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EPRI AP-5600
Project 2559-1
Final Report
January 1988

Downwell Pump Reliability: Geothermal Experience Update

Prepared by
Radian Corporation
Austin, Texas

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REPORT SUMMARY

SUBJECT	Geothermal systems
TOPICS	Pumps Failure analysis Reliability Availability Geothermal wells
AUDIENCE	Generation managers / R&D engineers

Downwell Pump Reliability: Geothermal Experience Update

Improved downwell brine pumps will enhance the development of moderate- to high-temperature geothermal resources and ensure high system availability. This survey updates recent technology advances aimed at increasing operating lifetimes, understanding the causes of pump failures, and improving pump performance.

BACKGROUND Geothermal resources with temperatures between 250–360°F (121–182°C) represent prime candidates for binary-cycle power production. The successful exploitation of these resources requires reliable downwell brine pumps. Although early attempts at pumping such moderate- to high-temperature geothermal fluids revealed severe difficulties, subsequent R&D efforts by pump manufacturers resolved many problems. EPRI report AP-3572 presents results of a 1983 survey of downwell pump experience under production conditions at the Magma Power Company Magma power plant, the Mammoth-Pacific Power Company Mammoth power plant, and the San Diego Gas and Electric Company 50-MW geothermal binary demonstration power plant.

OBJECTIVES To update data on downwell pump installation, performance, and failure at commercial geothermal facilities; to determine pump failure causes; and to evaluate progress toward increasing pump service life and reducing maintenance requirements.

APPROACH The project team surveyed power plant and well-field operators to gather information on their experience with downwell geothermal pumps. They queried plant operators about pump history, maintenance, modifications, brine chemistry environment, and new practices. In addition, the team conducted a literature search of published information on the reliability of moderate- to high-temperature downwell pumps. A database expanded from AP-3572 defined the chemical conditions under which high-temperature downwell production pumps may operate. The team used this data, as well as estimated near-term, mean-time-to-failure (MTTF) rates, to compile recent MTTF statistics and to compare causes of pump failures.

RESULTS Survey responses indicated that lineshaft pumps account for virtually all moderate- to high-temperature geothermal pumping in the United States. The industry has continued development of lineshaft pumping systems, resolving the problems that caused early failures. Operating lifetimes have increased from as little as 1000 h to more than 19,000 h for one unit in the ensuing period. Increases in projected near-term MTTF, summarized graphically in the report, resulted from improvements in pump and bearing design, as well as improved installation and maintenance procedures. However, R&D of both electric submersible pumps and close-coupled hydraulic turbopumps have fallen behind development of the lineshaft system.

EPRI PERSPECTIVE

Recent advances in downwell pump technology should help utility personnel analyze and reduce the economic and environmental risks in future commercial ventures using geothermal downwell pumps. This report provides data on pump design, performance, and operation. The documentation of power plant operating experience and the brine chemistry environment of geothermal pumps will allow for more-informed decision making regarding heat supply availability and reliability. Electric submersible pumps and close-coupled hydraulic turbopumps will require additional development before they find a niche in the geothermal marketplace.

PROJECT

RP2559-1
EPRI Project Manager: J. L. Berning
Advanced Power Systems Division
Contractor: Radian Corporation

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Research Project 2559-1

Final Report, January 1988

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Prepared by
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Austin, Texas

ABSTRACT

Geothermal resources with temperatures between 250° and 360°F (121° and 182°C) are prime candidates for binary-cycle power generation, and constitute about 80% of the power-capable resources in the United States. The successful exploitation of these resources requires reliable high-capacity downwell brine production pumps, but earlier experience showed that high-capacity, high-temperature geothermal production pumps had many problems which resulted in a mean time-to-failure (MTTF) of less than 1000 hours. However, steady progress has been made since 1981, and a large body of experience has been acquired by three geothermal binary plants. This survey of high-temperature geothermal downwell pump users and manufacturers updates a prior survey (AP-3572) completed in early 1983.

This survey traces the development of lineshaft pump technology from the late 1970s to the present (mid-1987), detailing the advances in design, materials selection, and operating practices. Case histories of 72 lineshaft pumps installed at three geothermal binary plants since late 1981 are documented, including some detailed cause of failure reports. In the recent past, pump lives in excess of 7000 hours have become common, but a high continuing rate of premature failures resulted in a mean time-to-failure (MTTF) of about 5000 hours. Based on recent advances which appear likely to eliminate most premature failures, the estimated near-term MTTF will be on the order of 8000 hours.

The survey found almost no development of high-temperature geothermal electric submersible pumps (ESP's) or close-coupled downwell hydraulic turbopumps, and concluded that considerable development and demonstration will be needed before these technologies are able to compete with existing high-temperature geothermal lineshaft pump technology.

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Geothermal resources with temperatures between 250° and 360°F (121° and 182°C) are prime candidates for binary cycle power production. About 80 percent of the identified resources hotter than 250°F (121°C) fall into this temperature range. The successful expolitation of these resources requires reliable downwell brine production pumps. Though higher temperature resources can be economically exploited by natural flow, pumping of resources up to about 420°F may be desirable to reduce field development costs.

Early attempts to pump geothermal resources hotter than 250°F (12°C) encountered severe difficulties, creating uncertainty about the resource producers ability to provide a reliable source of energy to a power plant. In response, EPRI sponsored a state-of-the-art assessment of high-temperature geothermal production technology. That assessment, completed by Radian Corporation in April 1983, was published by EPRI as report AP-3572 in June of 1984. It indicated that the average life of about 50 lineshaft and electric submersible pumps tested at that time in high-temperature wells was only about 1000 rotating hours (all reference to hours refers to rotating or operating hours), with lineshaft pumps giving the more consistent performance. On the other hand, that survey indicated that some promising advances had been made near the end of the survey period, advances which indicated that pump lives of more than 4000 hours were in the offing for both lineshaft and electric submersible pumps.

Since completion of the initial survey, considerable downwell pump operating experience, virtually all with lineshaft pumps, has been gained at three binary power plants in the western United States. This operating experience showed a marked increase in the run times of some pumps, with run times greater than 10,000 rotating hours being reported. These increased operating lifetimes generally resulted form the brine production operators working directly with the pump suppliers. Yet, premature failures, including "zero" runs, have continued to plague operations. Thus, uncertainty about the reliability of high volume geothermal downwell production pumps has remained an obstacle to the exploitation of geothermal binary-cycle power generation in the United States. In response to the utility industry's need for current, accurate information on the status of downwell pumps, EPRI commissioned Radian to update the prior survey.

OBJECTIVES

The objectives of this project were to:

- Document recent data on downwell pump installations, performance, and failures at commercial plants and demonstration facilities in the United States;
- Determine, where possible, the root causes of the failures; and
- Evaluate the progress made in increasing the operating lifetime and reducing maintenance requirements of the downwell pumps.

APPROACH

Radian updated and documented experiences with geothermal downwell pumps at Magma Power Company's Magma Power Plant at East Mesa, California; Mammoth-Pacific Power Company's Mammoth Power Plant at Mammoth Lakes, California; and the 50 MW Geothermal Binary Demonstration Power Plant at Heber, California through extensive interviews with cognizant personnel and visits to the sites.

One of the major pump manufacturers was interviewed at their plant site, while the other organizations active at the time of the previous survey were interviewed by telephone.

Once the updated experiences had been gathered, they were statistically evaluated for trends. A method was developed for estimating the remaining life of running pumps in a mixed population of running and failed pumps. These estimates were then used to estimate recent mean time-to-fail (MTTF) statistics, as well as projected near-term MTTF estimates. These statistics, in conjunction with updated estimates of the time to pull and replace a downwell pump were then used to estimate the effects of recent developments on the brine supply of a hypothetical geothermal binary power plant.

RESULTS: THE DOWNWELL PUMP ENVIRONMENT

A wide range of downwell conditions exist in the geothermal resources which are the most likely candidates for downwell pumping. Temperatures may range from approximately 280°F (138°C) up to more than 400°F (204°C). However, approximately 80% of the potential applications have resource temperatures of less than 365°F (185°C), and the resources likely to be of greatest interest in the near term will probably have production temperatures in the range of 330° to 365°F (165° to 185°C).

The chemistry of geothermal resources varies widely. The total key corrosive species (TKS), a first-order indicator of corrosivity, may vary from less than 500 ppm to more than 100,000 ppm. However, the resources pumped to date, and those likely to be pumped in the near term, all fall into Class IV of the Geothermal Corrosivity Classification System developed and published by Radian. The corrosivity of this class of brines is relatively mild to the materials commonly used in geothermal pumps, and corrosion per se should be a relatively minor problem. This conclusion is supported by the finding of this study that prolonged standby in a well has little effect on ultimate pump system life, provided that steps are taken to protect the lineshaft from brine intrusion.

Sand is a fact of life in many geothermal wells. It may come from the formation or result from well workover operations. High sand production can severely damage downwell pumps in several ways. If there is enough sand, it can simply clog the pump, or become so wedged between the impeller skirt and bowl throat that the pump "locks up" and won't turn. Sand produces severe erosion of some pump bowl geometries and is particularly damaging to the bearing technologies used in earlier pumps.

Calcium carbonate scale will rapidly plug pumps if the drawdown is sufficient to allow gas breakout in the pump suction. Maintenance of sufficient suction pressure will prevent this form of scaling. Production of brine from multiple zones in a formation can produce scaling problems which cannot be controlled by this method.

RESULTS: LINESHAFT PUMPS

At the close of the prior survey the mean life of high-temperature lineshaft pumps was less than 1000 hours, and most failures were attributed to lineshaft, rather than pump, problems. By late 1981, it was clear that water-lubricated lineshafts could not provide adequate service, that the existing oil-lubricated lineshafts had severe but surmountable problems, and that pump problems would become life-limiting when the immediate lineshaft problems were overcome.

Technology developments. Since the previous survey, virtually all high-temperature geothermal downwell pumping in the United States has been by lineshaft pumps. Since late 1981, several distinct improvements in lineshaft pump technology have occurred, including:

- Improvement of oil-lubricated lineshaft design and installation practices;
- Modifications of bowl geometry to minimize sand damage;
- Use of mating wear-resistant impeller skirt and bowl throat wear rings; and
- Introduction, in late 1983, of metallurgically hardened steel pump bearings which are much more resistant to sand than the prior bronze bearing technology.

The last innovation listed above was of particular importance because repeated failure analyses had shown that sand damage to the pump bearings was the root cause of many pump failures.

Recently applied practices. These technological improvements, along with the generally used installation and operating practices, can be defined as "recently applied practices." The longest pump endurance to date under these practices was just over 20,000 hours, while one-fourth of the pumps lasted at least 7100 hours. This demonstrates that current technology is capable of producing pumps that can run for one year under a wide range of resource conditions. However, one-fourth of the pumps lasted less than 1400 hours, and the median life was 4100 hours. This predominance of early failures has kept the mean time-to-failure low. The estimated mean time-to-failure (MTTF) under recently applied practices is about 5000 hours.

Life-extension program. In 1985, Chevron began a pump life-extension program at the Heber Binary Plant. Chevron found that high-temperature geothermal downwell pumps operate with very little margin for error due to the severity of the environment. Thus, imperfections which may be harmless under less severe conditions result in premature failure under high-temperature geothermal conditions. They concluded that the levels of quality control then being applied to high-temperature geothermal lineshaft pumps were not adequate, and implemented pump life-extension practices which include the following elements:

- Enhanced assembly quality control;
- Improved installation practices; and
- Semi-continuous monitoring of pump efficiency allowing detection and remediation of pump problems while damage is minimal.

