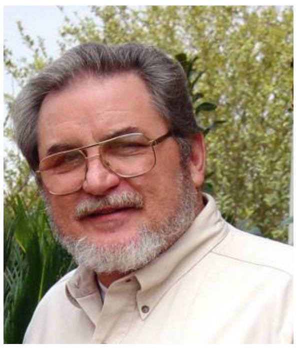
TPS
TURBOMACHINERY & PUMP SYMPOSIA | HOUSTON, TX
SEPTEMBER 15-17, 2020
SHORT COURSES: SEPTEMBER 14, 2020

A NEW APPROACH TO LARGE MOTOR DRIVEN PUMP TRAINS

Fred Lea PE
Rotating Machinery Services Division Manager
Becht
Houston, TX, USA



Mr. Lea started his career with DuPont as a support engineer. He developed and honed his expertise in machinery and rotating equipment over the next 40 years working in the oil, petrochemical, power, and mining industries as an engineer, mechanical superintendent, and rotating equipment consultant for a number of major companies including DuPont, Vista Chemical, Fina Oil, and ARCO Polymers. Mr Lea also worked for Fluor Corporation for number of years where he developed a number of machinerystandards and specifications for a number of clients as well as participating in the design, procurement, installation and start-up of a number of large rotating equipment installations. He is a Registered Professional Engineer in the state of Texas since 1984, and has served on the API Subcommittee on Mechanical Equipment (SOME) since 1987 where he has served on numerous standards task forces and, for several years, was the API 618 Reciprocating Compressors Task Force Chairman.

Mr. Lea received a BS in Mechanical Engineering from Lamar University.

ABSTRACT

Sometimes when tasked with an effort that has been done many times before, the same path is followed over an over again. Sometimes the saying "Just because we have always done it that way, doesn't mean it's right" is very applicable. Sometimes when we break old paradigms, we get better results.

INTRODUCTION

Recently on a large pipeline project in South America, several issues that constantly surround large frame motors in pump applications again reared their ugly heads. This time, instead of accepting this as "we've always done it this way"; these paradigms were questioned. The results were pleasantly surprising. Working with one of the major large motor vendors and pump vendors, the issues were worked through one by one, and the result was a much better application than if the "normal" practices had been accepted at face value.

1. Pump Selection

With the exception of the discharge pressure, these pumps were pretty non-descript. They were relatively high volume, high pressure water pumps. When looking for an industry standard to build the pumps to, there is not really any standard other than API 610 that addresses pumps of this pressure and capacity. ASME does not really have a standard that addresses them, and the Hydraulic Institute High Energy Pump document does not address pumps operating over 500 PSIG discharge pressure. These are large multi-stage, high-volume high-pressure pumps. They were going into a relatively undeveloped country as far as industry was concerned, and reliability and maintainability were key in the selection. The Decision was made to utilize API 610 Type BB3 pumps. All of the major pump manufacturers make this pump and can be competitive in the market.

Water pumps are normally furnished in Cast Iron materials for applications such as this. In this case, due to the remoteness of the application, concerns about corrosion in the desalinated water, and desire for repairability of the casings if necessary, API Appendix H Designation D-1 was chosen as the material class rather than the normally selected Class I-1, Cast Iron.

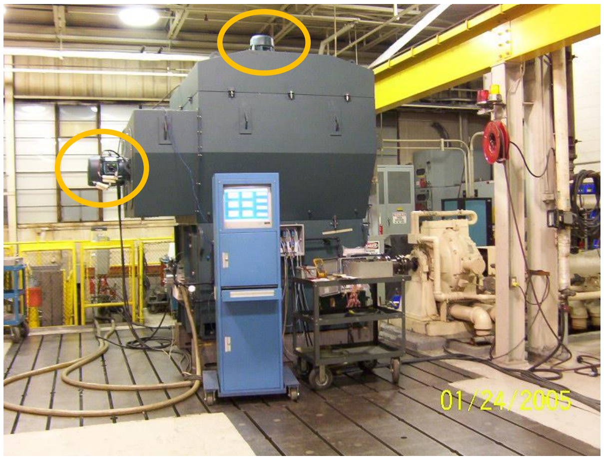
As usual, the actual pumps were a minor part of the total cost of the equipment train and auxiliaries. The packaging and auxiliary equipment systems comprise a large part of the cost. The evaluation of the pumps is pretty standard based on performance parameters (efficiency, %BEP, rotordynamics, etc.). The evaluation of the "other" components is the issue.

2. Motor Selection

All of the pump stations were housed in buildings as they were located in the desert. Blowing sand was a real issue. As a result of this, the motor heat had to be conducted outside of the pump station building. The decision was made to duct the cooling air from the motors outside the building.

