HMS

HEAVY MOVABLE
STRUCTURES, INC.

FIFTH BIENNIAL SYMPOSIUM

November 2nd - 4th, 1994

Holiday Inn Surfside
Clearwater Beach, Florida

SESSION WORKSHOP PRESENTATIONS

"OBSERVATIONS AND COMMENTS: AASHTO DESIGN PRACTICES FOR MOVABLE BRIDGE OPERATING MACHINERY"

by ROBERT L. CRAGG, P.E.
Steward Machine Company

OBSERVATIONS AND COMMENTS:
AASHTO DESIGN PRACTICES FOR MOVABLE BRIDGE OPERATING MACHINERY

presented by:

Robert L. Cragg, P.E.

Heavy Movable Structures, Inc.
Fifth Biennial Symposium
November 2-4, 1994
Holiday Inn Surfside
Clearwater Beach, Florida

The source document for this investigation is the American Association of State Highway Transportation Officials publication "Standard Specifications for Movable Highway Bridges" issued in 1988.

Article 2.5.4 states two limiting conditions for electric motor driven machinery to move the span:

1. Design at the normal allowable unit stresses for 150% of full rated electric motor torque (FLT) for the time specified at Condition A, Article 2.5.3 speed.

2. It shall also be designed for unit stresses caused by a torque that is 50% greater than the starting torque, or breakdown torque, whichever is greater.

From these statements at least two things are apparent. First, the designer must have a complete understanding of Article 2.5.3, Power Requirements and Machinery Design. Secondly, a question is posed about the difference between "normal allowable unit stresses" and "unit stresses". For the purpose of this discussion the "unit stresses" in Item 2, above, is interpreted to mean the stresses resulting from the stated condition shall not exceed the allowable unit stresses mandated in Article 2.5.11.

While the operating characteristics vary depending upon the type motor, generally the breakdown torque will be approximately 250% FLT and greater than than 150% of the starting torque.

This provision then imposes a design value for the machinery at 250% FLT which is a significant increase over the previous 150% FLT requirement. The result is dramatically larger shafts, gears, bearings, couplings and other machinery elements.

To further comprehend the influence of the breakdown torque restriction an understanding of Article 2.5.3 is necessary. Basically, selection of the prime mover is done by investigating the power required under three conditions:

A. Normal time for opening against frictional resistances, inertia, unbalanced conditions and a wind load of 2.5 lb./sq. ft.

B. Opening time in 1.5 times the normal opening time with an ice loading of 2.5 lb./sq. ft. in addition to the loads in A.

C. Opening in twice the normal time against frictional resistances, inertia, unbalanced conditions, 10 lb./sq. ft. wind load and 2.5 lb./sq. ft. ice loading. (Ice loading may be neglected in locations where temperatures are rarely below freezing).

AASHTO requires the maximum bridge starting torque for bascule bridges be determined under Condition C using friction coefficients for starting and neglecting inertia. Let's see what this means by looking at two bascule leafs for highway usage.

	Leaf I	Leaf II
Distance, Trunnion CL to Tip of Leaf	79 ft.	124.5 ft.
Distance, Center to Center of Girders	25.5 ft.	26.75 ft.
Rack radius	7.5 ft.	11 ft.
Area of Floor	1580 sq. ft.	4539 sq. ft.
Time for Opening	60 sec.	80 sec.

Except for size the mechanical operating machinery systems are similar on both bridges and include a drive motor, brakes, enclosed differential speed reducer, transverse shafts, rack pinions and racks and the necessary couplings and bearings, as illustrated in Figure 1.

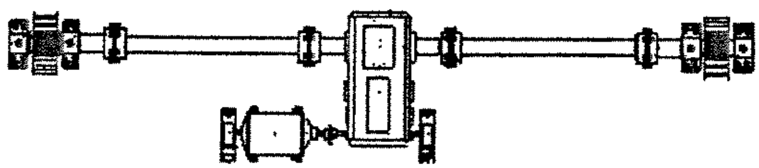

Figure 1. Mechanical Operating Machinery

Following AASHTO procedures the power requirements are determined using maximum leaf starting torques as required in Article 2.5.3:

	Horsepower Required		
	Cond. A	Cond. B	Cond. C
Leaf I	17.4	28	39.4
Leaf II	38.4	38.4	94.6

Motor horsepower selection in accord with Article 2.10.14 is: 40 HP for Leaf 1 and 100 HP for Leaf II. It should be noted the HP ratio of Condition C to Condition A is 2.26 for Leaf I and 2.46 for Leaf II. Also recognize that rarely, if ever, is the leaf operated under loadings of Condition C, and then only as an emergency measure.

Now, using the provisions of Article 2.5.4 the machinery is to be designed for stresses resulting from 150% FLT and also breakdown torque (approximately 250% FLT). By introducing these two factors it is seen the machinery is then designed for stresses about 3.3 times those experienced during Normal Opening Time (Condition A) at 150% FLT and over 5.6 times the Normal Opening Time stresses at Breakdown Torque.

