, reach stress levels that are unacceptable and potentially dangerous. Further, the fatigue-life effects of dynamic, high strain-rate loads caused by wheel/rail impacts from defects such as spalling, slid flats, etc., are not addressed in this paper and are likely to become more significant at higher axle loads.

TOWARDS A NORTH AMERICAN AXLE DESIGN ACCEPTANCE STANDARD

Currently the AAR requires that an S-660 finite element analysis computer simulation be conducted to determine the stress levels present in new wheel designs (AAR, 1998). The stresses are then compared with stresses for those wheel designs successfully performing in railroad service. Based upon this comparison of stresses, new wheel designs are accepted or rejected by the AAR.

However, there is no formalized method by which new AAR axle designs are accepted or rejected. This is largely due to the absence of axle problems and the fact that new designs are not often brought forth. Further, when new bearing designs (such as the new bearing for the 7 x 9 axle) are developed, some amount of design attention is paid to the axle by bearing manufacturers.

British, European (UIC) and other standards exist for establishment of acceptable axle design stresses. Also available is the Reuleaux method, which was adopted in 1896 by the Master Car Builder’s Association, and the modified Reuleaux method, first used in 1939 (Byrne, 1967). AAR adoption of an accepted axle design standard would have benefits for future axle designers.

AAR AXLE DESIGNS AND LOADS USED

Table 2 shows the standardized AAR axle designs and gross rail loads used for the stress analyses in this paper.

AAR Axle Designs	Gross Rail Load, lb.
6-1/2 x 12 – Class F	286,000
6-1/2 x 9 – Class K	286,000
7 x 12 – Class G	315,000
7 x 9 – Class M	315,000

Table 2. Axle designs and loads used for stress analysis.

THE VARIOUS AXLE ANALYSIS METHODS

Four different axle specifications were used to determine axle stresses for this paper. The axle specifications are: 1) 1939 modified Reuleaux (Byrne, 1967), 2) UIC 513-3 (International Union of Railways, 1994), 3) CEN prEN 13103 (European Committee for Standardization, 2000), and 4) BASS 504 (British Railways Board, 1985). The loads were applied to a free body diagram representing one wheelset of a multi-wheelset railroad car and the resultant stresses were compared. The free body diagrams used in developing the relationships were constructed using classical mechanics practices. A spreadsheet was then used for calculations.

Many of the equations for calculating axle stresses have their origin in the work done by Prof. F. Reuleaux in 1893 (Byrne, 1967). In 1896, the Master Car Builders’ Association, which is the predecessor of the Mechanical Division of the Association of American Railroads, adopted a variation of Reuleaux’s method. The method specified symmetric loading of the wheelset.

In 1939, the modified Reuleaux method, which calculates the axle stresses of a wheelset at the point where one wheel is completely unloaded and has just left the rail head, was introduced (Byrne, 1967). The free body diagram associated with this method is significantly different from that used for the other three. For the modified Reuleaux method, the wheelset is modeled as a beam that is cantilevered on each side of one wheel. All forces and reactions considered are in one plane. Forces due to braking are not considered.

In the BASS 504 procedure, unequal forces are applied to the axle bearings. All forces and reactions are not contained in one plane as in the modified Reuleaux method. Forces due to normal braking are also included.

The UIC 513-3 method also applies unequal forces to the axle bearings. All forces and reactions are not contained in one plane. As in BASS 504 method, bending moments due to normal braking are considered. In addition, however, frictional forces are also incorporated in the calculations.

The CEN prEN 13103 routine is virtually identical with the UIC 513-3 specification, except that the normal braking force is increased slightly thus resulting in the highest stresses.

RESULTS AND DISCUSSION

Stress values were calculated at five different locations on the surface of each of the four axle designs. Locations are designated as L1, L2, L3, L4 and L5, as shown in Figure 2. Tables 3 through 6 show results of stress calculations for 6-1/2 x 12, 6-1/2 x 9, 7 x 12 and 7 x 9 axles, respectively. Bold values in the table indicate the maximum value at each axle location.

Analysis Method	Stress for 6-1/2 x 12 Axle - 286 K GRL (psi)				
	L1	**L2**	**L3**	**L4**	**L5**
Mod. Rel.	12190	9193	12948	17110	13461
UIC 513-3	12397	9370	13416	18118	18265
CEN	**12412**	**9381**	**13422**	**18127**	**18278**
BASS 504	11822	8916	13283	17847	18040

Table 3. Stress analysis results, 6-1/2 x 12 axle, 286K GRL.

Analysis Method	Stress for 6-1/2 x 9 Axle - 286 K GRL (psi)				
	L1	**L2**	**L3**	**L4**	**L5**
Mod. Rel.	8477	9228	12949	17112	13463
UIC 513-3	8622	9405	13417	18120	18267
CEN	**8632**	**9417**	**13424**	**18129**	**18280**
BASS 504	8222	8950	13284	17848	18042

Table 4. Stress analysis results, 6-1/2 x 9 axle, 286K GRL.