In at least one case, the last practice alone almost doubled the life of a pump.

The first four pumps to receive the full benefit of these pump life-extension practices were each running without detected problems after more than 4000 hours at the conclusion of this study. It is highly likely that this portends a new breakthrough in lineshaft pump reliability resulting from the elimination of early failures as a major factor in pump life statistics. The estimated near-term MTTF, predicated on the virtual elimination of failures at less than 4000 hours, is on the order of 8000 hours.

Lineshaft problems have been rare in pumps with high endurances, but were a significant contributor to early failures. Realization of the full benefits of the pump life-extension program will also likely require improved quality control of lineshaft components and assembly.

Pump availability and brine supply capacity. Modeling of a hypothetical multi-well (10 well-no spares, 10 well plus 2 spares, and 10 well plus 5 spares) brine supply system showed that the MTTF under recently-applied-practices provides adequate individual pump availability (about 94 percent) as well as adequate brine capacity. This is not surprising since two commercial power plants have been operating for several years using these practices.

The model indicated that the projected improvement in estimated near-term MTTF will have little effect on either individual pump availability or brine supply capacity. However, the economic benefit, reflected in reduced pump and maintenance costs, as well as reduction in the number of spare wells required to maintain a reliable brine supply, will likely be a significant factor in the economic viability of existing and future geothermal binary plants.

RESULTS: ELECTRIC SUBMERSIBLE PUMPS

At the time of the prior survey, several manufacturers of electric submersible pumps had active programs to develop reliable pumps for 365°F (165°C) geothermal service. Developments at that time indicated that such pumps would be available in the near term. However, the cessation of DOE support of pump development, coupled with subsequent economic factors and the perceived small size of the potential market, resulted in the termination of these projects shortly after the conclusion of the prior survey. This was apparently an economic decision rather than an industry consensus that the technology was not viable. The current survey failed to identify any active development work since 1983.

RESULTS: DOWNWELL HYDRAULIC TURBOPUMPS

Close-coupled hydraulic turbopumps (CCTPs) can be set extremely deep, even in highly deviated wells. Their high speed means that high flow rates at high pressure boosts can be generated by smaller diameter pumps, which can even be set at the bottom of wells if necessary. Higher production rates from smaller diameter wells can result in much lower wellfield costs. CCTPs appear to be the only downwell pump technology which is likely to be readily extendable to temperatures above 400°F (204°C). Their "wire to pump" efficiency is only about 35 to 50%, compared to the nominal 70 to 75% associated with lineshaft pumps.

At the close of the prior survey, a Weir, Ltd. CCTP was awaiting trials in the United States. J. Hilbert Anderson, Inc. had bench-tested a prototype CCTP of their own design. The Weir pump was tested repeatedly at East Mesa in 1985 and 1986, with each trial resulting in fairly rapid plugging of the pump by scale. Weir concluded that they were not able to get an adequate setting depth and the testing was concluded. Weir considers the pump to be commercial, and they indicate no further development work is planned. Considerable demonstration effort will be required before any estimate of its high-temperature geothermal reliability will be possible. Lack of funds prevented testing of the J. Hilbert Anderson pump.

CONCLUSIONS

Lineshaft pumps.

- High-temperature, high-capacity, geothermal lineshaft pumps represent a young but demonstrated technology. Under recently applied practices, the MTTF has been about 5000 hours.

- Improved quality control, close monitoring of pump performance, and in-place maintenance intervention before downwell problems become severe are expected to extend the near-term MTTF to the order of 8000 hours. This extension will have little effect on individual pump availability or brine supply capacity, but should significantly improve the economics of large scale pumped brine production.

Electric submersible pumps.

- The experience accumulated in these two surveys indicates that the pieces needed to overcome several serious problems and create a durable electric submersible pump with a ceiling temperature of at least 375^{o}F (190^{o}C) probably do exist, but that a considerable development program may be required to bring them together with sufficient refinement to give consistent high endurances. It appears unlikely that electric submersible pump manufacturers will proceed unilaterally with such development programs. High-temperature geothermal electric submersible pumps with demonstrated reliability comparable to existing lineshaft pump technology do not appear likely in the near term.

Close-coupled hydraulic turbopumps.

- Close-coupled hydraulic turbopumps have certain advantages of inherent ruggedness, flexibility of installation, and equivalent production from smaller-sized units which may offset their relatively lower overall efficiency. This technology is the only one with the near-term potential for use above 400^{o}F (204^{o}C), and one manufacturer has a "commercialized" unit. However, it appears unlikely that manufacturers of these pumps will proceed unilaterally with the testing programs necessary to convincingly demonstrate the technology. Therefore, a high-temperature geothermal close-coupled hydraulic turbopump with demonstrated reliability comparable to existing lineshaft pump technology does not appear likely in the near term.

Section 1

INTRODUCTION

PROJECT RATIONALE

Geothermal resources with temperatures between 250° and 350°F (121° and 182°C) are prime candidates for binary-cycle power production. About 80% of the identified resources hotter than 250° (121°C) fall into this temperature range. The successful exploitation of these resources requires reliable downwell brine production pumps. Though higher temperature resources can be economically exploited by natural flow, pumping of resources up to about 420°F (215°C) may be desirable to reduce field development costs.

Downwell pumps may be installed to achieve any or all of the following advantages:

- Maintenance of brine temperatures at reservoir levels to maximize heat transfer to binary-cycle working fluids;
- Maximum production of geothermal fluid from a given number of wells drilled into a reservoir; and/or
- Prevention of gas break out (evolution) and resulting carbonate scale formation in the well bores or surface equipment.

Early attempts to pump geothermal resources hotter than 250°F (122°C) encountered severe difficulties, creating uncertainty about the resource producers ability to provide a reliable source of energy to a power plant. In response, EPRI sponsored a state-of-the-art assessment (SOTA) of high-temperature geothermal production technology. This assessment, completed in April 1983, was eventually published by EPRI as report AP-3572 in June of 1984.

This initial survey presented a bad news/good news picture of the then-existing status of high-temperature geothermal downwell production pump technology. The survey indicated that the average life of about 50 lineshaft and electric submersible pumps tested at that time in high-temperature wells was only about 1000 rotating hours, with lineshaft pumps giving the more consistent performance. On the other hand, the survey indicated that some promising advances had been made near the end of the survey period, advances which indicated that pump lives of more than six months were in the offing for both lineshaft and electric submersible pumps.

Since the previous survey was published, binary-cycle power plants in commercial demonstration and operation have demonstrated high reliability and availability, and will therefore require downwell pumps with comparable values. Also, the potential for using binary-cycle technology on geothermal resources above 360°F (180°C) is coming closer to reality, and such plants will require even higher temperature pumps than those currently available.

Since the initial survey was completed, considerable downwell pump operating experience has been gained at three larger binary power plants in the western United States. Experience at these plants has shown a marked increase in the run times of some pumps, with a median expectancy of 4000 to 5000 hours. Some pumps have run longer than 15,000 hours. These increased operating lifetimes have generally resulted from the brine production operators working directly with the pump suppliers, leading to changes in many areas, including:

- Lineshaft design and lubrication;
- Pump design;
- Pump bearing technology; and
- Installation, operation, and maintenance procedures.

On the other hand, premature failures, including "zero" runs, have continued to plague operations, a fact evidenced by the relatively low median run times. Very recent developments indicate that the causes of many of these early failures have been eliminated, resulting in a significant improvement in pump life.

Thus, it is important to the geothermal community, electric utilities, and equipment vendors that accurate information on the current status of high-temperature downwell pump technology be available. In response, EPRI has commissioned the current survey.

OBJECTIVES

The objectives of this project are to:

- Collect recent data on downwell pump installations, performance, and failures at commercial and demonstration geothermal plants in the United States;
- Determine, where possible, the root cause of failure of each pump system; and

- Evaluate the progress made in increasing the operating lifetime and in reducing the maintenance of high-temperature geothermal production pumps.

STUDY APPROACH

The bulk of the information incorporated into this report was gathered by direct interview with the operators of the power plants and/or well fields discussed in this report. These individuals provided a wealth of unpublished information, and then reviewed Radian's rendition of their contributions, correcting any inadvertent errors. Additionally, Radian contacted those organizations which had contributed to the initial survey to determine the current status of their activities. Lastly, Radian conducted an automated literature search for published information concerning the reliability of high-temperature downwell pumps. An annotated bibliography generated by this search is included in this report as Appendix C.

REPORT ORGANIZATION

This report is an update companion to the original survey (AP-3572). It recapitulates the key findings of the original survey and can thus be used alone, but it is not an expanded rewrite of the original document. Thus, the reader is referred to AP-3572 for details of the methodology or specific findings of the original survey.

Section 2 of this report presents an updated overview of the operating environments for which high-temperature geothermal pumps need to be designed.

Since the initial survey, virtually all pumping experience has been with lineshaft pumps. Sections 3 through 5 each deal with different aspects of the information relating to geothermal lineshaft pumps. Section 3 presents an overview description of high-temperature, high-capacity geothermal lineshaft pump technology, traces the development of this technology from its infancy to its current young mature status, and describes "typical" installations.

Section 4 reviews the status of lineshaft pumps as revealed by the prior survey, then summarizes the new experiences from Magma Power Company's operations at East Mesa, Mammoth-Pacific Power Company's operations at Mammoth Lake, and Chevron's operations at the Heber Binary Plant. The significant findings from each site are identified separately, followed by a discussion of further insights from examining the data collectively.

Section 5 focuses on near-term expectations. This section presents a statistical analysis for trends in the lineshaft pump experience data, as well as an estimate of pump endurance assuming that the results of the last four years represent the best performance that can be expected in the near term. This recently-applied-practices estimate is therefore a worst-case scenario.

However, recent events indicate that it may be possible to eliminate most of the early failures by enhanced quality control. Therefore, Section 5 also considers the effect on pump endurance expectations of eliminating early failures, yielding a better-case estimate of near-term pump endurance. Lastly, Section 5 considers the time required to pull and replace downwell pumps and the effect that this has on brine availability.

Only brief anecdotal accounts of the few trials of electric submersible and novel technology pumps were available. These accounts are presented in Section 6.

Section 6 addresses on-going pump development projects.

Section 7 discusses current pumps development projects.

Section 8 presents the findings of this survey in brief conclusion form.

Appendix A is a glossary of pump technology terms as used in this report.

Appendix B identifies the contributors to this survey.

Appendix C contains performance curves for the Peerless Pumps used by Chevron at the Heber Binary Plant. Because Johnston considered the curves for the Johnston pumps discussed in this report to be proprietary, those curves are not available.

Appendix D contains the annotated bibliography created by a literature search for published information on the reliability of geothermal pumps.

KEY TERMS

The technical terms used in this report will be defined explicitly or by context as they occur. However, to avoid confusion, it is important to clarify a few terms which will be generally used and which have caused confusion in the past.

Pump and pump system. The pump system is defined as the actual pump unit itself and all of its related downwell and wellhead equipment (excluding, in the case of lineshaft pump systems, the electric motor). However, in the common usage, the term pump may refer to either the actual pump unit or to the entire system. The context of usage should make it clear whether the word "pump" refers to the entire system or just to the actual pump unit.

Failure. In the initial survey, failure was defined as having occurred when a pump system was no longer capable of operation. However, the recent trend is toward preventive intervention, where the pump system is removed and repaired at the first sign is of illness, rather than being run to the actual point of inoperability. Have such pumps failed?

