CST, Blackstone collaborate on API 618 reciprocator retrofit

Many considerations necessary to transition from a gas engine to an electric motor

In 2015, CST supported Blackstone for the replacement of an engine driver with an electric motor with variable frequency drive (VFD) on a Dresser-Rand 5RDS-4 (Southwest Industries - Ingersoll Rand) reciprocating compressor. The machine was a four-stage reciprocating compressor with five cylinders that had been operational since 1954 in boil-off service in a natural gas plant located in Tulsa, Oklahoma.

The customer wanted to transition to an electric motor drive with VFD to expand the capacity control and increase the reliability of the unit. When a new motor is installed, especially when switching from a gas engine to an electric motor, the required brake horsepower and shaft diameter must be respected, along with the mass and torsional behavior.

The new train arrangement had to be designed and CST carried out the following engineering activities:

- Starting and brake torque diagram for selection of the new electric drive
- Torsional study of the complete train
- Selection of the coupling and flywheel
- Outline drawing of the modified flywheel and the assembly

**Torsional analysis**

The analysis of the torsional behavior of the complete driver-compressor system, composed of an induction motor coupled to the compressor through a soft coupling (Figure 2), was carried out according to API618 requirements to check coupling and compressor flywheel selection and to ensure the torsional compatibility of the system.

The compressor operated with a running speed in the range of 800 to 1000 rpm.

The first step was to create the equivalent shaft obtained with a "lumped parameters" model (i.e. a definite number of inertial masses connected by massless shaft intervals having appropriate torsional stiffness and damping).
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**Torsional Natural Frequencies**

<table>
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<th>Item</th>
<th>cpm</th>
<th>Hz</th>
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<tr>
<td>1st</td>
<td>142</td>
<td>2.37</td>
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<tr>
<td>2nd</td>
<td>944</td>
<td>15.73</td>
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<tr>
<td>3rd</td>
<td>7640</td>
<td>127.34</td>
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**Figure 3 Mode Shapes (Normalized Angular Displacement)**

**Campbell diagram**

Due to the wide speed range, several potential critical speeds were found within 10% of the operating shaft speed and 5% of any multiple of the operating shaft speed in the rotating system (Figure 4). As shown by the diagram above, the three calculated torsional natural frequencies were outside the first and second multiples of the electrical power frequency +/- 5% (necessary condition for the driver to avoid resonance). However, some speed values met one of the torsional natural frequencies within the operating speed range limits. These critical speeds are summarized in the following table:

According to API 618 5th Ed, the steady-state analysis under load had to be carried out, to evaluate and verify that no resonances could occur and that the oscillation amplitudes were below the maximum allowable values.

**Equivalent shaft**

The compressor shaft was modeled with six inertial masses, corresponding to the six cranks.

The flywheel was modeled with one inertial mass, rigidly connected to the compressor shaft end and the first coupling inertial mass.

The selected soft coupling was modeled with two inertial masses, rigidly connected respectively to the compressor flywheel and the electric motor hub and connected by an elastic section.

The electric driver rotor was modeled with one inertial mass connected to the previous sections by an elastic segment.

The coupling damping effect was considered as per manufacturer data. For all the remaining equivalent shaft sections, a viscous damping effect was considered, proportional to the stiffness and with an appropriate amplification factor.

**Calculation of torsional natural frequencies.**

The second step was to calculate the torsional natural frequencies and the relevant mode shapes.

Since the maximum speed to be considered for the compressor was 1000 rpm, only the first three torsional natural frequencies were analyzed. Figure 3 shows the first three mode shapes:

- Mode no. 1 and Mode no. 2 revealed that the first and second torsional natural frequencies strongly depended on the coupling.
- Mode no. 3 revealed that the third torsional natural frequencies depended on the compressor cranks and the coupling.
The compressor torsional response due to gas and inertia forces was therefore evaluated in the frequency domain. The mean and alternating torques in each shaft interval and the angular oscillations in each node were calculated in the speed range between 720 rpm (minimum running speed −10%) and 1100 rpm (max running speed +10%).

**Results**
The results of the analysis were obtained in terms of oscillating torque, angle and degree of irregularity. In particular, the mechanical response analysis showed that the torques and irregularity degree were within the maximum allowable value, confirming the correct choice of the coupling and the dimensions of the flywheel that were estimated during the basic engineering phase to prepare the purchase specifications:

- the diagram of the resulting torques
- the diagram of the most severe degree of irregularity showed that the flywheel inertia was enough to keep the speed oscillations below the maximum recommended values.

The above analyses were given high priority since the technical specifications for

<table>
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<tr>
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<td>3rd TNF</td>
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<td>3rd TNF</td>
</tr>
<tr>
<td>720</td>
<td>90% min</td>
<td>none</td>
</tr>
</tbody>
</table>

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the new driver, the coupling and the flywheel depended on the results of these analyses and the purchase of such important components could not be delayed.

The timeline to run the unit was, in fact, critical on account of November gas contracts and winter needs. The new flywheel was manufactured locally and had to be accommodated with the coupling in two weeks. The entire fix was executed and installed in one month, along with installation instructions and a new recommended service checklist.

**Conclusions**

The substitution of an engine drive with an electric motor drive with VFD is common in the industry for many reasons, including the need to update long-running fleets or to increase their operational flexibility. The implications of this exchange cannot be free from in-depth considerations on the torsional analysis of the train, transmission of torques and dampening with a flywheel.

A detailed review must be completed to obtain the equivalent shaft, and this is more challenging in machines that have seen many years of operation in various process conditions with modifications and inconsistent maintenance philosophies. The study must consider various working conditions and speed ranges. It requires the analysis of torsional natural frequencies and critical frequencies, and this goes beyond the standard sizing of componentry. Therefore, a correct torsional analysis is a critical evaluation necessary for a useful long-term solution.

All this must be done often considering the limited time frame for intervention, due to the production program of the plant that is not a research center but an economic enterprise that must fulfill its functions.

Blackstone and Compressor Service Technology (CST) worked together to evaluate, design and draw up specifications and manufacturing drawings and implement a strategy for the customer for a full train solution. The successful provision of these services proves that maintenance has many levels and even the biggest companies in the world need help to deliver solutions that do not just stop at the shaft end.

Thanks to detailed engineering and to the skills and expertise of the millwrights who worked on the machine, the new train was successfully set up according to schedule and, up to now, it has been operational for almost three years without any unexpected issue.

**FIGURE 7** Alternating torques and relevant mean values for each node.