Using Breakdown Torque stress levels has a substantial influence on the sizes of shafts, bearings, gears, couplings and other mechanical components in the operating machinery system.

Let's look at some of them:

1. Shafts

Article 2.5.15 gives the formula for proportioning shafts: $d^3 = 16 \cdot K \cdot \frac{\sqrt{M^2 + T^2}}{\pi \cdot s}$

where:
d = shaft diameter, in. s = allowable unit extreme fiber stress, tension or compression, lb/sq. in.
M = simple bending moment, in.- lb. T = simple torsional moment, in.- lb.

and the impact factor: $K = 1.0 + 0.03 \cdot \sqrt{RPM}$

It is noted from this formula that for a given condition of FLT and allowable stress the shaft diameter is a function of the FLT multiplier, either 1.5 or 2.5 as given in Article 2.5.4 :

$$d = f\left(1.5^{0.333}\right) \quad \text{or} \quad d = f\left(2.5^{0.333}\right)$$

It is also apparent, since weight varies as the square of the diameter it will increase dramatically. The calculated increase in weight is about 40% to handle the stresses resulting from 250% FLT.

Looking at the floating, transverse shafts coupled to the output shafts of the primary speed reducers of the bridge leafs described earlier we determine the following:

Leaf I. T =492478 lb. in. RPM = 2.4 s = 7500 psi K = 1.047 M = 0 Length = 16 ft.(2 @ 8 ft.)

d @ 150% FLT = 8.0 in. Weight = 1351 lb. d @ 250% FLT = 9.5 in. Weight =1892 lb.

Leaf II. T =1909000 lb. in. RPM = 1.6 s = 7500 psi K = 1.15 M = 0 Length = 14 ft. (2 @ 7 ft.)

d @ 150% FLT =12.5 in. Weight = 2886 lb. d @ 250% FLT =14.9 in. Weight = 4181 lb.

2. Gears

Open spur and bevel gears are described in Article 2.6.12, Strength of Gear Teeth. The formula for calculating the allowable load on gear teeth is:

$$W = f \cdot s \cdot p \cdot \left(0.154 - \frac{0.912}{n}\right) \cdot \left(\frac{600}{600 + V}\right) \quad \text{for Full depth teeth}$$

$$W = f \cdot s \cdot p \cdot \left(0.178 - \frac{1.033}{n}\right) \cdot \left(\frac{600}{600 + V}\right) \quad \text{for Stub teeth}$$

where:

W = allowable tooth load, lb. p = circular pitch s = allowable unit stress, psi
f = effective face width, in. n = number of teeth V = pitch line velocity, ft./min.

Now, since the torque is the product of the tooth load and pitch radius and the pitch radius is equal to the number of teeth times the circular pitch divided by two pi it is seen that for given conditions of torque, speed, circular pitch, allowable unit stress and number of teeth the face width is directly proportional to the torque :

$$f = K_T \cdot \text{Torque}$$

Therefore: f @ 250% FLT = 1.67 x (f @ 150% FLT)

Of course, to abide by AASHTO provisions a wider pinion and gear is necessary to accommodate the 250% FLT requirement. This is illustrated by considering the racks and pinions on the bascule leafs described earlier.

Leaf I:

$n := 17 \quad p := 3.5 \quad V := 11.96 \quad s := 22500 \quad W_{150\%} := 83012 \quad W_{250\%} := 138354 \cdot lb$

$$f_{150\%} := \frac{W_{150\%}}{s \cdot p \cdot \left(0.178 - \frac{1.033}{n}\right) \cdot \left(\frac{600}{600 + V}\right)} \qquad f_{250\%} := 1.67 \cdot f_{150\%}$$

$f_{150\%} = 9.2 \text{ in} \qquad f_{250\%} = 15.3 \text{ in}$

Leaf II.

$n := 17 \quad p := 5.5 \quad V := 12.2 \quad s := 22500 \quad W_{150\%} := 192426 \quad W_{250\%} := 321351$

Following the same procedure as for Leaf I the face widths are calculated to be:

$f_{150\%} = 17\,in \qquad f_{250\%} = 28.4\,in$

The faces of both the gear and pinion will have to be increased 67% to meet the allowable stress levels resulting from the 250% FLT condition. Of course the weight as well as the cost of the of the material will increase 67%.

Another influence on proportions of the gears is noted in AASHTO Art. 2.6.12 which limits the face width of spur gears to 3 times the circular pitch. It is evident in the case of these two bridges the 250% FLT condition would necessitate redesign to conform to the AASHTO specification. This is an extensive change affecting both the rack pinion and rack and, no doubt, influencing the structural arrangement.

3. Speed Reducers

Art. 2.5.22 outlines the load requirements for speed reducer design. Basically they are designed in accordance with the requirements of AGMA and rated with an AGMA Service Factor of 1 based upon the power demands of Art. 2.5.4. It is also stated the peak, or breakdown torque, of the reducer shall not be less than the peak, or breakdown torque of the motor.