Endurance or run time. Given this trend toward preventive intervention (which greatly increases pump salvage value with minimal impact on brine availability), "endurance" or "run time" is more descriptive of what the plant operator can expect than is a term such as "pump life". In this report, the terms "endurance" and "run time" will be used to indicate the number of rotating hours that may be expected from a given pump. The time starts when a pump is installed in the well and stop when the pump is pulled out of the well and sent to the factory for rebuilding.

The decision to use rotating hours rather than total hours in the well as a measure of endurance was based on the evidence that wear, not static corrosion, dominates the pump degradation process. Making this distinction was necessary because many of the pumps considered in this survey were operated for only a small fraction of time they were in the well, usually because of factors unrelated to the pump systems. While considering only rotating hours may slightly understate the pump endurance, it was clear that using the total time in the well would grossly overstate endurance.

Reliability. The term "reliability" has a precise statistical meaning as well as a more common, strictly qualitative, usage. Unless expressly indicated otherwise, the term "reliability" as used in this report will indicate the latter usage, rather than the more strict statistical meaning.

Section 2

THE DOWNWELL PUMP ENVIRONMENT

As a part of the original survey, Radian compiled a database defining the thermal and corrosive chemical conditions to which high-temperature downwell production pumps may be subjected. This database may be of significance in the selection of existing pumps, as well as in the development of hardened downwell pumps for hotter, more corrosive wells which may be exploited in the future.

CORROSION CHEMISTRY

Based on prior experience with geothermal materials selection (1,2,3), Radian identified seven chemical species, called the "key corrosive species", which appear to dominate the corrosion of most engineering materials in geothermal environments. These species are:

- Oxygen (from atmospheric contamination);
- Hydrogen ion (pH);
- Chloride ion;
- Carbon dioxide species (dissolved carbon dioxide, bicarbonate, and carbonate);
- Hydrogen sulfide species (hydrogen sulfide, bisulfide, and sulfide);
- Ammonia species (ammonia, ammonium ion); and
- Sulfate.

In addition, trace quantities of other ions, such as transition element ions, can have a significant impact on some specific engineering alloys such as aluminum.

Table 2-1 describes the principal effects of the key corrosive species. However, geothermal corrosion is a complex topic that is far beyond the scope of this report. The reader is referred to references 2 and 3 for a detailed discussion.

To characterize the combined effects of the individual key corrosive species and other factors on geothermal materials performance, Radian developed and refined a geothermal corrosivity classification system which divides the world's geothermal

Table 2-1

PRINCIPAL EFFECTS OF THE KEY CORROSIVE SPECIES

Oxygen	• Extremely corrosive to carbon and low alloy steels. 30 ppb shown to cause four-fold increases in carbon steel corrosion rate. • Concentrations above 50 ppb cause serious pitting. • In conjunction with chloride and high temperature, less than 100 ppb dissolved oxygen can cause chloride-stress corrosion cracking (chloride-SCC) of some austenitic stainless steels.
Hydrogen ion (pH)	• Primary cathodic reaction of steel corrosion in air-free brine hydrogen ion reduction. Corrosion rate decreases sharply above pH 8. • Low pH (less than about 5) promotes sulfide stress cracking (SSC) of high strength low alloy (HSLA) steels and some other alloys coupled to steel. • Low pH may cause breakdown of passivity of stainless steels. • Acid attack on cements.
Carbon dioxide species (dissolved carbon dioxide, bicarbonate ion, carbonate ion)	• Dissolved carbon dioxide lowers pH, increasing carbon and HSLA steel corrosion. • Dissolved carbon dioxide provides alternative proton reduction pathway, further exacerbating carbon and HSLA steel corrosion. • May exacerbate SCC.

Table 2-1 (Continued)

PRINCIPAL EFFECTS OF THE KEY CORROSIVE SPECIES

Hydrogen sulfide species (hydrogen sulfide, bisulfide ion, sulfide ion)	• Potent cathodic poison, promoting SCC of HSLA steels and some other alloys coupled to steel. • Highly corrosive to alloys containing both copper and nickel in any proportions. • May cause an active path chloride-sulfide-SCC of nickel-based alloys at high temperature.
Ammonia species (ammonia, ammonium ion)	• Causes SCC of some copper-based alloys.
Chloride ion	• Strong promoter of localized corrosion of carbon, HSLA, and stainless steels as well as of other alloys. • Chloride dependent threshold temperature for pitting and SCC. Different for each alloy. • Little if any effect on SSC. • Steel passivates at high temperatures in pH 5, 6070 ppm chloride solution with carbon dioxide. 133,500 ppm chloride destroys passivity above 150°F.
Sulfate ion	• Primary effect is corrosion of cements.

resources into six classes based on fluid chemistry and corrosiveness to certain engineering alloys (2-4). This system has been clearly useful for above-ground portions of geothermal plants, and there is good reason to believe that it may be applicable to downwell components as well.

One of the most important parameters of this system is the term "Total Key Species or TKS", defined as the mass sum of the key corrosive species, except oxygen, listed above. Some other significant factors are temperature, pH, and fraction of chloride in the TKS.

Table 2-2 alphabetically lists the 30 resources which were included in the original resource chemistry database, assigns a resource number to each, and identifies the references from which the resource characteristics were developed. This table has been updated with new data from Magma Power as East Mesa, Mammoth Power at Mammoth Lakes, and Chevron at the SDG&E binary plant at Heber.

Table 2-3 summarizes the available data on the key corrosive species and estimated pumped resource temperature from the various resources. Most of these values are averages, judged to be representative of typical values for the resource. In a few cases, the data suggest that more than one reservoir is being produced at the same site. In these cases, multiple lines of data are presented for the same site. Table 2-3 also contains a number of notes which indicate how well the analyses are likely to reflect downwell conditions. The reader is directed to these notes for caveats and other supplemental information.

Table 2-4 provides a further analysis of the data in Table 2-3. Analytical values from Table 2-3, which were judged likely to be representative of downwell conditions, were ranked and paired with their corresponding cumulative probability functions (P%), defined as:

$$(P\%) = (100)r_i/(I + 1)$$

where: r = the rank position of the i-th value of a corrosive species; and
I = the number of values in the rank listing.

Table 2-4 was then synthesized by interpolation as needed.

Table 2-4 describes the probability of encountering resources with a given corrosive species at no more than a given concentration. For example, consider chloride.

Table 2-2

IDENTIFICATION OF GEOTHERMAL RESOURCES IN THE DOWNWELL CONDITIONS DATABASE

Resources	Resource Log Number	Reference
Original Survey Data:		
Ahuachapan, El Salvador	1	5, 6, 7
Baca, NM	2	8, 9, 10
Beowawe, NV	3	11
Brady Hot Springs, NV*	4	12, 13
Casa Diablo (Mono-Long or Mammoth), CA*	5	14
Cerro Prieto, Mexico	6	6, 14
Cesano, Italy	7	11
Dunes, CA	8	11
East Mesa: DOE Wells*	9	14, 15, 16
East Mesa: Magma Power Co. Wells*	10	17
East Mesa: RGI Wells*	11	18
El Tatio, Chile	12	6, 11
Heber, CA*	13	11, 14, 19, 20, 21
Hveragerdi, Iceland	14	22
Kamchatka, USSR	15	11
Kawerau, New Zealand (NZ)	16	11
Krisuvik, Iceland	17	11
Namafjall, Iceland	18	7, 22, 23
Ngawha, NZ	19	11
Puga, India	20	11
Raft River, ID*	21	14, 24, 25, 26
Reykjanes, Iceland	22	11
Roosevelt Hot Springs (Milford), UT	23	11
Salton Sea, CA	24	14, 27, 28, 29
Steamboat Springs, NV	25	11
Sousaki, Greece	26	11
Svartsengi, Iceland	27	30
Wabuska, NV	28	11
Waiotapa, NZ	29	11
Wairakei, NZ	30	7, 31
Updated Data:		
East Mesa: Magma Power Co.*	31	34
Heber Binary Wells*	32	35
Mammoth Lakes, CA*	33	36

*Resource from which pumping experience has been documented.

Table 2-3

SUMMARY OF DOWNWELL PUMPING CONDITIONS DATABASE--CORROSIVE SPECIES AND ESTIMATED PUMPED TEMPERATURE

Resource Log Number	Corrosive Species (ppm)							Est. Pumped Temperature °F	Comments
	Cl	SO_4	ΣCO_2	ΣH_2S	ΣNH_3	pH	Approx TKS		
Original Survey Data									
1	8,074	34	540	18	0.1	7.0	9,830	445	B, h
2	3,060	64		2		7.2	3,233	600	D, a
3	65	174		.3		9.6	325	450	D, e, f
4*	1,100	301	36			7.2	1,470	338	D, a
5*	227	96	180	14	0.1	6.5	517	358	C, b, e
6	10,220	18	1,915	467		5.4	12,635	550	A, g
7	37,010	91,010			12	8.5	129,400	525	C, e
8	2,030	159			0.9	6.5	2,237	307	C, e
9*	500	173	500	1	5	6.3	980	326	A, b, d
	1,426	179	2,000	1.5	16	6.1	2,540	335	A, b, d
10*	4,654	79	1,782	1.2	18	5.5	6,825	353	A, b, d
11*	540	174	1,044			6.5	1,100	330	B, b, f
12	6,092	73		1	2	5.9	6,220	432	D, a
13*	8,120	197		2	9	5.8	8,500	365	A, e
14	125	65	176	30		7.8	396	392	C, c, e
15	197	61			0.1	8.2	260	360	C, e
16	930	10				6.2	970	473	C, e
17	16	8	56	<0.1		6.9	80	424	A, c
	1,234	325	316	6.6		8.9	1,880	424	B, c, f
18	18	38	62	107		7.5	225	575	C, e
19	1,625	17			46	7.4	1,750	610	C, e
20	443	117		6		7.3	1,180	390	D, a
21	490	50		0.1	0	6.7	570	287	C, e
	2,170	63		0.7	0.2	7.5	2,280	333	C, e
22	19,000	32	1,437			6.0	20,500	502	A, g
23	4,100	127				6.4	4,365	450	C, e
24	132,000	55	5,000	33		4.5	132,000	625	A
25	730	103		4.4	0.1	8.5	1,000	392	C, e
26	24,468	12,222	1,383		13.2		38,000	495	C, e
27	12,160	32	298	3.5		6.2	12,450	455	C, c, e
28	50	600		0.4	0.1	8.5	680	311	C, e, f
29	1,000	69		2	0.8	6.1	1,160	455	C, e
30	1,520	8	363	9.2	0.2	6.3	1,915	475	A, g

Table 2-3 (Continued)

SUMMARY OF DOWNWELL PUMPING CONDITIONS DATABASE--CORROSIVE SPECIES AND ESTIMATED PUMPED TEMPERATURE

Resource Log Number	Corrosive Species (ppm) Cl	SO_4	ΣCO_2	ΣH_2S	ΣNH_3	pH	Approx Tks	Est. Pumped Temperature °F	Comments
Updated Data									
31*	2,748	86	262	0.9	12.2	5.7	3,109	350	A, b, d
32*	8,132	58	230	2.0		5.7	8,420	360	A, b, d
33*	304	127	1,011	4.0	2.7	6.1	1,449	339	A, b, d

*Resources for which downwell pump experience is documented. °C = (F-32)/1.8

TKS = Total Key Species = WT sum of chloride, sulfate, ionized carbon dioxide species, hydrogen sulfide and ammonia.

