Effect of Vibration on Electric-Submersible-Pump Failures

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Summary. Electric submersible pumps (ESP's) historically have had short run lives. Their failures usually were unexplained and accepted as the norm. Because the equipment has generally been downhole, finding all the influences that shorten equipment life expectancy has been difficult. Horizontal surface installations for water injection have been made where machine operating conditions could be monitored and equipment performance analyzed.

Unbalanced pump vibration has been identified as the cause of many seal problems and ultimately motor and pump failures. In fact, excessive vibration appears to be an inherent mechanical characteristic of ESP's. To illustrate the severity of this problem, this paper presents data gathered from field installations with monitoring equipment, from equipment tear-down analysis, and from statistics on ESP failure modes.

Introduction

ESP's have been used for more than 50 years, typically for moving large volumes of fluid where other means of artificial lift have not been feasible. Recently, the application of these pumps has expanded. It is not uncommon to use these pumps to move 500 bbl [80 m^3] of fluid from less than 2,000 ft [600 m]. The industry has made great strides in improving ESP performance. Run times have improved significantly over those existing only 10 years ago. These pumps are now economically competitive with other pumping systems in many environments.[1]

A unique application of ESP technology has been in water-injection systems. Water-injection pumps often must move large volumes at high pressures. Not surprisingly, ESP's fit this criterion. Although ESP's are not a panacea for all pumping applications, these examples demonstrate the applicability of ESP technology to an increasing number of applications.

Nevertheless, the significant problems that still exist with the technology must be addressed before the equipment can achieve adequately long run times for it to be widely applied. Current estimates are that ESP's are used in only about 5% of the lift applications in the petroleum industry. This leaves considerable room for further application.

History and Design

Before the problems with ESP's are addressed, the construction of the pumping system should be analyzed. This analysis will demonstrate the problems to be solved.

The design of today's ESP's remains basically unchanged from the first pumps. The pump assembly is a slender centrifugal pump of up to hundreds of stages. The stages consist of an impeller that "floats" on a shaft. The impeller is mated with a diffuser for directing the flow between stages. The impeller/diffuser assembly is stacked inside a small-diameter tubular.

The impellers are usually made of a material that has been sand cast. The vane openings are very small, so machining is done only on the outer surface. Because so many stages are used and the impellers are not fixed to the shaft, there is considerable distance between shaft supports.

The pump is coupled to a seal-and-isolation assembly. The pump shaft is coupled to the isolation-chamber shaft. Two major functions of this chamber are to support any thrust from the pump and to prevent well fluid from entering the motor. Because of the severe service, this assembly is prone to failure, with catastrophic results.

Various attempts to build an adequate sealing assembly have met with less-than-phenomenal success. The major problem is preventing fluid migration around a moving shaft while allowing for expansion of the fluid in the assembly. The first efforts used a grease-packed housing with the shaft running through the grease. The next major designs used a manometer-type isolation section with a fluid heavier than water to prevent water migration. In addition, a Crane-type mechanical seal, usually made of a ceramic face with a carbon runner, was placed around the shaft. Later efforts involved multiple manometer-type isolation sections without the heavy fluid. Current technology uses an elastomeric bag and one or two manometer-type sections. All these later designs continue to use the Crane-type seal around the shaft. Occasionally, different materials are used for the seal surfaces, but the majority still use ceramic with carbon.

The isolation chamber is coupled to the motor, and its shaft is coupled to the motor shaft. The motor consists of multiple sections that are essentially small motors in the same housing and on the same shaft. The motor-housing diameter is very small compared with conventional induction motors. Each motor section, with a 5-in. [12.7-cm] diameter and 18-in. [46-cm] length, has a rating of about 10 hp [7.5 kW]. Therefore, 10 to 20 of these sections are often stacked together to provide the required horsepower.

