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LIGHTNIN SPX: HIGH TORQUE TEST STAND

Geoff Cusano, ME **Dana Harris, IE**

Ron Mendolera, EE **Aditya Sanghi, IE *Project Leader*** **Oriana Starr, ME**

Don Strong, ME **Jesse Warner, EE**

ABSTRACT

The goal of this project was to evaluate the final assembly and testing operations for the 700/800 Series gear reducers produced by Lightnin SPX of Rochester, NY. The project operated within constraints of time, space, and budget. Process inefficiencies were identified and solved, which specifically required the redesign and relocation of the company's torque test stand. The test stand was redesigned to minimize its footprint and height, as well as the setup and process time involved in its operation.

INTRODUCTION

This project was for a local Rochester company, Lightnin SPX. Lightnin has been producing pumps, mixers, and other products since 1923, however the need for this sort of team analysis arose when the gearbox production moved to Rochester from Wytheville, VA. The Rochester facility was not as well equipped for this sort of production, and as a result the process developed many inefficiencies over the years. The scope of this project includes just the final assembly and testing operations for these gear reducers. The assembly of these gear reducers is rather complex, due to their large size and weight. During assembly, each box must also be rotated 180 degrees upside down in order for a specific preload to be set on the roller bearings inside. After assembly, each unit must undergo a spin test. This test simply spins the gears for a given time, to ensure no gross malfunctions occur. In some instances, the gear reducer will then be subject to a torque test, to simulate the load experienced in the field. Not only are the assembly and testing processes of these reducers complicated, but simply transporting these large, heavy products is somewhat involved as well. The units must be transported by forklift and/or crane from the assembly station, to the spin test station, to the torque test station, and finally to the shipping area. In some cases, specifically with the torque test, the station was located on the opposite side of the facility from the other areas.

This project with Lightnin SPX had three main objectives. First, the torque test stand was to be removed from the shipping dock where it was currently located. This required the stand to be redesigned with a reduced height, in order to fit within the facility's height constraints. The second goal was to remove obvious waste from the final assembly process. Since Lightnin hoped to ramp up towards 100% torque testing from the 20% currently being tested, the group would ultimately develop a process that allows the gear reducers to be torque tested in the time it currently takes to spin test them.

These objectives were to be met given certain constraints. First, a budget was set at $200,000 and the duration of the project was to last six-months. The next constraints were facility-related. The largest gearboxes required a large crane capacity that was present only in certain parts of the facility. With regard to space, if the new torque test stand were to fit within the current assembly area, it had to fit within the area currently occupied by the spin test stand. The group was also constrained in that they had to adhere to the current design of the gear reducers, it could not be altered.

NOMENCLATURE

All nomenclature is defined below, corresponding to the variables for each equation that is listed.

PRODUCT DESCRIPTION

The product under consideration is Lightnin's 700/800 Series gear reducers which are part of heavy-duty mixers used for all sorts of industrial purposes, from pharmaceutical fermentation to heavy sludge waste treatment. This series of reducers offers four different sizes, ranging from the smaller 780 to the 5-ton 783 model. Each size is available in either double or triple reduction, can have different arrangements for the input and output shafts, and the shafts themselves can either be hollow or solid.

CURRENT PROCESS DESCRIPTION

<u>Assembly</u>

The scope of this project includes only final assembly and testing. The design of these reducers, coupled with their large size and weight results in the need for very skilled workers and special equipment for assembly. The gear reducers are assembled on a stand which has the capability of rotating the reducer 180 degrees upside down. This action is necessary to set the appropriate preload on the roller bearings inside the reducer. It is important to note that the smaller 780 and 781 reducers are assembled in a different location than the larger 782 and 783 reducers.

<u>Spin Test</u>

After assembly, each unit is transported to the spin test stand. The larger reducers are assembled next to the spin test stand, so may be transported by a large crane. The smaller reducers must be craned off of their assembly stand onto a pallet, fork trucked down the aisle, and finally craned from the pallet onto the spin test stand. Once on the spin test stand, the appropriate sheaves and belts must be connected from the input motor of the gear reducer to the shaft of a 60HP motor. The reducer is spun at 1780rpm for approximately 30 min. The purpose of this test is simply to test for gross defects in the product, and came to be performed simply as an alternative to torque testing every unit.