The typical enclosure selected for this application would be Open Drip Proof. In this case there were other issues involved: forcing the cooling air outside of the building as well as controlling the noise level inside the building. This was accomplished by using externally cooled TEAAC drivers with the external cooling fan providing the motive force to push the air through the exchanger and out of the building. There were two methods available to accomplish this cooling; shaft driven fans or external motor driven fans. The external motor driven fans were recommended in that the squirrel cage blowers would be more effective in moving the air through the ductwork as opposed to the typical shaft driven axial fans in that they could provide more static pressure than the axial fans.

While the reasoning behind the selection of the external motor driven fans was more effective air movement, there were some very interesting side benefits as a result of this selection. The normal shaft driven motor fans are not only noisy, on large motors they contribute to the overall shaft vibration of the motor.



The drivers in this particular application were 50 cycle, 3000 nominal rpm, 3350 hp direct driving six stage horizontally split API 610 Type BB3 pumps.

During the bidding stage, four of the top international motor vendors offerings were submitted by the various pump vendors. There was an incredible range in the proposals offered. While the motor horsepower was specified, the manufacturer's selection of frame size and motor configuration to achieve this horsepower varied significantly. As the price of the motors was a significant cost in the pump train, an independent evaluation of the motors was performed.

In evaluating a driver for this application, rotordynamics and driver efficiency are major considerations. Rotor shaft length and maximum diameter varied over a large range from vendor to vendor, as well as bearing sizing and centers, rotor overall length, rotor maximum diameter, and overall frame rigidity and dimensions.

After extensive analysis, a motor driver manufacturer chosen was selected. As a result of this decision, all pump vendors were required to submit revised proposals with the selected drivers during bid conditioning. As a result of this activity, the motor selection was taken out of the equipment equation and each of the pumps could be evaluated on their own merits and pricing.

Once the motor vendor was pre-selected, it was easier to work with a single motor vendor to improve some of the motor characteristics that were required for this application as well as issues that are inherent for maintenance of large drivers. These are described below.

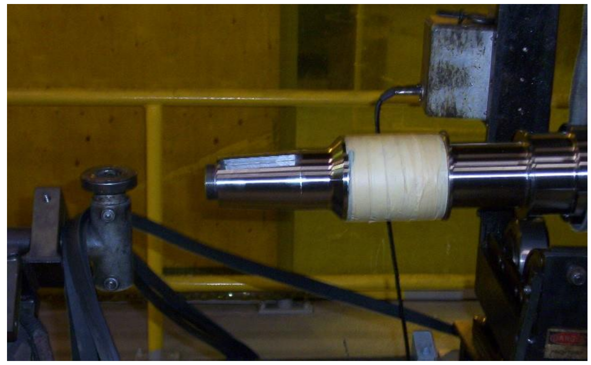
3. Shaft Junctures

The next step in the process is determining the type of shaft juncture required. Even though the horsepower of the pumps was 3500, the shaft speed was well below four thousand rpm. Hydraulic fit couplings were deemed unnecessary for this application. In discussions with the client, there was some concern about field removal of a straight shaft, shrink fit coupling of this size. Most of the pump stations on this pipeline application are very remote, and any tools required for coupling removal would have to be transported to and from the site by contract maintenance personnel. After reviewing the options, a keyed, taper fit hub was determined to be the most advantageous selection. It does not require heat or special tools for removal and installation, and it would easily meet the load requirements for the pump and motor.

As in most cases, the driver shaft diameter was much larger than the driven unit. With some innovative engineering, this was overcome which resulted in some other benefits.

In reviewing the drawings, the pump input shaft diameter was 3-5/8" nominal. The motor shaft, even though made of a stronger material than the pump shaft, was 7" in diameter. An inquiry to the motor manufacturer questioning the motor output shaft sizing revealed that the motor output shaft sizing criteria had a 10:1 safety factor built in.

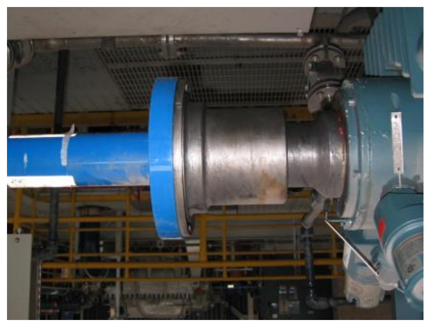
The motor manufacturer was asked to evaluate reducing the motor output shaft size to match the pump input shaft. When sized to meet the pump input shaft diameter, the motor output shaft still had a 4.3:1 safety factor. Since this appeared to be more than adequate for this service, the motor manufacturer agreed to reduce the motor output shaft diameter to the same as the pump input shaft diameter and put the same taper and key size on it as the pump shaft. The result being that the motor coupling hubs would be interchangeable with the pump hubs, thus reducing the amount of spare parts required



Pump Shaft End **Motor Shaft End**

3. Coupling Selection

The common industry coupling selection for this application is a disc pack style coupling. In going through the sizing program for couplings, as usual, the motor shaft diameter determined the coupling size. The motor output shaft diameter was 7 inches. In order to accommodate the maximum allowable bore size for the coupling, the coupling size was determined to be a balanced size 700 for reference. This disc pack coupling, including spacer would have weighed 613 lbs.