The motor insulating fluid is open to the isolation chamber fluid. Any problems encountered in the isolation chamber are eventually transmitted directly to the motor and will ultimately result in an electrical failure. Many motor failures can be attributed, directly or indirectly, to a failure in

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TABLE 1—FAILURE MODES AND FREQUENCY

Failure Mode	Frequency
Worn pump	27
Protector-fluid influx	13
Motor-fluid influx	10
Motor winding/overheating	10
Protector mechanical seal	9
Protector thrust bearing	9
Seized pumps	7
Separator failure	7
Motor torque	5
Motor thrust bearing	4

TABLE 2—HORIZONTAL-COMPONENT FAILURES DURING PERIOD OF OBSERVATION

Pump	Period (months)	Motor Bearings	Thrust Assembly
SSU1	18	1	2
SSU2	18	1	2
SSU4	30	3	3
SSU5	30	3	3
HSU1	18	5	1
HSU2	18	4	1
HSU3	18	6	—
HSU4	18	5	—

the isolation section. The original cause of failure may not be the isolation section, but it is eventually involved because it is the coupling and most vulnerable item.

Failures

ESP's historically have short run times. Even discounting the infantile failures, run times are still relatively short compared with the life of a project. Industry average run times have improved from 300 days a few years ago to more than 600 days now. Some aggressive users in reasonable environments have seen average run times in excess of 4 years. The best-performing systems have extended run lives of 7 to 10 years. But unfortunately, in some environments, system life expectancy is only 30 days.

What are the differences in these performances? Obviously, some concern the environment, but it appears that there is a much more subtle problem.

Jacobs[2] presented one of the better correlations of ESP failure modes. Although based on the very harsh environments of the North Sea, the information compares statistically to other experiences. **Table 1** gives a regrouping of the failure modes of interest to this analysis.

From observing similar failure analyses, we can expect that a significant part of the motor-fluid influx problems and many of the other motor problems result from problems in the isolation chamber. For example, if fluid exists in both assemblies, the fluid probably migrated through the protector. Similarly, a failure in the isolation-chamber thrust system would transmit to the motor. In general, motors have relatively few problems if the isolation system maintains its integrity.

Because of similar statistical comparisons, the isolation system has received its share of blame for downhole pump system failures, but most of the failures occur in the pumps. These failures have traditionally been attributed to well-fluid conditions like sand or to other environmental factors.

Because of the pump's location downhole, gathering information about the operating performance and conditions of the pump has been very difficult. With the advent of the horizontal pumps on the surface, additional information that can be extrapolated to downhole conditions can be obtained. The significant feature is that the fluid through the pump is controlled in terms of volume, composition, and temperature, while the external environment is much cleaner than a wellbore.

A study we conducted of the failure modes experienced on horizontal centrifugal pumps in operation since 1982. This represents a broad spectrum of fluid volumes and horsepower ratings. The predominant recent failure modes are listed in **Table 2.** Failure data for six pumps were gathered during an 18-month period; data for two additional pumps were available over a 30-month period. The number of failures in each period is noted.

The remainder of this paper addresses the hypothesis that many of the isolation-chamber, motor, and pump failures occur as a result of pump vibration from sources other than well fluids. If true, then the wrong problem has traditionally been solved. Correcting, mitigating, or designing to tolerate the vibration problem should dramatically improve run times and reduce failures.

Horizontal Installations

The first surface configuration of an ESP was installed in May 1981.[3] This was a 250-hp [186.5-kW] unit designed to pump 4,400 B/D at 2,250 psig [700 m^3/d at 15.5 MPa]. The pump ran about 5 years, but the thrust chamber had to be replaced periodically and the conventional induction motor experienced several failures. The second surface installation was made in another field in Aug. 1981. Four thrust-chamber failures experienced in the next 2 years were attributed to normal problems in the development of a new concept.

The next installation was a replacement for a failed 600-hp [447-kW] positive-displacement pump. This system had a 600-hp [447-kW] motor coupled to an 8,400-B/D [1335-m^3/d] pump at 2,250 psig [15.5 MPa] pump. Catastrophic failures with this installation were experienced from the beginning. The thrust chamber was replaced 10 times in 3 years; the pump was replaced three times and the motor was extensively overhauled. Because of these problems, future installations were restricted to 300 hp [224 kW]. This was the first ESP where vibration obviously was a serious and measurable problem.