<u>Torque Test</u>

Currently, about 20% of the annual production volume undergoes torque testing. The purpose of the torque test is to simulate how the unit would operate in the field under load, as well as to do a quality check on the wear pattern of the gears. The existing torque location is also capable of performing the spin test, so in some instances the gear reducer is transported directly from the assembly stand to the torque stand, thus eliminating the extra trip to the spin test stand. The torque test is performed for various reasons, including a request by the customer, a problem experienced during assembly or spin testing, or just "gut feel" by the production manager. The torque test stand is located approximately 500 ft across the facility from the assembly area, so the reducers being tested must be fork trucked and then craned onto the torque test stand.

The torque test stand's main components include an in-line drive system, a slave unit, and a brake. The drive system consists of a 60HP motor, a 40:1 gear reducer, and a 10HP motor. The 60HP motor may be used to run the spin test, while the 10HP motor and the gear reducer are used for the torque test. The output shaft of the reducer being tested is coupled to one of Lightnin's 782 gear reducers which is referred to as the slave unit. The slave unit's function is to convert the high torque to high speed for more favorable braking conditions. The output shaft of the slave unit is attached to a pneumatically-controlled brake, which has greater stopping power as the air pressure increases. This braking leads to an increased torque at the input shaft of the reducer.

Extensive setup is required before the reducer can be craned and bolted onto the stand. First, spacer plates must be added to the stand in order to set the reducer's input shaft in line with the test stand's motor. A second operator must work underneath the stand to set the slave unit at the appropriate height for the specific reducer size that is being tested. While this setup is taking place, the brake transmission oil and the lubrication oil for the reducer must heat to steady temperatures. Finally, couples must be bolted on to connect the reducer's input shaft to the stand's drive system motor, and the reducer's output shaft to the stand's slave unit. Once setup is complete, the reducer must be bolted onto the stand, and the lubrication oil must be pumped in.

During the torque test, the operator must manually control the air pressure to apply various torque levels. The operator must also continuously make fine adjustments during the test in order to keep a consistent torque, and the test must continue until the reducer undergoes at least ten revolutions. Once the test is complete, the gears inside the test reducer are inspected and photographed to ensure a normal contact pattern was created.

PROCESS ANALYSIS

Time studies were performed early in the project, in order to identify the largest wastes in the process. Unexpectedly, the torque test setup time, NOT transport time was identified as the largest inefficiency. Given this data, the first focus for the torque stand design was to reduce the setup time involved. Many concepts were considered, and three main areas for improvement in the stand were identified. The first involved removing the trial-and-error methodology of adjusting the proper slave height according to the size of the reducer being tested. Next came an effort to minimize the time spent waiting for the brake transmission oil and the lubrication oil to heat to steady temperatures. Finally, the precision tuning of the air pressure regulator to achieve consistent torque had to be standardized.

Soon after the time studies indicated the major inefficiencies, Lightnin's management met with the group and revealed further information regarding the company's future goals. They indicated that they would like to reclaim the shipping dock that the torque test stand was currently located in, and that given recent warranty issues they would like to ramp up to 100% torque testing. Other goals set for the team included removing waste from the current process and ultimately attempting to allow the torque test to be performed in the same time that it currently took to spin test.

In order to locate the current torque test stand within Lightnin's facility, the group had to make the stand shorter to accommodate height constraints. The stand also had to be relocated to an area that had the large crane capacity to support the 783 reducer. The limited crane capacity left two possible areas within the facility. These included the current large assembly area, which meant that the spin test stand would have to be replaced by the new torque test stand which could also perform the spin test. Or the "back bay" area which was located just inside the door from the shipping dock where the torque test stand currently sits.

REDESIGNED TORQUE TEST

The group quickly got to work focusing on these areas for improvement, while also considering the constraints of time, space, and money. The input motor system would be automatically adjustable both horizontally and vertically in order to easily align the output shaft of the motor drive to the input shaft of the reducer. A universal plate that could accommodate bolt holes for all of the reducer sizes would be designed, allowing all of the reducers to sit at the same height. It was determined that the setup time for the torque test stand could be reduced if the coupling method was universalized to be less tedious. Insulated oil tanks with heaters will also be incorporated into the new design in order to eliminate the time spent waiting for the oil to warm up. Finally, an electronic air pressure regulator will be installed, which will be controlled by a PLC collecting feedback from a torque sensor.

