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Milwaukee

Special Industries Development

Mr. C. W. Warner, Manager

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Brakes for the Proposed
AISE #800 Motors

This letter is a supplement to my letter of June 7, 1965, under the subject "Brakes for Motors Designed to New Standards". I understand that the question has been asked whether the brake can be dropped one frame size (keeping the same torque) as is proposed in the change from the #600 to the #800 mill motors (keeping the same horsepower). As pointed out in a discussion before I wrote the letter of June 7, there is that possibility; but a more logical solution from the design standpoint is to rearrange the sizes of brakes used with the motors.

Page 2 of my letter of June 7 lists several assumptions made in analyzing brakes to be used with the #800 motors. Those assumptions still hold, but a more detailed explanation of section 1.A. is now desirable. The NEMA brake standard IC 1-20.21 is primarily based on the PRESENT NEMA Industrial Control Standards parts 43 and 44 for Cab and Floor operating cranes. These standards state that the electrically operated brakes are to have sufficient torque to hold at least full load motor torque. Thus, in the present standard setup for the NEMA-AISE brakes with the #600 motors, the ratio of brake torque to F.L. motor torque ranges from 1.01 to 1.72. I particularly stress the PRESENT standards which permit brake torque to be equal to F.L. motor torque because, since the letter of June 7, I received a copy of Mr. Thom's letter of June 4. In that letter he states that NEMA is considering changing the Industrial Control Standards parts 43 and 44 to specify in part "The holding brakes for hoist motors with control braking shall have a rated torque of not less than 150% of the full load hoisting torque at the point where the brake is applied". Note that this is not F.L. motor torque! I am quite sure that NEMA-AISE will set up brake torques based on a minimum percentage of F.L. motor torque rather than on a minimum of 150% full load hoisting torque.

The following discussion is based on brake torque being not less than 100% F.L. motor torque in accordance with present NEMA Standards.

I. Print SK063065-1-AEL shows three brake combinations for each of the proposed #800 motors.

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A. Columns 5 and 6 show the present brakes retained with the motor horsepower. As explained in my letter of June 7, some brakes will have to be mounted in a "well" because the height from the mounting surface of the brake to the center line of the wheel is greater than the height from the mounting surface of the motor to the center line of the motor shaft.

B. Columns 7 and 8 show the brakes required if they are dropped one frame size, that is the brakes are retained with the motor frame size, not with a horsepower rating. With these brake-motor combinations, no "well" is required for mounting any brake.

However, there are objections to this setup. First, the torque rating will have to be 2 to 2.8 times the present ratings depending upon the size of the brake. These great increases in torque ratings will present extremely difficult design problems. And second, the width of wheel face will change from a range of 37-49% of wheel diameter to a range of 52-81% of wheel diameter. The greater the width of wheel face, the better the quality of machining required to ensure proper performance of the brake. We have machining problems today because of the great width of wheel face.

C. Columns 9 and 10 show brake and motor combinations which I would propose. The wheel diameters of the present Bul. 505 brakes are still used, but the brake is not dropped one frame size as is being done with the motors.

There are two advantages to this proposed setup. First, the torque rating will be 1.2 to 1.85 times the present rating of the brake depending on size; and second, the width of wheel face will be 56 to 62% of the wheel diameter. Even this width of wheel face will present difficult machining problems.

II. The increase in torque rating and the increase in horsepower with which a specific wheel diameter is to be used present real problems in the design and manufacture of brakes. Following are some of those problems:

A. Torque

How can the torque of the brake be increased?

For a specific wheel diameter, torque can be increased by changing the coefficient of friction of brake line or by increasing shoe pressure. Increasing wheel width in itself does not increase torque.

1. On the surface, it would seem easy to simply select a lining with a higher coefficient of friction, but unfortunately our experience indicates that we cannot use the coefficient published by lining manufacturers. That is why for the greater part of more than 40 years, Cutler-Hammer has conducted tests on brake linings almost continuously. Four years ago, however, the formal test program was dropped. We did check a Johns-Manville #160 lining for a specific application. That lining was claimed to have an extremely high coefficient of friction. Our tests showed that at high shoe pressures, the coefficient was .50, but that it dropped radically to .20 when the lining pressure was in the range of that on the one hour duty series brakes.

