DR 53-7844-1

Milw. H81

Mr. C.W. Warner, Manager

June 7, 1965

Subject: Brakes for Motors
Designed to New Standards

My letter of March 15, 1965 to Mr. H.L. Te Selle under the subject "AISE Standardization of A.C. Mill Motors and Brakes" pointed out that the problems of standardization of brakes for the A.C. mill motors of AISE Standard I-A cannot be segregated from the standardization of the proposed AISE #800 series of D.C. motors. Furthermore, the proposed new D.C. line of mill motors was at that time in the formative state, so a determination of brake torque and heat dissipating capacity requirements could not be made. However, recently you requested that I study the effects of the new 800 motors on our brakes. In the following paragraphs, I discuss brakes for those proposed 800 motors based on certain assumptions. Assumptions are necessary because the 800 motors are still not set up even on a tentative basis.

In the following paragraphs, I discuss brakes for the new line of 800 motors. Briefly, the study shows that:

A. It appears that based on torque and heat dissipating capacity, the Bul. 505 brakes now standard with the AISE #600 motors can be used with the same horsepower rated motors in the proposed 800 series. Some brakes will require a "well" for mounting because the height of the brake from the mounting surface to the centerline of the wheel is greater than the height of the motor from the mounting surface to the centerline of the shaft.

B. If the brake standards specify that brakes must be used without a "well", then our Bul. 505 brakes are not satisfactory and a new line will be required.

The following points are considered in the analysis.

1. Basic Assumptions:

   a. The ratio of brake torque to motor full load torque is to be approximately the same as now standard for the #600 mill motors as listed in NEMA Standard IC1-20.21.

   b. WK^2 of the motor not to increase more than 15% above the present WK^2 of the frame having the new horsepower rating. For example, the 50 H.P., 1 hour motor in the present standard series is frame 610. The G-E motor in that frame has a WK^2 of 72#'^{2} . The new motor will be the 808 frame which will be in the present 608 motor frame. The present G-E 608 motor WK^2 is 45#'^{2} . The 808 motor would then have a maximum WK^2 of 52#'^{2}.

   Obviously, the 15% increase in WK^2 is nothing more than a guess, but I believe it is reasonable. When the #600 motors replaced the #400 motors, the <u>increase</u> in WK^2 ranged from 40 to 120%. This great increase resulted from longer rotors which could be used because the 600 motors were built with roller bearings only while the 400 motors were built with sleeve or rotor bearings. I doubt whether any such increase in space is available when changing from the 600 to the 800 motors.

   c. The speed of the #800 motors has not been published and that is an important point. When changing from the old #400 motors to the #600 motors, the speed was retained with the horsepower rating. Thus full load motor torque for a specific horsepower rating remained the same. However, I understand that in the change to the #800 motors consideration will be given to retaining speed with the motor horsepower <u>or</u> retaining speed with the motor frame. The latter means an increase in motor speed for the horsepower rating and a reduction in full load motor torque.

d. There was no 600 motor for the 250 H.P., 1 hour rating but that rating will be for the new 818 motor frame. However, the old #421 motor did have this rating for that frame and the speed for the 618 frame was used in this study.

2. The attached tables SK052865-5-AEL, -6-AEL, and -7-AEL show pertinent data for the present #600 series motors and the proposed #800 series motors using our present standard Bul. 505 brakes. For the 800 motors, speeds are given for two conditions, that is, where the speed is retained with the horsepower, and where retained with the motor frame.

   a. Note that using the same brakes on the 800 motors as on the 600 motors for the same horsepower ratings, that WK^2 of the motor plus the brake wheel is less for the 800 motors than for the 600 motors. This results in a lower kinetic energy due to the motor plus the brake wheel whether speed is retained with the horsepower rating or with the motor frame.

      Because of this lower WK^2, there will be a greater downward drift of the load when stopping from very low speed hoisting because of the time required for the brake to apply. Furthermore, unless taken care of by the control, there will be an increase in overhauling speed because of this lower WK^2.

   b. When changing from the 400 to the 600 line of motors, one of the requirements was that the height from the centerline of the wheel to the brake mounting surface must be less than the height from the centerline of the motor to the mounting surface of the motor. In other words, no "well" would be required for mounting the brake. If this requirement is carried over to the brakes for the 800 motors, some serious problems are encountered.