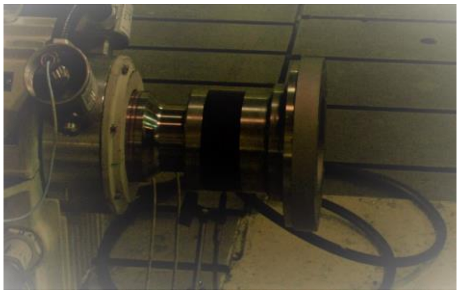
Complete Coupling **Motor Coupling Hub**

This reduction in shaft size somewhat mitigated the coupling overhung weight issue common in this type of installation. To further reduce the weight, a double diaphragm coupling was selected over the disc pack style coupling. And, combined with a better coupling selection (Diaphragm over Disc Pack), resulted in reducing the total coupling weight to 76.5 lbs., a savings of 536 lbs. of coupling overhung weight. The originally proposed coupling spacer weighed 461 lbs. The selected coupling spacer weighed 27.5 lbs.; a reduction of over 433 lbs. of spacer weight from the original selection. This obviously translated into better rotordynamics as well as some other benefits. It was much easier to handle the smaller spacer, eliminating the need for the use of a crane to install and maintain the coupling.

As with the motors, an independent evaluation of the coupling offerings resulted in the requirement for each pump Vendor to furnish the same make and model of coupling in each of their offerings. Again, the intent is to evaluate the pumps without having to worry about the differences in the effects and cost of the periphery equipment.

4. Balancing

Balancing has seemed to suddenly become an issue, even in industry standard documents. For years the standard API balance Criteria was 4w/n. There was no question as to what the balance criteria was to be. It was 4w/n. Over recent years, for whatever reason, G 2.5 has seemed to creep into a lot of standards. It is hard to understand the reasoning behind it. There is essentially no cost savings to be had by reducing the balance requirements of a rotor. With today's computerized balance machines, once the rotor is prepared and placed in the balance machine, the difference between G 2.5 and 4w/n balance is about one more balance run than G 2.5 requires, or approximately 5-10 minutes more in the balance machine, depending on the size of the rotor (somewhat more if an at speed balance is required). In this application, all components and rotors, both pump and driver, were balanced to 4w/n as opposed to the standard G 2.5 required by many specifications. During the mechanical run testing of the motor, a balanced coupling moment simulator should be used. Note: a minimum of ISO G 1.0 or better is recommended by this author.



Moment Simulator Mounted

Another issue is coupling balance. Many specifications call for couplings to be assembly balanced and match marked. This is totally unnecessary and has been proven in practice. If all coupling components are balanced to 4w/n, or better (this author usually requires coupling components to be balanced to the limits of the balance machine), no assembly balance correction is normally required (although a balance check may be advisable on a spot basis as assurance of compliance). Again, by doing this, it gives a lot more flexibility in the utilization of spare parts. You don't have to scrap an entire coupling if one of the components is damaged.

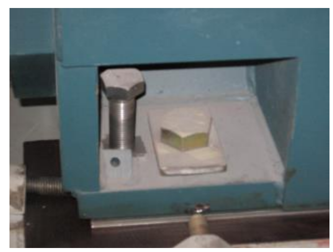
6. Alignment Concerns

Many times, there are issues with bolt binding on drivers in the field during alignment. Since these unit were going into a very isolated location, some changes were made in the motor hold down arrangement.

Instead of the standard 1/8" diametral clearance normally found in motor installations (0.062 in. per side) the bolt holes in these motors were elliptical in shape. This allowed for a liberal amount of clearance around the hold-down bolts to facilitate horizontal alignment and axial positioning. The motors were supplied with a hold-down plate designed to cover the opening and provide a secure clamping force on the motor.

Standard Motor Bolt Hole

Elongated Hole



Motor Hold-down Clamp

CONCLUSIONS

The results of attention to detail for all the little things resulted in the smoothest running units this author has ever installed. Even the electric motor driver field representative remarked that he had never seen pumps trains run as smoothly as these. Unfiltered vibration for fully loaded 3350 HP trains was less than 0.15 mils nominally per the installed proximity probes. While balancing of the trains to better than 4w/n was a contributing factor, the absence of the shaft driven fans on the motor rotor quieted the motor vibrations significantly. The conclusion is that the use of TEAAC Externally Fan cooled motors had a large impact on the overall performance of the train as did the use of the Diaphragm Style couplings as opposed to the conventional disc-pack style couplings.

Aside from the excellent operating results, the noise levels of the trains were extremely low. The majority of the noise in motor driven pump trains ironically comes from the shaft driven fan on the motor. While we did not have instrumentation to measure the Sound Pressure Levels, the estimated sound levels were in the 80-85 dB range as opposed to the 95-105 dB levels of an open, drip-proof motor driven train. The end result was quiet, smooth-running trains.

.