   Apparently, the major lining manufacturers have learned a lesson and now qualify ~~for~~ their published ratings. For example, Johns-Manville formerly listed the coefficient of friction of their #600 lining (our present standard) as .45 ± .07, but their literature today adds a qualifier saying that this coefficient is a guide only, and that the values should be used with a 20 to 40% factor of safety. Except for the #160 lining mentioned earlier, the #600 lining has as high a published coefficient of friction as any Johns-Manville lining.

   Raybestos-Manhatten now say that the normal coefficient of friction which they quote must be used with a factor of safety ranging from 25 to 50% to take care of mechanical losses (?), variations due to temperature changes, design of brake, etc.

   Lining manufacture representatives send literature and call quite regularly to discuss lining problems. I have not heard of any lining suitable for heavy duty steel mill service with a coefficient of friction that will always remain above .30 during its life. That is the coefficient of friction used in the design of the Bul. 505 brakes. The original design of the type M brake was based on a .25 coefficient. The NEMA-AISE committee which set up the brakes for the #600 motors agreed on the .30 coefficient although some brake manufacturers wanted to use .35 and others felt that .30 was too high based on their experience in the field.

If we go into a new brake design, we would certainly check thoroughly the possibility of linings with higher coefficient of friction.

2. This leaves us with the alternative of increasing torque by increasing brake shoe pressure. This can be done by using a stronger magnet and raising the magnet center line above the center line of the wheel. As a matter of fact, increasing magnet pull means a larger magnet which in most instances will require that the magnet be higher.

   A larger magnet will be necessary because we already use Class B insulation in our series brake coils and changing to Class H insulation alone will not permit the higher torque ratings required except possibly in the 8" brake as proposed in Columns 9 and 10 of SK063065-1-AEL. Keep in mind that the brake coil is stationary and does not have the advantage of rotation and built-in fan cooling which is possible in a motor.

   Increasing brake shoe pressure will require strengthening the brake parts. The torque spring will have to be larger and, if kept at the center of the magnet, will increase the size of the magnet structure. The higher forces will also require a change in the pivot construction at the shoe lever and armatures in order to reduce wear of parts.

B. Brake Wheel

1. The wheel area is determined by the heat dissipating capacity at a selected temperature limit.

At committee meetings which set up the present NEMA-AISE steel mill brakes for the #600 motors, there was much discussion of this feature. Some favored a maximum lining pressure of 30 PSI and some favored a maximum of 35 PSI. The final setup was a compromise of a progressive increase from 25 to 35 PSI as wheel diameters increase. All felt that liberal wheel and lining area was a most important consideration for adequate heat dissipating capacity and lining life.

2. With the diameter of the wheel determined, the minimum width of wheel follows because of the area required for heating dissipation and lining life. Because of the reduced wheel diameter at the same motor RPM, the wheel peripheral volocity is reduced. Therefore, the heat dissipation capacity of the wheel is also reduced. This fact will have to be taken into consideration in the final design of the brakes.

   Decreasing wheel diameter and increasing wheel face will undoubtedly result in serious objections by crane builders and steel mill people for two reasons. First, there will be a large overhang of the wheel on the motor shaft; and second, the brake mounting screws on the motor side will generally be less accessible. These were two of the critisisms of crane and mill people to the present NEMA-AISE brake standard at the time the committee was setting up those standards. These people appeared to be less concerned about a "well" to mount the brake than they were about the requirement for greater wheel offset necessary because of the wide wheel face. Crane manufacturers are usually severly limited in horizontal space for the motor and brake; and the greater the wheel offset required, the greater the problem.

3. From the manufacturing standpoint, the wider the wheel face, the more difficult the machining problem because of the tolerances in parallelism of pivots, wheel face, lining etc.

These are some of the problems involved and are only indications of what might be encountered in setting up new designs. A major point to be taken into account is that the #800 motors have not yet been standardized; and there is the question of what affect a change in Industrial Control Standards parts #43 and 44 for Cab and Floor operating cranes will have on brake torque ratings. I might mention that at one time the NEMA committee on brakes for the #600 motors considered specifying that brake torque must not be less than 125% full load motor torque. It is possible that that specification may be revised.

A. E. Lillquist

A. E. Lillquist
Development Supervisor

AEL/mlk

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