Several other surface installations were made in other locations, and other companies also began using the technology. Although thrust chambers and motor bearings failed after very short run times, the

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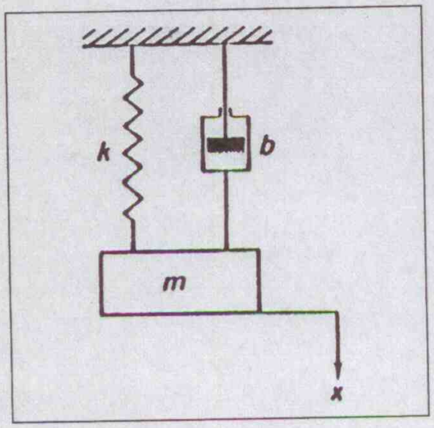

Fig. 1—General vibration model.[4]

TABLE 3—HORIZONTAL CENTRIFUGAL PUMP DESCRIPTION WITH AVERAGE INJECTION-SYSTEM CONDITIONS

Pump	Type	Stage	Length (ft)	Motor (hp)	Discharge (psig)	Volume (B/D)
SSU1	E127	149	40	300	2,250	5,000
SSU2	A177	67	32	600	2,250	10,000
SSU4	E127	149	40	300	2,250	5,000
SSU5	E127	149	40	300	2,250	5,000
HSU1	E127	149	40	300	2,250	5,000
HSU2	E127	149	40	300	2,250	5,000
HSU3	E127	149	40	300	2,250	5,000
HSU4	E127	149	40	300	2,250	5,000
NEF1	A177	26	15	200	1,000	7,000
NEF2	A177	26	15	200	1,000	7,000

"The operating performance of each pump has been monitored since installation. Two major failure modes were revealed: premature motor-bearing failure and frequent seal-assembly failure."

economics continued to favor these pumps. One of the later major installations was a new waterflood plant where the only pumps were four surface ESP's operating at 300 hp [224 kW]. Other pumps have been installed, but only the currently operating pumps are described in **Table 3.** The pressures and volumes are system averages.

The pumps typically are mounted horizontally on an I-beam with various support configurations. The motor is a conventional NEMA-type surface motor. The motor shaft is coupled to the thrust-chamber shaft with a metal spline or a flexible coupling. The earth for the Sims Sand Unit (SSU) pumps is very compacted from long use as a plant facility. The foundation at Northeast Fitts (NEF) was originally somewhat soft, so large concrete supports were poured under the pumps. The foundation at the Humphrey Sand Unit (HSU) was gravel fill typical of new plant locations.

Horizontal-Pump Failure Analysis

Because horizontal pumps are installed on the surface, their failures are much easier to investigate than conventional downhole pumps, where the primary cause of the failure may be masked by other conditions. When a problem occurs, it can be evaluated independently before other failures happen.

The conventional air-cooled motors use ball bearings on either end of the shaft for mechanical support. The bearings ride in a lubricant-packed race. The motor-bearing failures on the horizontal pumps have been caused by a loss of bearing lubricant in a very short period of time. This loss allowed metal-to-metal contact between the race and ball bearing. The friction precipitated a tremendous amount of heat and excessive cyclic loading, which led to catastrophic failure. Vibration is recognized as the cause of this failure type. The excessive shaft movement during vibration allows the loss of bearing lubrication.

The thrust chamber uses a Crane-type mechanical seal to isolate fluids in the pump housing from the thrust bearings. The most common failure mechanism in the thrust chamber is a chip or crack in the ceramic ring of the seal. The crack permits pumped fluid to contaminate the lubricant-filled environment of the thrust and journal bearings. The loss of lubrication causes excessive bearing wear, which results in uncontrolled shaft movement. Bearing failure from continued operation causes excessive vibration and motor or pump failure.

Downhole-Pump Failure Analysis

When an ESP system fails after a short run time, a failure analysis is often performed. From participating in many of these teardown analyses on pumps for all the manufacturers, from different producing companies, and in different environments, we have noted some common failure modes. The effects of excessive vibration can be observed visually on the exterior of the equipment before it is opened. When a worn area on the housing of the ESP that matches the ribs on the armor of the cable is seen, the isolation chamber usually has failed and the motor has experienced a failure.