Adjustable Input Motor

An adjustable input motor drive system eliminates the need for the current shaft and coupling system. A hydraulic scissor lift will provide vertical motion, while heavy duty rails will accommodate horizontal movement. A traveling nut lead screw actuator will power the horizontal movement, which will result in a maximum travel time of about 40 sec.

Input Motor Controls

The scissor lift will be controlled by a position sensor which will be used in a feedback loop with a PLC to allow exact height adjustments. Instead of performing tedious adjustments, all the operator must do now is select the appropriate reducer model on the PLC interface. The lift will adjust to one of eight different elevations, accordingly.

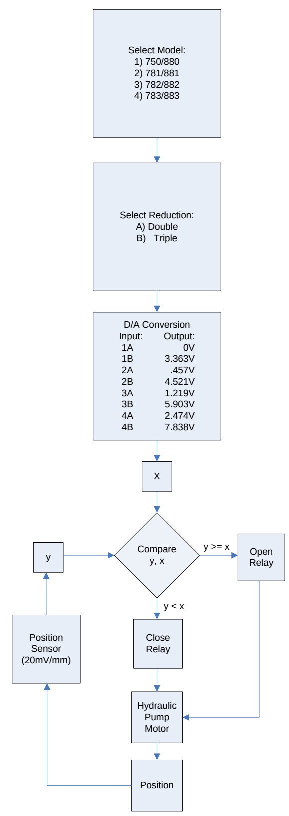

Figure 1: Block Diagram for Position Control

Input Couplings

The connection between the input motor's output shaft and the input shaft of the reducer will now be achieved by a simple coupling system, as seen below. The male coupling will be bolted to the output shaft of the input motor, and the female coupling will be slid over the end of the male coupling to attach the input motor to the input shaft of the reducer. Two keyseats have been cut into one end of the coupling which means that the most the operator will have to adjust the coupling is 90 degrees, thus making alignment easier and less time consuming. The running fit of the keyseats accommodates easy removal.

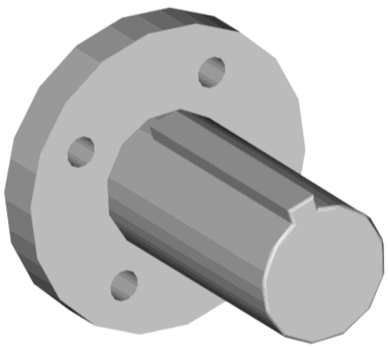

Figure 2: Male Universal Input Coupling

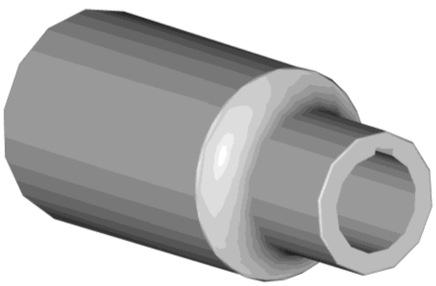

Figure 3: 783/883 Input Coupling

Output Couplings and Splines

The new output couplings will eliminate the very tedious task of positioning the slave unit underneath the torque stand. A permanent female coupling with female splines is attached to the slave, which never has to be removed. Male splines are cut into the end of each coupling so that the slave can be attached to the coupling without any adjustment of the couplings themselves. Once the coupling is attached to the output shaft of the gear reducer it only needs to be craned into place, and the attachment to the slave is complete. The number of output couplings has been reduced to three for the 700 series and three for the 800 series, which are solid and hollow shafts, respectively. Eye bolt holes have been incorporated into each coupling, so that they may be transported by crane.

Heated Oil Tanks

With the addition of these insulated oil tanks with heaters, the test operator will no longer have to spend an hour running the unit under low torque while the oil heats up. The tanks will not need to be heated all of the time, just for an adequate amount of time before a test will be run.

Brake Control

Air pressure will now be adjusted automatically through the use of feedback from a torque sensor, an electronic air pressure regulator, and a PLC unit. The operator will simply enter the desired amount of torque in the PLC interface based on the model being tested. The PLC also incorporates a redundant safety measure to ensure the input shaft of the test unit never experiences a destructive amount of torque.