A - Probable concordance with downwell chemistry.
B - Probable concordance with downwell chemistry except pH.
C - Questionable concordance with downwell chemistry.
D - Probable noncordance with downwell chemistry.

a - Flashed or degassed sample.
b - Unflashed sample.
c - Deepwell sample.
d - Demonstrated sampling/analysis techniques.
e - Sampling and/or analysis method not known.
f - pH suggests flashing or degassing.
g - Published calculation of deep well conditions.

Table 2-4

PERCENTILE DISTRIBUTION OF CORROSIVE SPECIES CONCENTRATIONS IN GEOTHERMAL BRINES

%-Tile	pH	Concentration (ppm) Cl	SO_4	ΣCO_2	ΣH_2S	ΣNH_3
10	4.8	65	11	62	0.2	0.1
20	5.4	258	32	180	0.8	0.1
30	5.7	524	45	262	1.2	0.2
40	5.7	1,000	61	316	2.0	0.6
50	5.9	1,520	69	500	2.8	1.8
60	6.1	2,175	97	1,011	4.4	7.0
70	6.1	6,020	140	1,383	9.3	12
80	6.3	9,383	177	1,782	23	14
90	6.7	24,460	600	2,000	85	18
Max Value	6.9	132,000	91,010	5,000	467	46

Table 2-4 indicates that the probability is 80% that any given resource will contain not more than 9383 ppm chloride.

The variables in Table 2-4 are not linked, but are independent of each other. This means that each must be considered individually. There is no reason to expect that a resource having high chloride will, for example, have high sulfate. In fact, the converse is often true.

TEMPERATURE

USGS (U.S. Geological Survey) data for geothermal resources in the western United States indicate that half of all resources hotter than 250°F (121°C) will be between 250° and 300°F (121° and 149°C). A little more than 80% will be cooler than 360°F (182°C), and less than 10% will exceed 400°F (204°C) (32).

GEOTHERMAL CORROSIVITY CLASSIFICATION OF PUMPED RESOURCES

Three of the six corrosivity classes defined by Radian characterize resources which may be exploited by downwell pumping in the near term. Tables 2-5 through 2-7 characterize these three resource classes.

Consideration of the USGS temperature data, along with the data presented in Tables 2-3 and 2-4, indicates that all of the resources pumped to date in the U.S. have fallen into Corrosivity Class IV. Table 2-7 indicates that the corrosivity of this class of brines to most commonly used pump materials is relatively mild, a prediction supported by the available downwell pump case histories. Much more severe problems can be expected when or if attempts are ever made to pump Class I resources.

SCALE AND SUSPENDED SOLIDS

Scale and suspended solids, though often extremely important in determining pump life, were not included in the above tables because their occurrences usually depend more on operational factors than on the innate characteristics of the resource.

In the resources pumped to date, the sole culprit in scaling problems has been calcium carbonate. Calcium carbonate scale will form rapidly in most geothermal brines at the point where dissolved carbon dioxide first comes out of solution. If the pump setting depth (the depth of the pump intake or suction) and well drawdown are such that the pressure at the pump suction falls below the gas breakout pressure (the pressure at which gas is first evolved from solution), then cementitious

Table 2-5

GEOTHERMAL RESOURCE CORROSIVITY CLASS I

Defining Parameters:

Resource type	Liquid-dominated
Total key species (TKS)[a]	Greater than 100,000 ppm
Chloride fraction in TKS	99%
pH (unflashed fluid)	Less than 5
pH (flashed fluid)	5 - 6
Vol. gas in steam	Less than 2.5%
Plant inlet temperature	Greater than 390°F (199°C)

Sites Reviewed:

USA - Salton Sea, CA (619°F, 326°C)

Observed Corrosion of Carbon Steel:

- In nonaerated, separated fluid, the uniform rate exceeds 50 mpy at 390°F, (199°C) but may decrease to about 15 mpy at 300°F (149°C). Serious localized corrosion.
- In separated, nonaerated steam, the corrosion rate is 10 to 50 mpy with serious pitting. Scrubbing steam does not reduce corrosion rate.

General Performance of Other Alloys:

- T316 stainless steel susceptible to pitting, crevice corrosion, and stress corrosion cracking.

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[a]Total chloride + sulfate + carbon dioxide species + sulfide species + ammonia species in separated fluid.

[b]Average resource temperature.

Source: Reference 3

Table 2-6

GEOTHERMAL RESOURCE CORROSIVITY CLASS III

Defining Parameters:

Resource type	Liquid-dominated
Total key species (TKS)[a]	10,000 to 20,000 ppm
Chloride fraction in TKS	45 to 99%
pH (unflashed fluid)	5 to 6
pH (flashed fluid)	Greater than 6
Vol. gas in steam	Not established
Plant inlet temperature	300^{o} to $375^{o}F$ (148^{o} to $191^{o}C$)

Sites Reviewed:

Mexico - Cerro Prieto ($572^{o}F$, $300^{o}C$)[b]
U.S.A. - East Mesa (6-1)[c], CA ($353^{o}F$, $178^{o}C$)

Corrosion of Carbon Steel:

- In nonaerated, separated fluid, the uniform corrosion rate is typically 5-10 mpy at temperatures of 300^{o} to $375^{o}F$ (148^{o} to $191^{o}C$). Some localized attack occurs.
- Steam separated with <0.01% moisture is equivalent to Class VI.

General Performance of Other Alloys:

- Type 316 stainless steel is susceptible to pitting and stress corrosion cracking in liquid streams.
- High strength low alloy (HSLA) steels are susceptible to sulfide stress cracking.
- Monel 400 and cupronickel are unsatisfactory.

---

[a]Total chloride + sulfate + carbon dioxide species + sulfide species + ammonia species in separated fluid.

[b]Average resource temperature.

[c]East Mesa Well 6-1 is an anomalous well.

Source: Reference 3

Table 2-7

GEOTHERMAL RESOURCE CORROSIVITY CLASS IV

Defining Parameters:

Resource type	Liquid-dominated
Total key species (TKS)[a]	500 to 10,000 ppm
Chloride fraction in TKS	45 to 99%
pH (unflashed fluid)	Greater than or equal to 5
Vol. gas in steam	Less than 2.5%
Plant inlet temperature	250° to $390^{\circ}F$ (121° to $199^{\circ}C$)

Sites Reviewed:

El Salvador - Ahuachapan ($446^{\circ}F$, $230^{\circ}C$)[b]
Iceland - HTA (four sites) (420° - $536^{\circ}F$, 216° - $280^{\circ}C$)
Japan - Hatchobaru ($482^{\circ}F$, $250^{\circ}C$)

Japan - Otake ($446^{\circ}F$, $230^{\circ}F$)
New Zealand - Broadlands ($491^{\circ}F$, $255^{\circ}C$)

New Zealand - Wairakei ($446^{\circ}F$, $230^{\circ}C$)

U.S.A. - Baca, NM ($464^{\circ}F$, $240^{\circ}C$)
U.S.A. - Brady, N.S., NV ($338^{\circ}F$, $170^{\circ}C$)
U.S.A. - East Mesa, CA ($350^{\circ}F$, $177^{\circ}C$) (except well 6-1)
U.S.A. - Heber, CA ($356^{\circ}F$, $180^{\circ}C$)
U.S.A. - Raft River, ID ($279^{\circ}F$, $137^{\circ}C$)

Corrosion of Carbon Steel:

- In nonaerated, separated fluid at 250° to $390^{\circ}F$ (121° to $199^{\circ}C$), uniform corrosion rates are typically less than 5 mpy with minor pitting.
- High strength low alloy steels are susceptible to sulfide stress cracking.
- Steam separated with <0.1% moisture is equivalent to Class VI.

General Performance of Other Alloys:

- Type 316 stainless steel is susceptible to stress corrosion cracking in aerated fluid.
- Corrosion-fatigue of 12 Cr turbine blades by geothermal steam is twice as severe as corrosion-fatigue by boiler quality steam.

---

[a]Total chloride + sulfate + carbon dioxide species + sulfide species + ammonia in separated fluid.

[b]Average resource temperature.

[c]Range of average temperatures.

Source: Reference 3

calcium carbonate scale will rapidly clog the pump. This can result in several modes of failure, including: seizure of the pump, broken impellers, and/or broken pump or line shafts. This mode of scale damage can be prevented by operating the pump so that the pump suction pressure always exceeds the gas breakout pressure.

Carbonate, or for that matter other forms of scale, can also be formed irrespective of pump suction pressure if incompatible brines from different production zones are mixed. Calcium carbonate scales are formed by this process at some East Mesa wells. Such scales form at depths below the pump, but may be carried into the pump during normal operations. The effects of such scales are more like those of suspended solids.

Suspended solids may result from downwell mixing of incompatible brines, but are generally sand particles. Sand may come from several sources. The most common source is probably sand left in the well following completion, or workover activities such as frac jobs or sand packing and redrilling. Sand may also be released by some formations, particularly at the beginning of production or following acid treatment.

Sand can damage downwell pumps in several ways. If there is enough sand, it can simply clog the pump, or become so wedged between the impeller skirt and bowl throat that the pump becomes "sand locked" and won't turn. Such failures are generally fairly prompt and may result in secondary damage such as broken impellers, pumpshafts, or lineshafts.

Sand is particularly damaging to many of the bearing technologies which have been used in downwell pumps, and can produce severe localized erosion of some pump components.

Suspended solids are not tabulated in Table 2-3 for two reasons. First, such data are very sparsely reported. Secondly, the above discussion indicates that suspended solids, unlike the chemical components, may vary widely with time. Thus, there is concern that because the few reported suspended solids data may not be representative, their use could lead to faulty conclusions.

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Section 3

LINESHAFT PUMPS: TECHNOLOGY OVERVIEW

Essentially all of the high-volume, high-temperature geothermal production pump experience gained since the previous survey has been with lineshaft pumps. The information in this report related to geothermal lineshaft pump experience is divided into three sections. This section is intended to acquaint the reader who is unfamiliar with lineshaft pumps with this technology as it has developed in response to the needs of the geothermal power industry. Section 4 documents the recent utility experiences with geothermal lineshaft pumps, while Section 5 presents results of different statistical analyses of the results of the current survey. This section is, in turn, divided into three parts. The first part provides a general description of a geothermal lineshaft pumps system. The second part traces the evolution of geothermal lineshaft pump systems and describes some of the design variations in more detail. The last part describes typical installations as of late 1986-early 1987.

GENERAL DESCRIPTION OF A GEOTHERMAL LINESHAFT PUMP SYSTEM

Figure 3-1 illustrates a typical geothermal lineshaft pump system. The nomenclature used in this figure and throughout this report was the predominating terminology at the time that this report was completed (June 1987). However, there was at that time no standard terminology for many components, and different manufacturers may use different nomenclatures.

Any lineshaft pump system consists of a multi-stage downwell centrifugal pump, a surface mounted motor (usually electric), and a connecting lineshaft assembly--containing the actual lineshaft which transmits the motor torque to the pump; the ancillary components which are necessary to support the lineshaft--and a conduit to bring the pumped brine to the surface. This conduit is usually referred to as the production column, or more rarely as the production tube.