Analysis Method	Stress for 7 x 12 Axle - 315 K GRL (psi)				
	L1	**L2**	**L3**	**L4**	**L5**
Mod. Rel.	8628	8342	11146	14673	11624
UIC 513-3	8729	8583	11549	15559	15771
CEN	**8740**	**8594**	**11554**	**15566**	**15783**
BASS 504	8367	8091	11434	15305	15577

Table 5. Stress analysis results, 7 x 12 axle, 315K GRL.

Analysis Method	Stress for 7 x 9 Axle - 315 K GRL (psi)				
	L1	**L2**	**L3**	**L4**	**L5**
Mod. Rel.	7835	10170	11146	14673	11624
UIC 513-3	7920	10578	11549	15559	15771
CEN	**7929**	**10591**	**11554**	**15566**	**15783**
BASS 504	7599	9863	11434	15305	15577

Table 6. Stress analysis results, 7 x 9 axle, 315K GRL.

We note that the European CEN prEN 13103 specification yields the highest stress results of the four analysis methods used. As stated previously, this is because the analysis method includes the highest levels of braking forces.

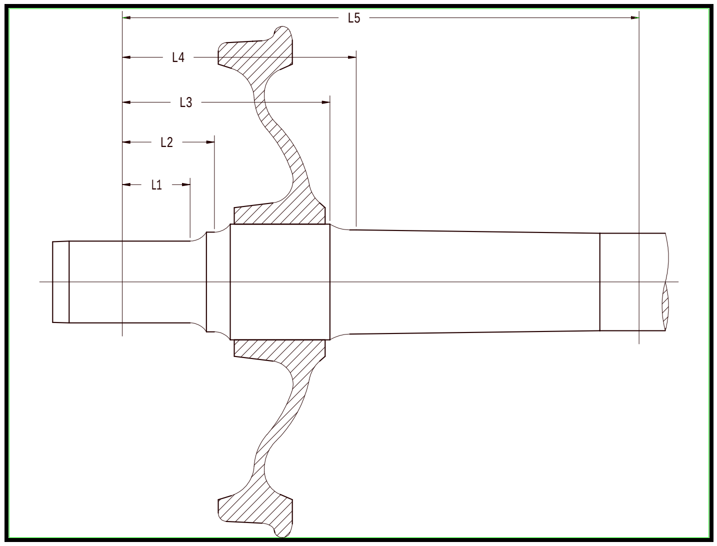

Figure 2. Stress Locations L1 through L5

As the data in Tables 3 through 6 show, axle stresses in the journal area (L1) are lower for the 6-1/2 x 9 and 7 x 9 axles than the 6-1/2 x 12 and 7 x 12 axles, respectively. Bearing manufacturers intended this result when they designed the new Class K and Class M components. Lower stresses in the journal area, due to a shorter moment arm, will help to reduce bearing related problems.

The difference in calculated journal (L1), wheelseat (L3) and body (L4 and L5) stresses seen between axles used in 286K GRL service and 315K GRL service is striking. Although the larger section size of 7 x 12 and 7 x 9 axles results in additional component weight, the reduction in stresses is significant over the stresses seen in 6-1/2 x 12 and 6-1/2 x 9 axles. However, note that the L2 stresses are higher for the 7 x 9 axle than the other axle designs. This is due to the dimensions of the 7 x 9 journal diameter and wheelseat shoulder diameter areas.

It may be possible to decrease axle weight, but any such efforts should be done with great care. We know that service stresses in 6-1/2 x 12 and 7 x 12 axles currently are below the endurance limit, and are therefore satisfactory, due to the large number of such axles in railroad service for many years and with few service failures.

What should be the acceptable maximum stress levels for axles in North American freight service? This question can be answered simply - At levels where service fatigue failures do not occur. Since we have an absence of failures now, we are within the zone of "safe" stresses. How much greater can axle stresses be before service fatigue failures begin to occur? Unfortunately, the answer to this question is not so simple. The effects of increased gross rail loads, and in particular wheel/rail impact loads from shelled/spalled, slid flat, built-up-tread and out-of-round wheels, have not been addressed by this paper. Such impact loads will likely increase in frequency and severity at 315 K GRL.

CONCLUDING REMARKS

Although axle failures are a rarity in North America and existing axle designs are clearly succeeding in service, we recommend that the AAR adopt a formal methodology for acceptance of any new axle designs. At some point in the future, with axle loads likely to continue to increase, axle stresses could reach levels that are greater than an axle's endurance limit. Since the European CEN prEN 13103 method of stress calculation yields the highest stresses, and thus provides the most conservative approach, we recommend that AAR adopt this method to qualify axle designs. Adoption of an axle design qualification method brings axles "in line" with wheels since AAR uses the S-660 simulation to accept or reject new wheel designs. Computer spreadsheets allow for rapid calculation of axle stress values and thus adoption of a new analysis method poses no hardship for those involved with axle design.

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