Observing a problem is only the first step. The next obvious task is to try to determine the cause of the vibration and then to correct the problem. Many possible scenarios can be investigated. If the pump had excessive wear, well fluids containing sand are often credited with causing the problem when nothing else is apparent. If the area has no sand or scale and no evidence of other conditions exists, all too often the failure report would give several possibilities, would state that the cause is unknown, or, worse, would indicate that it is just a problem well.

Experience gathered from surface installations and failure analyses of problems with these systems indicate that vibration and the resulting ESP failures are caused by imbalance from the pump stages and from minor deflections of its shaft.

Vibration

Vibration is the oscillating motion of a machine from its position of rest. The general model used for vibrational analysis is a second-order mass(M)-spring(B)-damper(K) system (**Fig. 1**). The Laplace transform equation for this system is

$$(Ms^2+Bs+K)L(s)=F(s),$$

where L represents displacement and F is the total force on the system.

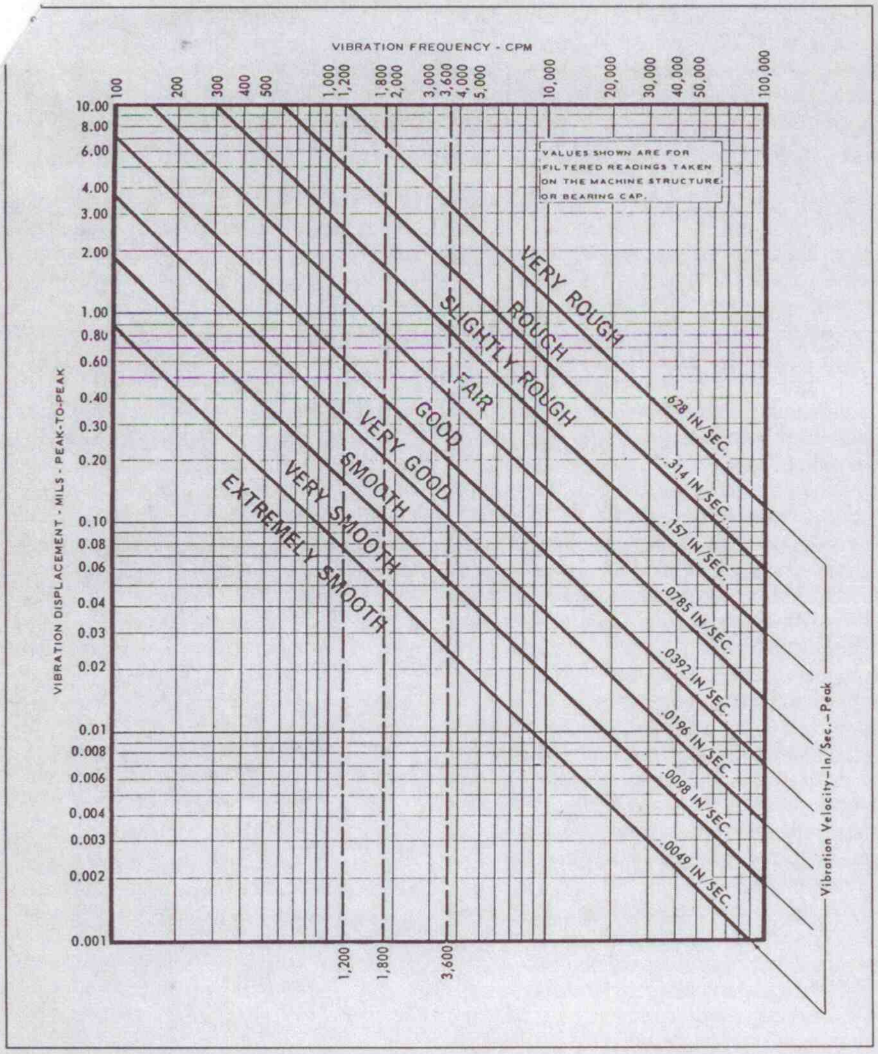

Fig. 2—General machinery vibration severity chart per IRD Mechanalysis. [5]

If a forcing function is applied to the mass, it will move. Without adequate damping, the system will experience sinusoidal oscillations or vibration. The undamped natural frequency of vibration is determined by the ratio of the spring component to the mass component:

$$F_n=(K/M)^{0.5}/2\pi.$$

Resonance occurs when the frequency resulting from the forcing function is the same as the natural frequency. The displacement, $L(s)$, becomes very large at this frequency. If the amplitude is too great, catastrophic failure will occur. Therefore, operating the equipment near its natural frequency should be avoided. Alternatively, adequate damping can be applied or the design must be modified to change the mass or the spring constant.