Lineshafts

There are two basic types of lineshaft assemblies, referred to as open and enclosed systems. In either system, the lineshaft is supported by lineshaft bearings, which

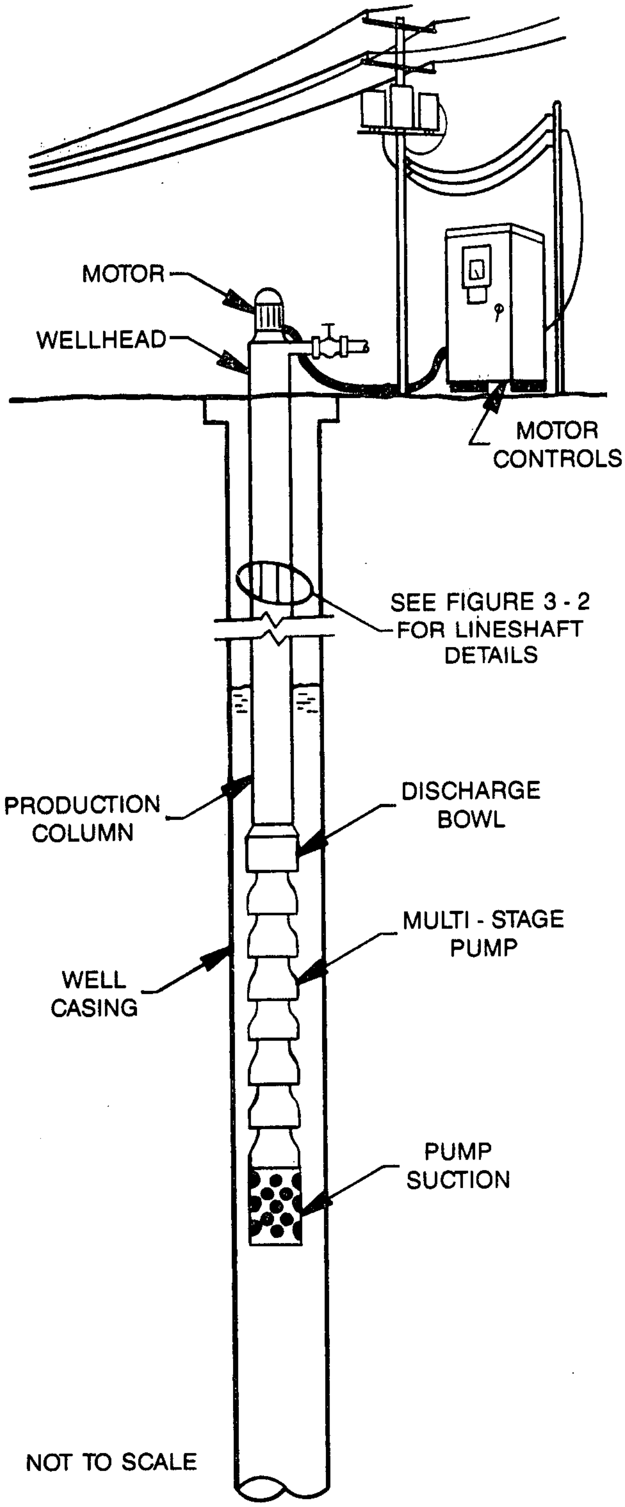


Figure 3-1. Typical Lineshaft Pump System Installation

are in turn centralized within the production column. Figure 3-2 illustrates the components and differences of the two technologies.

Open lineshafts are commonly used in domestic water well and agricultural lineshaft pumps, but have proved totally unacceptable for high-temperature, high-capacity geothermal brine production applications, probably primarily as a consequence of both suspended solids and the inherent poor lubricity of hot brine.

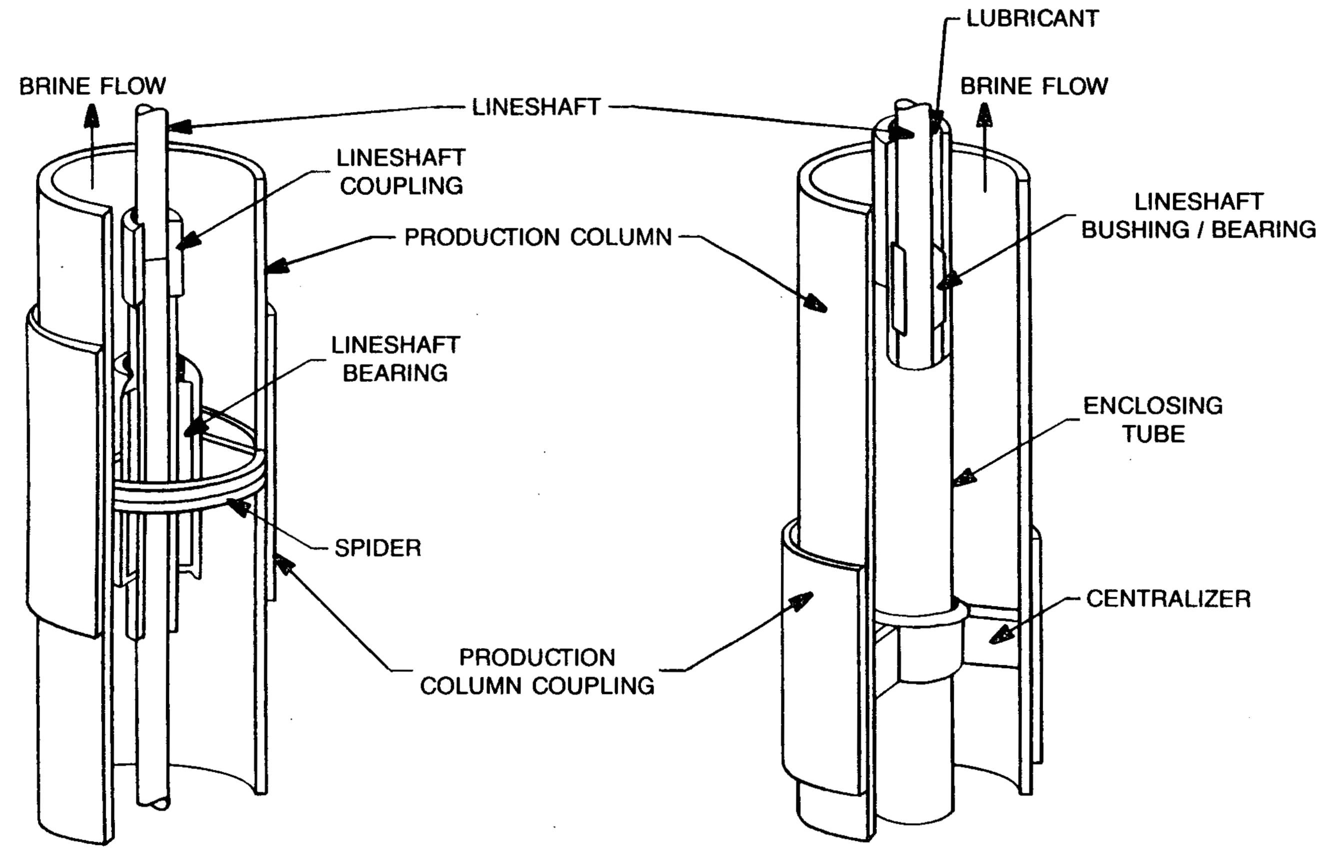
All but a few early experimental geothermal production pumps have used the enclosed lineshaft design. In this design, the lineshaft and supporting bearings are housed within an enclosing tube, alternately referred to as a lubrication tube. The enclosing tube is in turn supported within the production column by centralizers. Lubricating fluid is pumped down the enclosing tube from the wellhead. In high-temperature geothermal production pumps, the lubricant is high-viscosity oil.

Pumps

All geothermal lineshaft pumps use intermediate specific speed enclosed impellers similar to the agricultural pumps from which they evolved. Like most features of geothermal pumps, this basic impeller design is a necessary compromise. Low specific speed impellers operating at high rpms are characterized by high head per stage relative to flow, thus requiring fewer stages. However, the relatively larger thrust of such impellers generally requires thrust-compensated design. Thrust compensated impellers are more vulnerable to erosion of the "balancing holes," which dramatically reduces pump efficiency. Or the balancing holes may become plugged, resulting in rapid thrust bearing failure. Thrust compensated impellers are not commonly used for these reasons.

High specific speed impellers have a high flow-to-head ratio, requiring too many stages to be economical. Thus, intermediate specific speed enclosed impellers without thrust compensation have turned out to be the most economical design, providing the best compromise between higher efficiencies, lower velocities (hence less erosion), and preferred mechanical designs. This essentially fixes the basic impeller and bowl diffuser geometries.

Most, if not all, geothermal lineshaft pumps have been driven by 4-pole, 60 Hz electric motors at 1750 to 1780 rpm, since this speed generally provides the most efficient pump with midrange specific impeller speed and economical number of stages. Doubling the speed would double the flow and raise the head pressure by 40%, but would cause the pump to wear four to eight times faster.



OPEN LINESHAFT BEARING LUBRICATION

ENCLOSED LINESHAFT BEARING LUBRICATION

Figure 3-2. Lineshaft Details

All lineshaft pump impeller assemblies have a certain latitude of up and down motion, called axial end-play (alternately referred to as "lateral motion"). This axial end-play is required to accommodate differential thermal expansion during pump warm up, as well as a shaft characteristic called "relative shaft elongation", explained in the next two paragraphs.

The entire load of the pump impeller assembly and lineshaft is a hanging weight, supported by the motor thrust bearing. When the pump is started, the unbalanced downthrust on each impeller elastically stretches or elongates the lineshaft. The impellers also support a portion of the hydraulic head produced by the brine in the production column when the pump is operating, which further stretches the lineshaft. The amount of elongation increases with greater setting depth and larger production column size, and decreases with increasing lineshaft diameter.

The production column is also elongated by the hydraulic head of the brine in the column, but this elongation is a function of the column wall thickness and is independent of the shaft elongation. The difference between the column and shaft stretch is the "relative shaft elongation".

The axial end-play is the total axial movement of the impellers within the pump bowls. It must be larger, with a suitable margin of safety, than the relative shaft elongation over the full operating range of the pump. Otherwise, the impellers will "top" or "bottom out", destroying the pump. Today's high-temperature geothermal pumps have 4 inches (10 cm) of axial end-play. If necessary, lineshaft diameter is increased to limit relative shaft stretch to not more than about 2.5 inches (6.4 cm).

All other factors being equal, the productive capacity of a geothermal well increases with well diameter, reducing the number of wells required to provide a given quantity of brine. However, cost per well increases rapidly with diameter, and there is an optimum well size. Most geothermal power wells in liquid-dominated resources are cased in their upper reaches with 13 5/8-inch (nominal) casing. (API pipe sizes and pump diameters are descriptive labels rather than precise dimensions. As such, they do not have precise metric equivalents.) The lower parts are then commonly cased with 9 5/8-inch pipe.

The largest pump which can be fitted into 13 5/8-inch casing is the nominal "12-inch" pump, which is literally 11.675 inches (29.65 cm) in diameter. This size pump is the most commonly used in geothermal wells and, in general, the following discussion will relate to this pump size. This discussion will first address the lineshaft and production column, then the pump, and finally other components.

By convention, components of a pump system are numbered in the order in which they enter the well. Thus, the lowermost pump stage is designated as the "first", with successive stages being numbered upward. The same convention applies to components of the lineshaft.