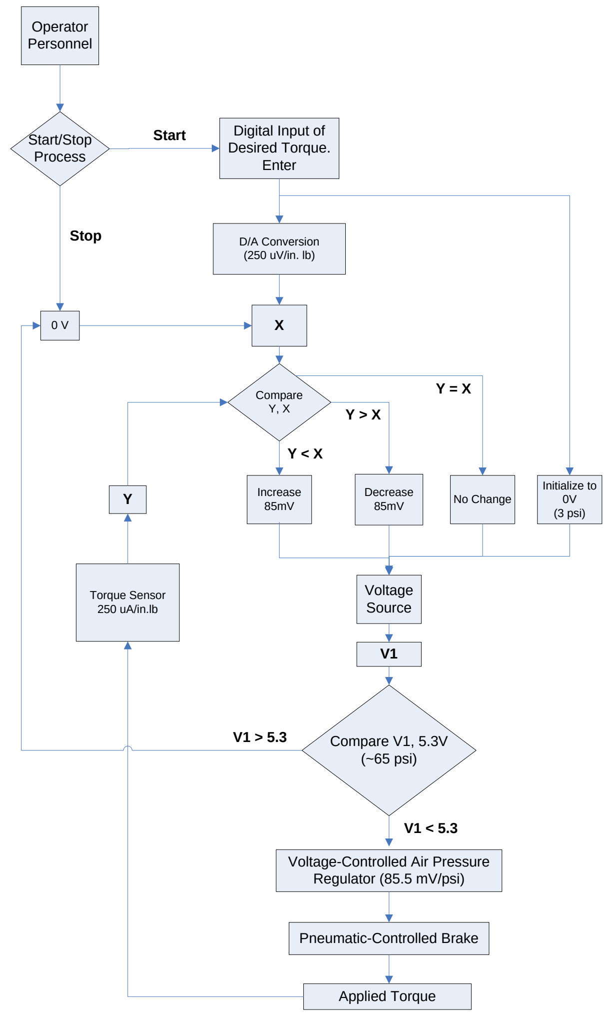

Figure 4: Torque Brake Control Digram

Overall Structure

The overall stand footprint is 9' x 19'. It will be a fully welded, two-tiered steel construction with all legs anchored to the ground.

Data Collection

Another desirable outcome of the project was to condense the data for each reducer tested into a more simplistic form. An interactive program was written using Visual Basic to keep accessible records of the tested reducers. An easy to fill out form was created to capture data including reducer specifications, sound, and temperature data.

DESIGN TESTING

Horizontal Motor Movement

The rail system for horizontal movement of the input drive system was tested for critical loading which includes the overall weight, the transverse force due to the torque of the gear reducer, and the moment caused by the transverse force when the lift is fully extended.

Couplings

Stress analyses on the all couplings were done using the energy distortion method. A recommended 75% of the shear strength (for shafts with keyseats) was taken as the allowable stress on the shaft. Below are the general equations and constants used in the stress analysis.

$$s_{sy} = .577 s_y \quad Shear \quad Strength \quad (Energy \quad Distortion \quad Theroem)$$ **Equation (1)**

$$\tau_{all} = .75 s_{sy} \quad Allowable \quad Shear \quad Stress$$ **Equation (2)**

$$\tau = K \frac{Tr}{\frac{\pi}{2} D^4} Shear \quad Stress \quad Solid \quad Shaft$$ **Equation (3)**

$$\tau = K \frac{Tr}{\frac{\pi}{32}(D^4 - d^4)} Shear \quad Stress \quad Hollow \quad Shaft$$ **Equation (4)**

Where,

S_y = Yield Strength
τ = Sheer Stress

A fatigue analysis was also performed on all coupling designs. All coupling exceed a million cycles which indicates infinite life. The following are the equations and constants used for this analysis.

$$Se' = .504 S_{ut}$$ **Equation (5)**

$$Se = se' kakbkcke$$ **Equation (6)**

$$N = \left(\frac{\tau}{a}\right)^{\frac{1}{b}}$$ **Equation (7)**

$$a = \frac{0.9 S_{ut}^{2}}{Se}$$ **Equation (8)**

$$b = \frac{1}{3} \log\left(\frac{.9 S_{ut}^{2}}{Se}\right)$$ **Equation (9)**

Where,

S_{ut} = Ultimate Tensile Strength of Material
Se' = Endurance limit of material
Se = Endurance limit of mechanical element
N = Cycles to failure