      Columns 9, 10, and 11 of the tables show that our present Bul. 505 brakes are unsatisfactory when used with the 802, 804, 808, 810, 814, and 818 motors. This means that new designs for brakes are required for these motors. Actually, a new line of brakes to replace the Bul. 505 brakes will be needed. These new brakes will undoubtedly require smaller diameter wheels. In order to obtain the same wheel heating dissipating surface and lining areas, the wheel face must be increased. The wider the wheel face, the greater the accuracy of machining required in the brake itself.

Furthermore, as mentioned earlier in this letter, I feel quite certain that any new setup of D.C. brakes to be used with the proposed 800 D.C. motors will also have to take into account the new A.C. mill motors per AISE Standard I-A.

The preceding comments give some indications of the problems which will be encountered if and when the 800 motors are set up as standard. I believe that a Brake Committee will be set up after those 800 motors are standardized.

In your letter of February 8, 1965, you also mentioned brakes for the motors covered by the latest revision of NEMA Standards. In past discussions, I have pointed out that it appears impractical to set up brake standards for the NEMA Standard motors other than those for the NEMA - AISE mill motors. Brakes which we have furnished for those other NEMA Standard motors are used on applications where the brake torque requirements vary greatly for the same motor frame. For example, our records indicate that during the past five years the 8" Bul. 505 brake has been used with the following NEMA frames:

213, 215, 254, 256, 284, 286, 324, 326, 364, 365, 404, 405, 445, and 504 motors. The quantities involved for each frame was not only one, but in most cases, four or more.

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(10-24-67)

12. <u>Coil</u>

(a) Terminals are preferred. Protects coil internal leads. Will do away with complaints of coil lead failures caused by swinging, twisting and right angle bends of leads.

(b) If leads are used, do we have sufficient data to enable us to use smallest copper section for current thru short length. Leads on Bul. 505 brakes do not follow codes for long leads, yet we haven't had complaints on leads having proper junctions with coil.

(c) What type of insulation? Class B, F, or H? Not A. I recommend Class H. Epoxy fill is H, so balance of insulation could be H to permit higher operating temperatures. Kapton, Pyre ML, Teflon. "Kapton" tape .002" thick now cheaper than .003" thick #3 Quinterra according to NECCO (4-11-67) who now uses Kapton on Heavy Duty Copper wound magnets.

(d) We need to improve insulation to ground. On Bul. 505 brakes had failures to ground at core, side plates, and outer shell.

(e) If leads are used - provide solderless connectors.

(f) Terminals require conduit boxes - leads do not.

(g) Stationary coil is preferred.

(h) Coil epoxy encapsulated, but if possible of such construction that it can be repaired by customer - at least lead or terminal repair.

(i) Desirable that coil be easily replaced without disturbing adjustments, and without releasing brake.

(j) Brakes on blooming mill screwdowns may have to operate 20 to 40 times per minute. We recommend shunt brakes <u>with very low voltage coils</u> and <u>permanent series resistors</u>. For example 889-3 coil (1.96) with 10.5 ohms in series for 23" brake.

(k) If leads are used, clamp lead to coil case, so if failures do occur because of swinging, twisting or right angle bends, those failures will be in customers cable - not in coil leads.

AEL

13. Pull Rod

(a) Find more positive method of holding nut in position on pull rod.

(b) Provide clearance at pull rod pivot, between pin and hole, greater than normal to eliminate freezing due to fretting corrosion.

(c) Fabreeka shock absorber if necessary similar to that on 30" Bul. 505 brake.

(d) Pull rod above wheel as on previous C-H brakes - not below wheel as on some competitive brakes. The latter construction greatly increases the stress at pivots.

(e) A unitized pull rod and spring assembly which can be removed as a unit would be desirable.

AEL

14. Torque Spring

(a) Torque spring should be out of the magnetic circuit - now a major problem in locations with "Kish".

(b) If spring must be in center of magnet as on Bul. 505 brake, use a non-magnetic spring to reduce collection of "Kish" etc.?

(c) In order to replace coil without releasing or disturbing adjustments, the torque spring must be away from magnet.

AEL

15. Shoe Levers

(a) Outer shoe lever adjustable as on 30" H. M Brake to compensate for machining tolerances. Specific instructions to users? Clamp shoe on wheel to properly locate, and then bolt down? Dowel?

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16. Shoes

(a) Replace shoes and/or linings without special tools.