COMPARATIVE ANATOMY OF HIGH-TEMPERATURE GEOTHERMAL LINESHAFT PUMPS

In biology, comparative anatomy is the discipline which compares the structure of different organisms and determines how they are related and how they have evolved. The development of high-temperature deepset geothermal lineshaft pumps has been an evolutionary process, since the common ancestors of all current high-temperature lineshaft pumps were the agricultural lineshaft pump and close-coupled barrel pump. This section traces the development of current geothermal pumps from those origins. Considerable insight into the problems which these pumps face may be gained from a review of their evolution.

Lineshaft Anatomy and Development

Virtually all domestic high-temperature geothermal lineshaft pumps use enclosed, oil-lubricated lineshafts. Early experience, reviewed in Section 4, showed that open bearing and water-lubricated enclosed bearing systems were inadequate for the temperatures of most interest for geothermal binary power production.

Early enclosed oil-lubricated lineshafts at East Mesa and Heber experienced severe shaft vibration and rapid bearing failure. Most failed in less than 1000 hours. The best available information suggests that these lineshafts were operating at their critical speed, resulting in shaft resonance and whip which rapidly damaged the bearings.

In late 1981, Johnston Pump Company apparently made several modifications in the shaft design and installation practice which essentially eliminated this sort of early shaft failure. The two main modifications were apparently a change in bearing spacing to separate shaft critical and operating speeds, and increased tensioning of the enclosing tube to keep it as straight as possible.

The best available information indicates that little additional change in lineshaft design has occurred at East Mesa since 1981. The East Mesa lineshaft design has thus become an unofficial industry standard, typical of all lineshaft designs for 12-inch pumps set at about 1000 feet (305 m).

For 12-inch pumps set at a nominal depth of 1000 feet (305 m), the assembly consists of an 8 to 8 5/8-inch diameter production column, 3- to 4-inch diameter enclosing tube, and 1.6875- to 2.000-inch (4.286 to 5.080 cm) diameter lineshaft, all made from carbon steel. The bearings are simple bushing journals with a single spiral lubricating groove. They are 4 inches (10 cm) long and are externally threaded to also serve as couplings for the enclosing tube.

The production column and lineshaft come in 20-foot (6 m) lengths, while the enclosing tube is supplied in 5-foot (1.52 m) sections. Four lengths of enclosing tube are coupled together by lineshaft bearings to form a "stick" which is supported in a section of production column by centralizers spaced every 20 feet, while the lineshaft is inserted through the bearings.

The bearings are SAE 67 bronze, a leaded red bronze; this appears to be a universal choice. These bearings come in three basic configurations: solid bronze, bronze bearing insert pressed into a steel sleeve, and bronze bearing insert with longitudinal grooves which form channels between the bronze insert steel sleeve when the insert is pressed in place. The solid bronze bearings cost about one-fifth as much as the sleeved versions. Johnston favors the solid version and asserts that, due to its thicker cross-section, the solid bronze version has as good a tensile load capacity as the steel-sleeved bearing. However, the lineshaft bearing fractures documented in this survey have all occurred in the solid bronze configuration.

The argument for providing channels between the bronze journal and steel sleeve in that design variation is that this makes it easier to fill the enclosing tube with oil during startup. This is true, but these channels also bypass oil around the bearing/shaft interface, where the maximum possible lubrication is needed.

Medium carbon steel, such as AISI 1035, apparently remains the shaft material of choice, though other alloys have also been successfully used. These alternates include AISI 4130 and 4140 high strength low alloy steels, as well as Type 416 stainless steel. The general consensus seems to be that the carbon steel is adequate, and also less expensive.

Over the years there has been a gradual trend toward thicker, hence stiffer, lineshafts. Lineshafts for 12-inch pumps supplied in 1981-1983 apparently were 1.6875 inches (4.286 cm) in diameter, except for the top 200 feet (61 m), which had a diameter of 1.9735 inches (4.921 cm). Lineshafts for the same size pumps supplied in 1986 were 1.9735 inches (4.921 cm) in diameter, except for the top 200 feet (61 m), which had a diameter of 2.000 inches (5.080 cm).

During normal operation, high-temperature oil is continuously pumped into the enclosing column at the wellhead, frequently at a rate of about 1.5 gpd (5.7 L/day). This oil flows continuously downward, flushing and lubricating the bearings. In most installations, the oil is discharged through ports in the discharge bowl, into the casing annulus. The accumulated oil is periodically removed by briefly producing the well through the casing annulus. At other fields, notably at the Heber binary plant, the spent oil is vented directly into the pump discharge bowl and carried away in the produced fluid. This option results in a much higher oil injection pressure.

Keeping the enclosing tube as straight as possible is crucial in avoiding uneven bearing wear. Putting the enclosing tube under a static tensile load (tensioning the tube) straightens the curves which would otherwise result from hanging 600 to 1000 feet (182 to 305 m) of small diameter pipe in the well. Proper tensioning of the enclosing tube during installation is considered critical by both the pump manufacturers and the field operators, though there is sharp disagreement on just how much tension should be applied.

Pump Anatomy and Development

Protopumps (1970-1979). The first few high-temperature lineshaft pumps documented in the original survey were low-power, essentially unmodified agricultural pumps with an axial end-play of about 1.5 inches (3.81 cm). Figure 3-3 is a conceptual cut-away drawing of this type of pump. This drawing, like the next three, was prepared by Radian to illustrate the points of this discussion, and may not provide a completely accurate portrait of any particular pump.

The first few attempts to use this design at high power showed that the axial end-play was totally inadequate.

Peerless Geothermal Pump (1979-present). In early 1979, Peerless Pump Company introduced their high-temperature geothermal pump. The basic design appears to have changed very little since that time.

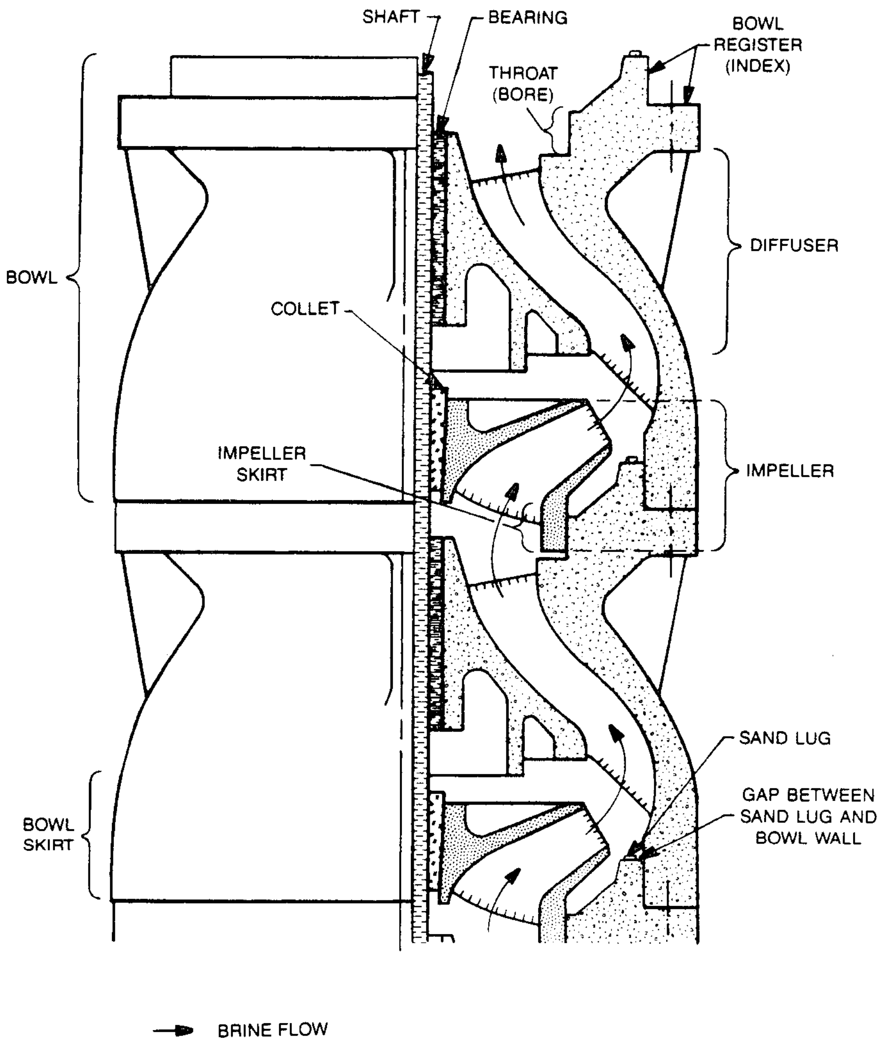


Figure 3-3. Protopump (1970-1979)

Figure 3-4 illustrates the basic design. The axial end-play was increased to 4 inches (10.2 cm) with concomitant extension of the impeller suction skirt and deepening of the throat. The deeper throat was accommodated by elongating the neck of the bowl above the diffuser and machining the throat down into the next lower bowl. Since the throat is recessed below the bowl flange, this will be termed a "recessed throat" design.

The impellers and bowls are cast iron, while the shaft is generally Type 416 stainless steel. At different times, Peerless has apparently tried a variety of bronzes, as well as Ni-resist cast iron, as bowl bearings. Both tungsten carbide hardfaced and uncoated shafts have been used. Pumps supplied in 1985-1986 were equipped with SAE 660 bronze bearings and Type 416 shafts with tungsten carbide hardfacing at the bearing locations.

Sand, a fact of life in geothermal operations, tends to recirculate in the areas just below the lower shroud of the impeller discharge, causing severe erosion. The recessed throat design places the lower shroud of the impeller discharge at about the height of the raised face of the bowl discharge flange. To combat erosion in this area, Peerless incorporated sand lugs (see Figure 3-4) which protrude upward from the bowl discharge flange to break up vortex flow in this area. However, these sand lugs are relatively small and do not extend radially all the way to the register of the flange face. As a result, entrained sand can circulate at high velocity between the outboard ends of the sand lugs and the bowl wall. This erodes the flange face and lugs. More seriously, in some cases the sand erosion has made knife-like cuts into the bowl walls, intersecting the bolt taps, slicing into the bolts which hold the bowls together.

Modified Peerless Pumps (circa 1981-present). Johnston Pumps entered the geothermal market in about 1981 by rebuilding and modifying existing Peerless pumps. Johnston made two major modifications to the Peerless design, both illustrated in Figure 3-5. They brazed on new, more robust sand lugs which extended flush with the register, leaving no gap between the lug and the bowl wall. They also fitted both the impeller suction skirt and the throat with hard alloy wear sleeves or rings. It is Johnston's theory that when the main bearings have worn to the point where the skirt and throat make contact, these wear sleeves serve as a "second bearing", stabilizing wear at that point.