Endurance Limit Modifying Factors[1]

k_a = Surface Factor
k_b = Size Factor
k_c = Load Factor
k_e = Stress Concentration Factor

1. Endurance limit Modifying Factors taken from Shigley & Mischke.

Below are the equations used for the Female Output Couplings.

$$p = \frac{E\delta}{R}\left[\frac{(r_o^2 - R^2)(R^2 - r_i)}{2R^2(r_o^2 - r_i^2)}\right]$$ **Equation (10)**

$$\sigma_{it}(R) = -p \frac{r_o^2 + r_o^2}{r_o^2 - r_i^2}$$ **Equation (11)**

$$\sigma_{ot}(R) = p \frac{r_o^2 + R}{r_o^2 - R}$$ **Equation (12)**

Where,

p = Interface Pressure
σ_{it} = Tangential Stress of inner member
σ_{ot} = Tangential Stress of outer member
r_o = Outer Radius of outer member
r_i = Inner radius of inner member
R = Transition Radius
d[1] = Radial interference

1. Fit tolerances given by Machinery's Handbook 26th Edition. Class 5 Press or Force fit was used.

Splines

A 6/12 pitch was chosen as a good compromise between strength, ease of indexing, and cost to manufacture. With 60 teeth the spline can only be 6 degrees out of alignment.
The table below from the Machinery's Handbook 26th Edition shows the basic geometry of the involute spline.

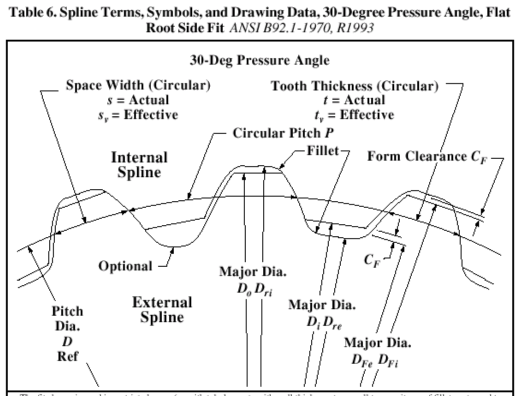

Figure 5: Involute Spline Geometry

The Handbook also gave all of the required equations to determine the appropriate sizing of the tooth profile as well as to analyze the stresses within the spline.

Once dimensions were determined the splines were analyzed for shear stress, compressive stress and tensile stress loads that would occur in the coupling. The splines had to hold up to a torque of 310,00 in-lb as well as a maximum speed of 232 rpm.

The resulting stresses were then calculated and compared with the known maximums to determine if the splines will fail. The splines are able to easily withstand the required torque due to the oversizing of the Diametral Pitch made necessary by the need to mount the male coupling on existing solid shaft gearboxes with no changes to existing gearbox parts.

Stress Comparison for Steel of Brinell 230-260 Hardness					
	Actual			**Maximum**	
S_s	2034.986	psi		30000	psi
S_h	2918.008	psi		30000	psi
S_p	3493.164	psi		30000	psi
S_c	301.5758	psi		2000	psi
S_1	1428.861	psi		32000	psi
S_2	15.29651	psi		32000	psi
S_3	3015.758	psi		32000	psi

Figure 6: Stress Comparison for Steel

FINAL LAYOUT

The final design of the torque test stand allowed two final layout options to be proposed.

Option 1

This option simply involves replacing the existing spin test stand with the new torque test stand. While this option is definitely the easiest to implement, inefficiencies will still be present as a result of the large and small assembly areas still being separated.

Option 2

This alternative is slightly more involved, and entails relocating all final assembly equipment, including the redesigned torque stand, to the "back bay" area. This option would create a final assembly and testing cell. The vast majority of parts for these reducers are purchased, meaning that they could be received into the dock now vacated by the torque test stand, and could immediately be placed on shelves without having to travel further inside the building. Once assembled and tested, the finished reducers would flow towards the front of the building for shipping. Minimal backflow would occur from the small amount of labor added to the non-purchased parts.

ACKNOWLEDGMENTS

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