(b) If linings rivetted (groov-pins) must have correct size of holes. Had trouble on Bul. 505 brakes with loose groov-pins.

(c) If linings are held with groov-pins, they should be predrilled and shoe holes should be jig drilled to match.

AEL

17. <u>Linings</u>

(a) Bonded or rivetted

(b) If rivetted - Pre-drilled

(c) Clamp on linings - or mounting plate?

(d) Replace linings easily without special tools.

(e) Replace linings without disturbing brake adjustment.

(f) What type of lining? Woven and compressed? Molded? (EC & M molded 1/8 to 7/16" thick depending on size)

(g) Consider replacing J-M 600 lining with J-M 900 lining. Increase thickness.

(h) Thicker linings? Or will bonding of linings give enough improvement in life?

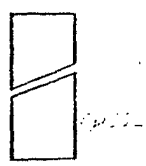
(i) 9-8-48 AISE - NEMA Brake Committee for #600 motor brake standard. Operating men agreed that lining ought to last 2 years. Vic Schlossberg of Inland said that life is required on all the jobs including ore bridges. I agree that 2 years life would be desirable, but there are too many severe service jobs where 2 to 4 months life would be an improvement.

(j) Lining pressures. NEMA - AISE 1945 Brake Committee discussed this point at some length. Some brake manufacturers wanted to go to 60 psi, but others were equally sure that 25 psi was high enough. The Cutler-Hammer standard was 30 psi. The following pressures were finally agreed upon for the proposed brakes for the #600 motors.

8" Brake	----------	26.4 psi
10" Brake	----------	29 psi
13" Brake	----------	30.3 psi
16" Brake	----------	30.6 psi
19" Brake	----------	33.2 psi
23" Brake	----------	35 psi

(k) Rivetted linings on Bul. 505 brakes. Had trouble with improper depth of counter bore for groov-pin head - Too shallow - Too deep.

(l) Consider scavenging strips in lining.



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(m) The Johns-Manville #600 folded and compressed lining has been standard on our brakes since 1930. It was selected after careful tests which showed that it had a minimum coefficient of friction of .30 and good lining life. Most of our competitors used molded linings and claimed that their lining was superior to that used on C-H brakes. However our tests on their molded linings indicated that the J-M #600 lining was superior. As a matter of fact, we learned that most brake manufacturers did not conduct lining tests but simply accepted the lining manufacturers data. Some of the brake manufacturers increased the thickness of the molded lining and told customers that their lining life was greater than the life of our lining by the ratio of thicknesses. Our tests disproved this point and indicated that the heavier lining was used in order to approach the life of thinner #600 linings used on our brakes. Some steel mills carried #600 linings in several thicknesses so that they could replace molded linings with the #600 material.

There have been improvements in molded linings, but I recommend that when going to the bonded linings we do not change from the folded and compressed lining to the molded type unless our tests show that the new molded lining is truly superior. The folded and compressed lining has been an important feature of C-H brakes for many years. A change to a molded lining shouldn't be made lightly.

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18. Wheels

(a) Ductile Iron? Hardened Steel? Or both?

(b) Wheels and linings must be matched. No lining works equally well with all types of wheel material - and vice versa.

(c) Wheel finish in micro-inch.

(d) Wheel should be removable from above the brake.

(e) Wheel should be removable without disturbing adjustments if that is possible.

(f) Keep in mind wheel expansion as well as flexure of parts, lining wear allowance, and idling clearance when setting up magnet stroke.

(g) Orderly progression of wheel diameters would be:

- Torque varies as cube of wheel diameter. This is based on the assumption of uniform lining pressures and uniform ratio of face width to diameter.

(h) The heat dissipating capacity of our brake wheels has never been checked by tests. Tests would be costly and time consuming because they should be made at various speeds and percent time rotating and stationary. During the past 30 years we have used the following wheel capacities to limit wheel temperature rise to approximately 150°C.

300 ft. lbs. per minute per sq. in. wheel rim peripheral area. This capacity is based on a wheel peripheral velocity of 2500 fpm or greater with the wheel rotating at least 50% of the time.

2000 ft. lbs. per sq. in. wheel rim peripheral area for a single stop when the wheel is permitted to cool to room temperature before the next stop.