The provision concerning breakdown torque leaves the designer in a dilemma; it is incomplete since AGMA does not define a "breakdown torque" rating for speed reducers. It is not clear from AASHTO if this is a static or dynamic condition or if the basis for the rating is surface durability or bending strength.

The AASHTO procedures outlined in Art. 2.6.12 are modifications of the basic Lewis formula which considers the tooth as a cantilever beam and evaluates the bending strength at the critical point in the root of the tooth. No attention is given to surface durability, or pitting, which frequently is the critical rating, particularly for high speed pinions.

AGMA, on the other hand, offers means to rate gear sets on the basis of Pitting Resistance as well as Bending Strength. The AGMA Standard applicable to speed reducers is AGMA 6010-E88 (a revision of AGMA 420.04), "American National Standard for Spur, Helical, Herringbone and Bevel Enclosed Drives". While this standard is appropriate for all elements of the speed reducer; gears, shafts, bearings, fasteners, etc. the fundamental gear rating methods are those developed in other AGMA Standards:

ANSI/AGMA 2001-B88 "Standard for Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Teeth"

ANSI/AGMA 2003-A86 "Rating the Pitting Resistance and Bending Strength of Generated Straight Bevel, Zerol Bevel and Spiral Bevel Teeth"

In ANSI/AGMA 2001-B88 the relationship of pitting resistance and bending strength is discussed in Sec. 4.1.

"There are two major differences between pitting resistance and bending strength ratings. Pitting is a function of the Hertzian contact (compressive) stresses between two cylinders and is proportional to the square root of the applied load. Bending strength is measured in terms of the bending (tensile) stress in a cantilever plate and is directly proportional to the same load. The difference in the nature of the stresses induced in the tooth surface areas and at the tooth root is reflected in a corresponding difference in allowable limits of contact and bending stress numbers for identical materials and load intensities."

Accordingly:

Pitting Resistance stress: $Sac = Kc \cdot \sqrt{Pac}$

Bending Strength stress: $Sat = Kt \cdot Pac$

where:

Kc and Kt are constants which include modification factors and gear characteristics required in the fundamental rating formulas.

Pac is the applied load for which the gears are being rated.

Both pitting and bending strength are fatigue phenomena, one related to the contract surface stresses and the other to the resistance to cracking at the tooth root fillet in external gears. As such they are dynamic conditions.

In view of this AGMA recognizes that speed reducers may be subjected to momentary overloads, stall conditions and low cycle fatigue, so the unit must be evaluated to assure the allowable stress of any component is not exceeded. For such investigations the maximum allowable stress is determined by the yield properties rather than the bending fatigue strength of the material. This stress is designated Say (ANSI/AGMA 6010-E88 Sec 4.4) and, similar to the Bending Stress, is proportional to the applied load.

Additionally the AGMA standard acknowledges that the allowable stress numbers for bending and yield are dependent upon the type of heat treatment and minimum hardness of the material. In the case of allowable stress in bending AGMA goes even further and permits a range of values dependent upon the quality of steel and type of heat treatment. For the purposes of this discussion the lowest (most conservative) values of Sat will be used.

AGMA relates the allowable stress numbers to the material Brinell hardness according to the following relationships for through hardened gears and pinions: (Note: Surface hardened gears have different values)

$$Sat = 167 \cdot BHN - 0.152 \cdot BHN^2 - 274 \qquad Say = 482 \cdot BHN - 32800$$

where:

BHN = Brinell Hardness
Say and Sat = Allowable Stress Number, lb./sq. in..

of the machinery and associated structural components with little, if any, tangible benefit of improved integrity, safety or reliability of the system. Sometimes it appears the guiding philosophy is "If a little is good a lot is better". However, imposing inordinate design demands upon the machinery is an innocuous practice that feeds upon itself in escalating project costs. In addition to increased prices of the operating machinery components the larger size equipment requires heavier foundations and attachments to the pier and structures as well as more skilled labor hours, or dollars, to complete the installation. Succinctly stated, the owner is not getting much bang for the additional bucks. In the two cases illustrated it is estimated the increased cost of the installed machinery system will be about 40% greater to satisfy the motor breakdown torque condition.

Considering the historical reliability of mechanical operating machinery systems verifies that previous conservative design practices have adequately fulfilled the operating requirements. It is common for this equipment to perform its duty for 50 to 100 years without failure due to inadequate design.

While machinery elements have failed and will continue to fail in the future the causes are usually not related to the components being under designed on load carrying capacity. Present and past practices confirm the design capacity far exceeds the magnitude of applied loads, even under the most adverse emergency situation.

Certainly, while it is essential to continue following conservative practices, recognition should be made of the difference between momentary loads of high magnitude and those loads experienced during normal operation.

Accordingly, it is suggested:

1. That loads encountered at motor beakdown torque be considered a static condition and not a dynamic loading.

2. In designing for stresses resulting from motor breakdown torque that the allowable stresses be based upon the yield properties of the materials in a manner similar to the way in which it is accommodated in the AGMA Standard.

3. Base the allowable stresses of gears, shafts, keys, etc. upon the type of material, heat treatment and resultant hardness of that material.