Johnston reports that they launched a bearing development program in 1982, initially experimenting with different hardfacings. They found that all of the hardfacings

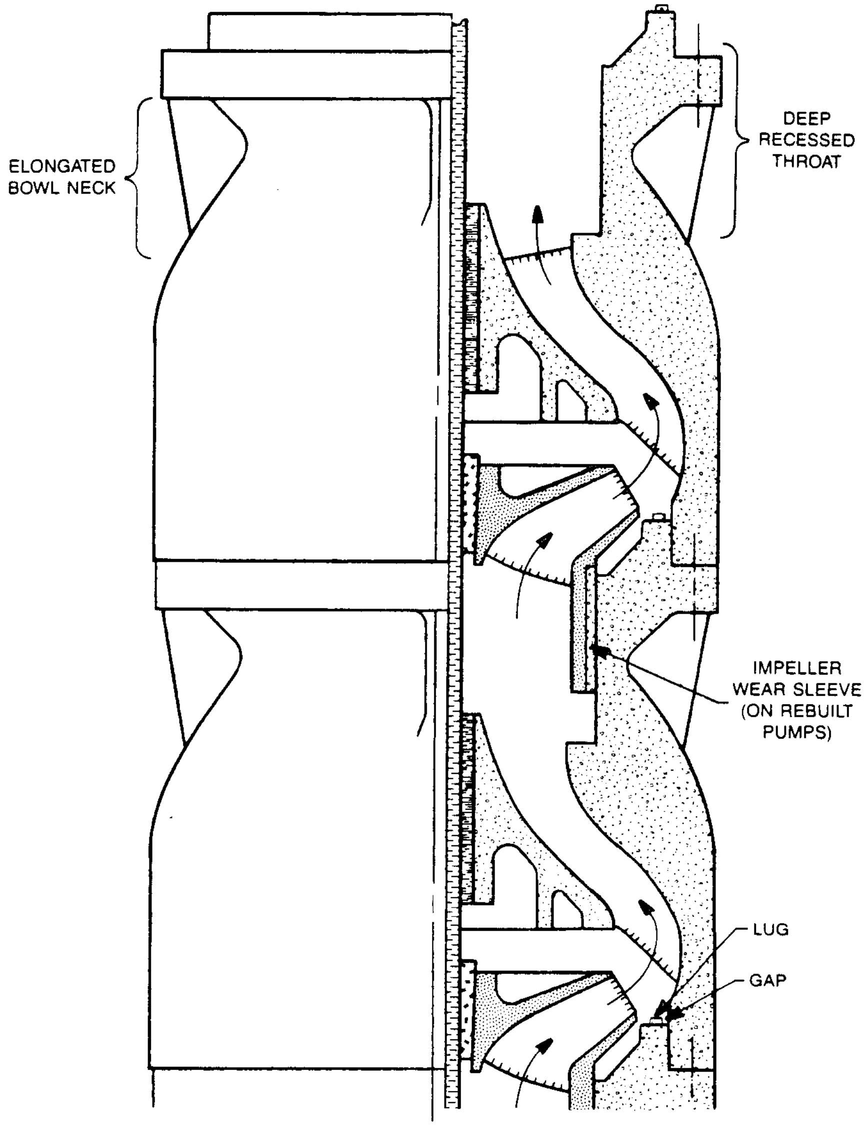


Figure 3-4. Peerless geothermal pump (1979-present). Note deep recessed throat and elongated bowl neck.

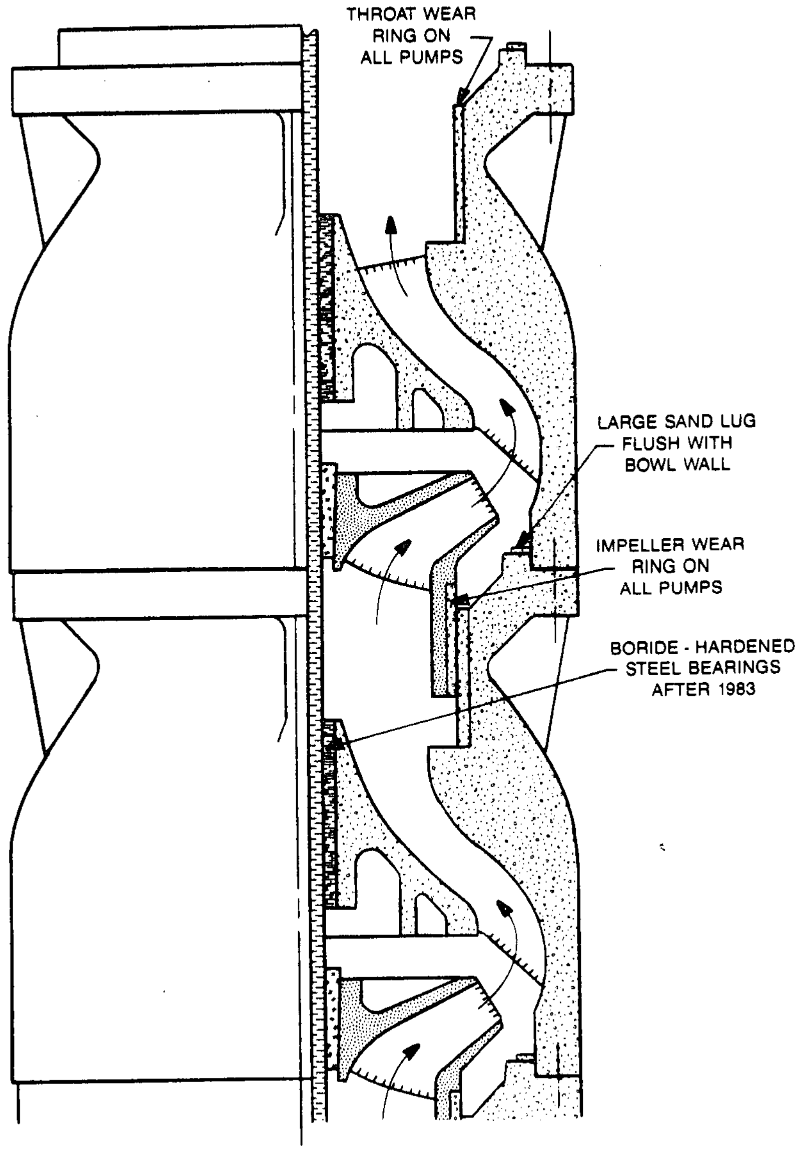


Figure 3-5. Modified Peerless pump (circa 1981-present). Note deep recessed throat, elongated bowl neck, enlarged sand lugs, and wear rings.

they tried were susceptible to spalling under some operating conditions, creating severe bearing wear. In 1983, Johnston deployed an alloy steel bearing metallurgically hardened with boron. This boride-hardened steel bearing was run against uncoated 17-4PH stainless steel shafting. Johnston considers details about this bearing to be proprietary, and they continue to use this bearing technology today. In fact, by late 1986, virtually all geothermal lineshaft pumps in utility service were equipped with this bearing technology.

Johnston Geothermal Pump (1984-present). In 1984, Johnston introduced their own geothermal pump design. They accommodated the axial end-play requirement in a different way, illustrated in Figure 3-6. They modified the bowl casting so that an integral neck extends upward above the bowl discharge flange into the suction of the next stage. The throat is machined into this internal neck, creating a throat raised above the discharge flange; a "raised throat" design. The diffuser-to-bowl discharge height is not elongated as in the recessed throat design. The suction skirt of the bowl is elongated instead.

This design raises the lower edge of the impeller discharge well above the height of the bolt taps, eliminating the risk of erosion through the connecting bolts. This design also allows for extremely deep and robust sand lugs (not shown in Figure 3-6) extending axially along the outside of the integral neck which houses the raised throat.

Johnston indicates that all of their pumps have used the boride-hardened bearing/uncoated 17-4PH shaft technology. Like all of the previous pumps, the impellers and bowls are cast iron. The impeller suction skirts and throats are lined with hard alloy wear sleeves similar in function to those installed in the modified Peerless design.

Johnston has also made the castings for their pumps more robust, with thicker bowl walls and diffuser vanes, as well as thicker impeller skirts, shrouds, and vanes, than are typical of other vertical turbine applications, such as hydrocarbon condensate or firewater pumping.

These heavier castings may be necessary for the harsh geothermal environments, but there is a penalty in the form of lower flow at peak efficiency. For example, a typical Johnston 12-inch geothermal pump installation may produce 1600 gpm (100 L/s) at peak efficiency, about 25% lower than could be obtained by eliminating the

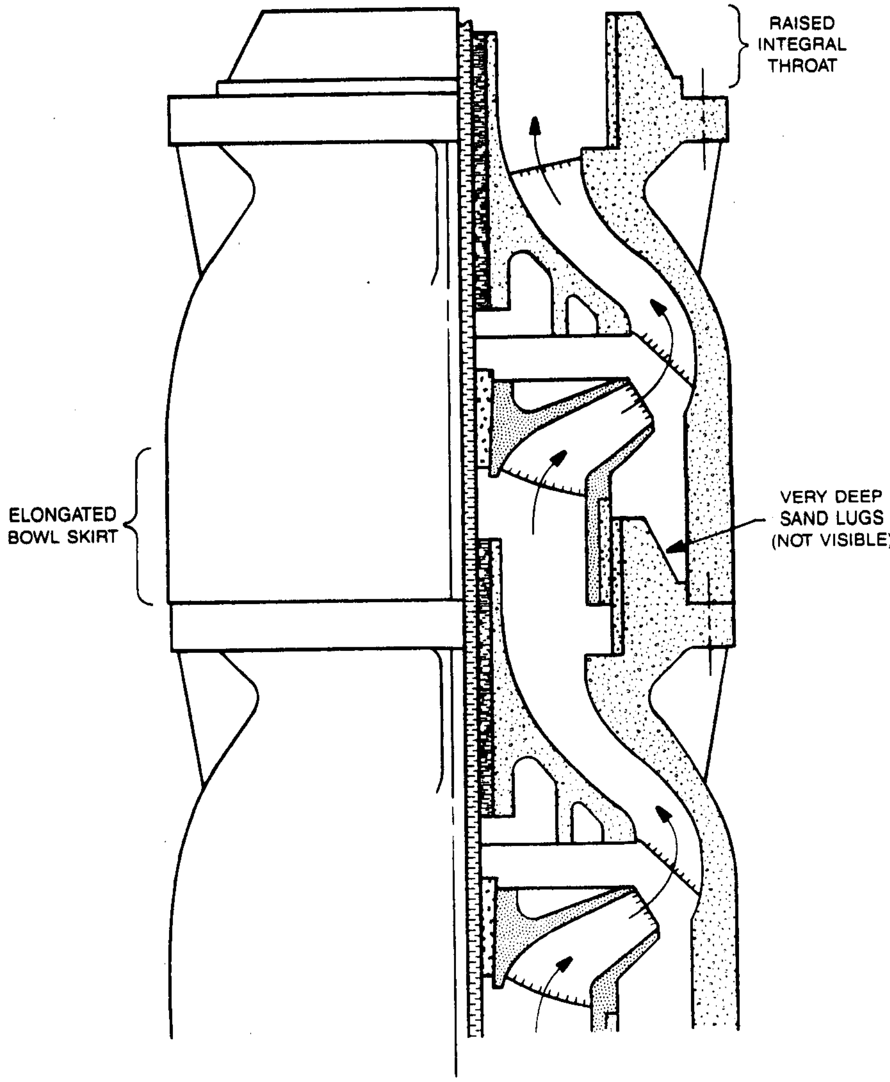


Figure 3-6. Johnston geothermal pump (1984-present). Note raised throat and elongated bowl skirt. Very deep sand lugs are present, but not shown in this drawing.

erosion and wear allowance and making the castings as thin as possible. However, geothermal wells capable of sustaining greater flows are rare.

Other Components

Suction Bowl and Screen. The intake of the pump unit is commonly called the "suction". The first stage, the suction bowl, contains the tailshaft bearings. The opening of the suction bowl is commonly protected by a conical slotted screen which protects the intake as the pump assembly is lowered into the well. It also screens out any large debris which may be carried up from below. It provides no protection against sand or finer suspended solids.

If the pump suction is not set deep enough, so that drawdown lowers the brine pressure in the suction bowl to the point that gas begins to evolve, then it is likely that this bowl will become plugged with cementitious calcium carbonate scale.