A guide accepted by the electric motor industry for vibrational evaluation is shown in **Fig. 2**. Because a conventional motor was used for the horizontal application, this guide is appropriate. Similar results would have been achieved with other standards.[6]

Fig. 2 correlates the effects of vibrational displacements at various machine operating speeds. For a 3,600-rev/min machine, such as an ESP, this chart indicates that good performance exists with displacement amplitudes less than 0.4 mil [0.1 μm]. Rough operation begins above 0.8 mil [0.2 μm], and impending failure above 3 mils [0.76 μm].

Exciting forces that induce vibration have numerous causes. The more common ones are manufacturing defects like bent shafts, bad bearings, and unbalanced rotating parts; installation problems like misalignment; and external effects like electromagnetic fields and hydraulic forces. When these exciting forces are imposed on a shaft, the effect is transmitted down the shaft to metal-to-metal contacts, such as bearings and seal assemblies. The incremental-force amplitudes are then added to the natural-frequency amplitudes of the machine. As the peak amplitudes exceed the cyclic fatigue design ratings of any of the machine components, the machine fails more quickly.

Field Data

The operating performance of each pump has been monitored since installation. Two major failure modes were revealed: premature motor-bearing failure and frequent seal-assembly failure. Table 2 summarizes these failures for each pump during its period of observation. Analyses of the motor bearings and the seal assembly indicate that vibration caused these failures. Three different tests

TABLE 4—VIBRATION DATA

Pump	Pump Shaft (mils)	Motor Shaft (mils)
SSU1	5.0	2.1
SSU2	6.5	3.0
SSU5	>10	1.2
HSU1*	>10	0.6
HSU2	>10	5.0
HSU3*	6.0	5.0
HSU4*	6.0	5.0
NEF1	>10	1.0
NEF2	>10	1.0

*After foundation reinforcement.

were conducted on the horizontal pumps and their motors to determine the source and level of the vibration.

First, the motors were run with the pump disconnected. The vibration amplitude and frequency were measured for the motor shaft with a mechanical analyzer and a connected shaft stick. The amplitude of vibration was no greater than 0.25 mil [0.06 μm] for any of the motors. Because this is well within the acceptable vibration tolerances, we concluded that the motors were not the source of vibration.

Second, the entire pump assembly, with the motor disconnected, was tested to determine the system's natural frequency of vibration. With the analyzer connected to the pump, an impulse (blow from a hammer) was applied to the pump housing while the pump was not running. The unit vibrated freely at or near 3,600 rev/min for more than 5 seconds. This indicates that resonance was occurring at operating speed.

Third, the analyzer was used to measure motor and pump-shaft vibration while the machine was running. The results, measured with a shaft stick, are given in **Table 4.** The abnormally high pump-shaft vibration, coupled with the first test, indicates that the pump shaft was the source of vibration.

Several modifications were made to eliminate resonance from the system. A concrete foundation was poured beneath each pump. The steel C-face coupling that connected each motor housing with each pump housing was removed. A rigid spline-type coupling between the pump and motor shafts was replaced with a flexible coupling. These modifications changed the spring constant and damping coefficient of the pump assembly. This also totally isolated the electric motor from the pump. Motor run times immediately increased from 2 months to more than 1 year. However, this is still very short for motors that often run longer than 20 years.

We have demonstrated that the pump shaft of ESP's causes pump vibration. Only the motor life has been improved by isolating the motors from the pumps and by damping the resonance. The pump-shaft vibration still exists.