It is recognized that this is only an approximation to the wheel heat dissipating capacity, but it has been successfully used for 30 years. The heat dissipating capacity per unit area of a wheel depends upon the

- Wheel peripheral velocity
- The time on time off cycle
- The degree of ventilation provided. The capacities listed are for brakes which are not enclosed.

(1) Limiting Horsepower Rating of Motors used with the Bulletin 505 brakes.

-It is obvious that no universal limiting HP ratings can be set up because of the varying conditions encountered in service. However, our salesman must have some guide to determine when an application should be checked for heat dissipating requirements. Some years ago I set up the following basis for determining the limiting horsepower. (This was before the 30" bulletin 505 brake was developed. However, I have added the 30" data.)

-For series 1/2 hour and 1 hour, and the shunt 1 hour brakes.

One operation per minute is not frequent and many hard working jobs require the brake to stop the load more frequently. Small brakes operate more frequently than large brakes. Assume that the 8" brake operates 1.5 times per minute and the 23" brake operates once per minute. Assume further that the frequency of operation of the brakes between the 8" and 23" changes in 5 progressive steps as follows:

$$\sqrt[5]{1.5} = 1.085$$

Then	Brake	K	
	8"	1.5	operations per minute
	10"	1.39	operations per minute
	13"	1.28	operations per minute
	16"	1.18	operations per minute
	19"	1.085	operations per minute
	23"	1	operations per minute
	30"	*0.87	operations per minute

* K for 30" obtained by extending curve made from K of the other sizes.

The wheel areas of the brakes set up for the #600 motors took into account the heat dissipating capacity which would satisfy the majority of steel mill applications. It appeared logical then to select the largest brake - motor combination then standard - that is the 23" Bulletin 505 brake and the #618 motor rated 265 HP 1/2 hour duty - as basic and determine the limiting horsepowers for the other sizes. Then

(i) Cont.

$$\text{Limiting HP} = \frac{A2}{A_1} \times \frac{1}{K} \times 265$$

Where A_1 = 813 = Area of 23" wheel
A_2 = Area of wheel for which limiting HP is to be determined.

$$\text{Limiting HP} = \frac{.326\ A_2}{K}$$

Bul. 505 Wheel	A2	K	Limiting HP	
			Calculated	Use
8	81. in.	1.5	17.5	17.5
10	118 in.	1.39	27.7	28
13	235 in.	1.28	60	60
16	339 in.	1.18	93.7	95
19	522 in.	1.085	157	160
23	813 in.	1.0	265	265
30	1345 in.	0.87	503	500

- For continuous Duty Brakes.

Brakes on continuous duty applications operate infrequently. The brake can absorb 2000 '#/in^2 wheel rim peripheral area based on a single stop. If we then assume that the torque available to decelerate the load is the difference between rated brake torque and 81% (90% efficiency) F.L. motor torque, we find that the calculated limiting horsepower decreases as the wheel diameter increases. This obviously is incorrect. For this reason I have used the following basis for limiting horsepower:

$$\text{Limiting HP} = \frac{T\ N}{5250}$$

Where T = Continuous Duty Brake Torque
N = 1/2 Rated Maximum speed of brake wheel. (Takes into account fact that top speed may be 2 x F.L. motor RPM)

AEL

(i) Cont.

Brake	Cont. Duty Torque	1/2 Maximum RPM = N	Limiting HP = TN ÷ 5250
8"	75 #1	2500	35
10"	150	2000	57
13"	400	1590	121
16"	750	1250	179
19"	1500	1055	302
23"	3000	870	479
30"	6750	670	860

Continuous duty is such an elusive quantity as far as brakes service is concerned that values are a guess at best. Therefore, although there isn't any real logic to the plan, make

Limiting HP Cont. Duty = 2 x Limiting HP Int. Duty

(j) Recently a customer asked how much the rim thickness of our brake wheels could be reduced. This question has been up numerous times in the past and the answer has been that we cannot quote a minimum thickness for several reasons.

- The rim thickness as specified on our drawing for new wheels has been kept low in order to obtain a low WK^2. However some wear is expected.

- A customer asks this question when the working face of the wheel becomes deeply grooved in service. Any why does the wheel become deeply grooved? The major cause is severe over heating so metal particles are picked up and become imbedded and welded into the lining. Scoring and heating are aggrevated when this occurs. What then is the condition of the wheel rim? Does it have heat checks and/or cracks, and has the rim become warped? We do not know!

For these reasons I have taken the position that we cannot tell a user how much the rim thickness can be reduced. We cannot accept that responsibility particularly today with the court emphasis on product reliability.

(j) Added 9-13-67

(i) Cont.

Brake	Cont. Duty Torque	1/2 Maximum RPM = N	Limiting HP = TH ÷ 525D
8"	75 #1	2500	35
10"	150	2000	57
13"	400	1590	121
16"	750	1250	179
19"	1500	1055	302
23"	3000	870	479
30"	6750	670	860

Continuous duty is such an elusive quantity as far as brakes service is concerned that values are a guess at best. Therefore, although there isn't any real logic to the plan, make

Limiting HP Cont. Duty = 2 x Limiting HP Int. Duty

19. Pivots

(a) Hardened pivot surfaces are desirable for long life.

(b) Replaceable wearing surfaces?

(c) Oilite bushings at pivot points?

(d) Hardened steel pivot pins where used.

(e) Design stresses - at pivot - 150 - 200 psi
At channel sections in base - 4000 psi max.

20. Terminal Box

(a) Required if terminals are used.

(b) Watertight standard? Or option?

21. Manual Release

(a) Provide screw type (No charge?) and lever type (B88-1227).

(b) On all sizes?

(c) Manually held lever type - not automatic reset type which is not a safe device particularly on hoists.

(d) Lever type preferred. - Safe.

(e) Hydraulic operated type manual release for brakes on ladle cranes where no delay to release in emergency is vital. Large sizes only?

22. Shock Applications

(a) For Manipulator finger and other shock type applications.

(b) Manipulator finger drives. Keep in mind trouble occurs only when brake moves with the carriage and pull rod is perpendicular to the direction of the shock. That is edgewise shock.

(c) Severe shock on ingot buggy at Sheffield Steel

- and at Republic Steel. The latter installation - 60000 # ingot dropped on end of buggy knocking buggy of the track. Bul. 505 brake bases broke plus other problems on motors etc. Frequent!

(d) Need a Hi Shock construction for above applications. Also for Navy service.

(e) Widen support at tail of the armature for greater resistance to side blows and shock.

AEL

23. Hydraulic and Electro-Hydraulic Brakes

(a) Some brake parts for magnetic, Hydraulic, and Electro-hydraulic brakes. Note that Harnischfiger makes quite a point of this in their advertising.

24. Options

(a) Brakes suitable for Navy and Marine service.

(b) Weatherproof, Driptight, Dust tight, Waterproof. See C88-1138, 688-1139, C88-1157, C88-1165, and C88-1166. Also photos 505 P31, 505 Page 32.

(c) Shaft seals on NEMA IV - V enclosures. We prefer to attach enclosures to motor end bell. (Brake floor mounted). However, if shaft seal is required for motor shaft, the seal must be adjustable for proper alignment. Two shaft seals for through shaft applications requires that enclosure cover be moved several feet along shaft to make brake accessible.

(d) Lever type manual Release.

(e) High shock construction? Or can this be made standard?

(f) Vertical and ceiling mounting. A substantial construction is necessary because there may be vibration and some shock as was encountered on a stripper crane at Republic Steel - Warren.

(g) I do not believe we should consider vertical shaft mounting.

(h) G.E. catalog on IC 9528 Steel Mill Brake lists "Wall mounting with magnet up" as the only mounting option.

(i) High speed brakes as required for blooming mill screwdowns - very low voltage coil with permanent series resistor. - Special price.

(j) Time delay of application. I do not recommend listing this as an option. Obtain delay by control means.

(k) Wheel covers

25. Miscellaneous

(a) Navy - Bronze pins or bushings?

(b) Should we obtain data on time delay of application; and influence of machining variations?

(c) Our Bulletin 505 brake is the only one made by the major heavy duty brake manufacturers which does not have a flat faced armature to permit free drop of "Kish" and other dirt.

(d) Make very complete test instructions. New men now testing the Bul. 505 brake.

(e) Coil data - Drawings or "L" sheets to list

- Permissible watts - series, shunt, duty
- Minimum ampere turns to release - and hold. See A88-2034 and A70-3395

(f) Set up wheels and coils for oversize and undersize brakes for the #800 motors. Also dimension drawings. See B88-1168 and B88-1073 for Bul 505 brakes.

(g) Set up wheel "L" sheets.