Correlation of Horizontal and Downhole Pumps

Horizontal and downhole centrifugal pumps have very similar failure modes. Although

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the applications are different, the accessibility to horizontal pumps may well provide a significant breakthrough in the correction of downhole problems. From downhole and horizontal failure analyses, it is apparent that vibration contributes dramatically to the short run life of submersible systems.

Two factors appear to contribute to the actual cause of shaft vibration. First, the method of manufacturing and assembling these pumps causes inherent imbalance. Because of the long shaft lengths, stiffening between stages is inadequate. This factor may be corrected by dynamically balancing individual machine parts and redesigning the pump to include more interstage stiffening. The second, more pernicious factor lies in the way the machines are installed.

We tend to view ESP's as either horizontally "level" or vertically "plumb." The only difference is that the two installations are rotated 90° from each other in a single plane. In fact, the horizontal pump must cope with the effects of gravity along its length. Gravity tends to force the rotating elements to deflect and also causes the slender tube that is the pump casing to deflect. The rotating elements are constrained by the pump casing, but the casing is constrained only by the supports on which the pump rests. Whether or not the pump is level depends on the method of field installation. Field measurements with a surveyor's transit have indicated that a long, horizontal submersible may be out of level as much as 0.5 in. [1.27 cm] either way over the pump length. A deviation from shaft concentricity of only 0.0025 in./ft [0.0208 cm/m] can cause enough vibration to shorten bearing lives from years to months.[5] The undulating installation of a horizontal ESP will certainly cause at least that much vibration.

When the pumps are put in the vertical configuration, all such gravity problems supposedly go away. In fact, few wellbores are truly vertical. The individual deviation of each wellbore has a dramatic impact on the vibration levels in each ESP. The location of a downhole ESP may even cause the pump to have a slight bow, which would compound any gravitational effects on vibration. If a pump that was perfectly balanced on a test stand on the surface were put into a deviated well, the vibration level of the pump would increase markedly. One cause would be the change in shaft concentricity, as indicated previously; Brinner *et al.*[7] noted other causes.

In short, the very attribute that makes ESP's so useful to the petroleum industry—their extreme flexibility and ability to fit into well casings—makes them inherently unstable in terms of vibration. We in the industry have assumed that the need for lateral stiffening of the pumps and bearing assemblies is eliminated when the pumps are used in the vertical position. We have been mistaken.

Later designs of horizontal pumps have added radial support bearings in the thrust chamber. These supports have increased the run time for the chambers. Realization that this support is necessary implies a continued weakness in the conventional vertical installations without this radial support.

Another design change made by one manufacturer is to replace the ceramic/carbon Crane-type mechanical seals with a hardened material. Our failure analysis indicates a significant improvement in seal life. Because a seal is seldom damaged by operation in the protector, this replacement will ultimately improve run life.

Conclusions

ESP's are very viable tools for many applications requiring the pumping of fluids. With all their merits, however, they have been plagued with a short run life. In many instances, the cause of the failure has been glossed over. Our field measurements and observations indicated the following conclusions.

1. Vibration of ESP's is significantly higher than is generally acceptable for other industrial equipment.

2. Vibration of the system is inherent in its design, manufacture, and application. In fact, the built-in flexibility of these machines that makes them useful in pumping wellbore fluids and fitting in wellbores demands that a higher level of vibration be tolerated than in other industrial equipment.

3. As published data have shown, a large percentage of downhole ESP failures may be attributed to failures in the seal assembly and thrust chamber. It appears that the major contributing cause of these failures is the inherent pump vibration and the inability of the seal assembly and bearings to withstand high radial vibration.

4. The method by which run lives of the system may be extended lies in redesigning the thrust chamber to incorporate radial bearing surfaces that are more resistant to vibration than the sleeve bearings commonly used and in the selection of a material for the seal face that provides both wear resistance and reduced brittleness.

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SI Metric Conversion Factors

bbl	× 1.589 873	E−01	= m^3
ft	× 3.048*	E−01	= m
hp	× 7.460 43	E−01	= kW
in.	× 2.54*	E+00	= cm
mils	× 2.54*	E−05	= m
psi	× 6.894 757	E+00	= kPa

*Conversion factor is exact.

Provenance

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