Sectional Couplings. There is a practical limit to the length of single pump shafts. When the pumping application requires more stages than can be assembled on this shaft length, two or more assembled pump sections are connected in tandem. This connection can be made in three ways.

Peerless Pumps uses a "tandem adaptor", a field-assembled device in which the separate pump shafts are joined together with a threaded coupling. This allows the pump to be shipped to the site in two or more sections.

Johnston uses a coupling-less shop splice, resulting in the creation, at the factory, of a single very long pump assembly. This unitized pump is shipped inside a length of appropriately sized casing for rigidity.

It is also possible, though apparently not common, to connect the pump shafts via a threaded coupling while the pump sections are still at the factory. This also creates a factory aligned unitized pump. It is shipped inside a length of casing suitably sized for the required rigidity.

Discharge Bowl. The discharge bowl is the topmost component of the pump unit. It connects the pump to the production column, and contains the termination of the enclosing tube as well as the coupling between the line and pump shafts. The design of the discharge bowl determines whether the spent lubricating oil will be discharged into the produced brine or into the annular space between the production column and the cemented well casing.

Seals. Each pump bowl is sealed against the register of the preceding bowl by an elastomeric O-ring. Viton is generally used, apparently with good results, though it is necessary to guard against substitutions, since all O-rings look alike unless color coded.

More recently, Y267 EPDM, a specialized geothermal elastomer developed by DOE, has been used. This elastomer has remarkable resistance to geothermal brines and oil, but is currently more expensive and much harder to obtain.

The other, and more problematic, seal is the dynamic mechanical seal between the lineshaft and enclosing tube at the wellhead. This seal must contain the pressurized hot oil confined in the enclosing annulus. A rupture can spray workers with scalding oil.

Both single cartridge and double/tandem seals are used. Single seals can run dry under certain circumstances, leading to rapid failure. Mechanical seals which can be operated as double or tandem seals may be preferable since the oil pressure may be either higher or lower than the buffer fluid pressure under different operating conditions. The double/tandem seal arrangement can cope with either eventuality and provides protection against running dry. Seal failure manifests by the appearance of oil in the buffer fluid. Some single seal applications incorporate a seal lip modification, in effect creating a double/tandem system.

In the failures documented in this study, most of the wellhead seal failures, other than infant mortalities, appear to presage pump failure, and are probably the result of the impending failure. However, the wellhead seals may require periodic replacement, particularly as pump endurance increases. Johnston has developed a sliding seal and aligned coupling device which fits over the lineshaft at the wellhead. This device makes it possible to replace the mechanical seal without removing the motor or releasing hot, pressurized oil. Johnston considers the details of this device and its corresponding seal box to be proprietary.

TYPICAL CONTEMPORARY INSTALLATIONS

The next section provides fairly detailed descriptions of lineshaft pump installations at three commercial scale geothermal binary-cycle power plants. Table 3-1 provides an overview of the physical (i.e., dimensional) characteristics of these installations. This table can be viewed as representing "typical" installations.

Table 3-1

TYPICAL RECENT LINESHAFT PUMP INSTALLATIONS

	Magma Power Co.		Mammoth	Heber Binary Plant	
	Standard	Slimline	Pacific	Zone 1	Zone 2
Well					
Total depth (ft)	6800-7900	6800-7900	655-752	-	-
Production zone					
Top (ft)	4500	4500	413-691	2000	4000
Bottom (ft)	7500	7500	443-752	4000	6000
Static water level (ft)	-	-	58-219	-	-
Pump setting depth (ft)	936-1006		597-612	750	1000
Casing diameter at pump set (inches)	13 5/8	9 5/8	13 3/8	13 5/8	13 5/8
Gas breakout pressure (psia)	200-280	200-280	120	-	-
Lineshaft					
Column dia (in)	8 5/8	6 5/8	-	10 3/4	10 3/4
Enclosing tube dia (in)	3-4	2 1/2	-	-	-
Topshaft diameter (in)	1 15/16-2	1 1/2	11 11/16	-	-
Shaft diameter (in)	1 11/16 - 1 15/16	1 1/2	11 11/16	-	-
Brg spacing	5	5	5	5	5
Seal type	Chesterton	Chesterton	Mechanical	Borg-Warner	Borg-Warner
Pump					
Manufacturer	Johnston	Johnston	Johnston	Peerless	Peerless
Model	12EMC	8CC	12EMC	12HXHG	12HXLG
Stages	14-16	33-38	11	20	18
Rated GPM	600-770	325-350	800	1650	1320
Diameter (inches)	11 3/4	8	11 3/4	12 (nom)	12 (nom)
Motor					
Make	GE	GE	Elect.	GE	GE
Type	Vert.	Vert.	HSRGZCH	TEFC	TEFC
HP	300-350	125-150	250	600	600
RPM	1780	1780	1770	1790	1790
Volts	440	440	4160	4160	4160
Phase	3	3	-	-	-
Hertz	60	60	60	60	60

- Data not available

Table 3-2 summarizes the state-of-the-art materials selections for high-temperature, high-capacity geothermal lineshaft pumps as of early 1987. Almost all pumps in service as of mid-1987 had these materials selections.

Table 3-2

CONTEMPORARY (EARLY 1987) LINESHAFT PUMP SYSTEM MATERIALS OF CHOICE

Component	Material
Lineshaft Assembly	
Lineshaft	1035 carbon steel preferred; 4130, 4140, and Type 416 sometimes used
Bearings	SAE 67 leaded red bronze
Enclosing tube	Carbon steel
Production column	Carbon steel
Pump Assembly	
Bowls	Gray cast iron
Impellers	Gray cast iron
Impeller wear ring	Type 410 or 420, HRC 25 max
Throat wear sleeve	Type 410 or 420, HRC 25 max
Shaft[a]	Alloy 17-4PH, H1150 heat treatment, HRC 33 max
Shaft hardfacing[a]	None
Bearings[a]	Low alloy steel interstitially hardened with boron
O-rings	Viton or Y-267 EPDM

[a]Older pumps used SAE 67 or SAE 660 leaded red bronze bearings and Type 416 shafts with or without flame-sprayed tungsten carbide hardfacing. Virtually all had been removed from service by early 1987.

Section 4

UPDATED LINESHAFT PUMP EXPERIENCE

The experience gained since the original survey has been limited almost exclusively to lineshaft pumps; there has been a steady evolution of this technology. The last section presented an overview of lineshaft pump technology and traced its development. This section begins with a brief review of the status of lineshaft pumps at the close of the prior survey, followed by an experience update presented on a site-by-site basis. This section concludes with a discussion of the overall significance of these experiences, while the next section presents the results of a statistical evaluation of the results.

PRIOR STATUS OF LINESHAFT PUMP TECHNOLOGY

This section presents a brief overview of the status of lineshaft pump development as of the spring of 1983, in order to provide a perspective for the more recent results reported below. The reader is referred to EPRI Report AP-3572 for more details.

A total of 33 domestic attempts to produce high-temperature geothermal brines using lineshaft pumps were documented in AP-3572. Table 4-1, modified from Table 3-3 of that report, summarizes the failure and distress modes of these lineshaft pumps. This table indicates that the almost exclusive cause of failure was related to the lineshaft.

Fifteen years of Icelandic experience at a 270°F (132°C) resource near Reykjavik demonstrated that water-lubricated graphite-filled teflon bearings in enclosed lineshafts provided runs of two to four years. Domestic experience, documented in AP-3572, indicated that the practical limit for such bearings was about 320°F (160°C) due to softening of the teflon.

There were no documented attempts to use water-lubricated lineshaft bearings of any material above 328°F (164°C). All attempts at higher temperatures used leaded bronze bearings in enclosed oil-lubricated lineshafts.

Table 4-1

COMPONENT DEGRADATION DISTRIBUTION FOR LINESHAFT PUMPS
(AP3572 Data Only)

Incidence of Failure/Distress	Instances as Fraction of Pumps Surveyed			
	Failure[a]		Distress[b]	
	Total	Operational[c]	Total	Operational[c]
Overall Pump Condition	19/33	2/33	4/33	2/33
By Cause/Location				
Lineshaft				
- shaft wear	0/33	0/33	1/33	0/33
- bearing wear	17/33	0/33	2/33	0/33
Pump				
- impeller/bowl erosion	0/33	0/33	3/33	2/33
- shaft/bearing wear	0/33	0/33	3/33	1/33
- scale-up	2/33	2/33	1/33	1/33

Notes: Insufficient data to assess the location/cause or extent of damage to 8 of the 33 pump systems surveyed.

[a]Deemed to be the component whose degradation made the pump inoperable.

[b]Degradation was not sufficient to render the pump inoperable, or damage was judged consequential to the component which had failed. Distressed components would likely have produced their own failures given longer pump life.

[c]Indicates that some identifiable error in installation and/or operation was a key contributor to the observed degradation.

One of these latter pumps ran 2200 hours prior to lineshaft failure, though the remaining 16 units each ran less than 1000 hours. All suffered severe lineshaft vibration leading to rapid wear of the lineshaft bearings and scoring of the shafts.

Late in 1981, Johnston Pump determined that these lineshafts were being operated very near their critical speed, producing the vibration which then led to rapid bearing failure. Johnston apparently corrected this problem by modifying the bearing spacing and shaft diameter. The first pumps with this improved lineshaft had been put into service and were still running at the conclusion of the previous survey. At that point in time, they appeared to offer a significant improvement in expected run life.

Radian concluded that the paucity of pump, as opposed to lineshaft, problems resulted from the relatively short run lives which had been demonstrated to that time. Radian projected that the near-term pump life would be in the 6 to 12 month range, and that pump, rather than lineshaft, problems would then predominate.

EXPERIENCE UPDATES

Magma Power Company at East Mesa, California

Magma Power Company began operation of their 10 MW binary plant at East Mesa in the 1970's. Since this was the first large geothermal power plant to use pumped brine, Magma Power has accumulated more downwell pump experience than any other domestic operator. Most of the lineshaft pump experience documented in the initial survey, and reviewed above, occurred at this site. Since the prior survey, more than 40 additional pumps have been installed at this plant.

Resource Characteristics and Pump System Design. The resource at Magma's location is known to consist of three separate, but almost vertically overlapping, production zones. Based on the well perforation programs, the three zones lie between roughly 4500 and 7500 feet (1370 and 2290 m). The formations apparently consist of multiple layers of shale and fine sandstone. Apparently both fracture flow and formation permeability play significant roles in individual well productivity.

The composite chemistry of the produced brine is summarized in Table 4-2. The brine produced from different wells varies considerably, as indicated by the ranges shown in this table. The pH ranges between 5.5 and 5.8, while chloride and total dissolved solids (TDS) range from 810 to 3657 ppm and 2160 to 8994 ppm, respectively. The gas breakout pressure ranges from 244 to 280 psig (1.8 to 2.0 MPa). The pumped

Table 4-2

PROPERTIES OF EAST MESA BRINE
(ppm except pH)

Component	Mean	SD	N	Range
pH	5.68	0.11	10	5.52 - 5.80
Chloride	2748	1375	10	810 - 3657
Sulfate	86.4	19.1	10	56 - 113
Total CO_2[a]	262	53.6	10	194 - 323
Total H_2S[b]	0.88	0.29	10	0.5 - 1.5
Total NH_3[c]	12.2	2.3	10	8 - 15
Calcium	28.6	27.4	10	9 - 103
TDS	5340	2317	10	2160 - 8994
TSS	3.3	3.1	10	1.8 - 11
Gas Breakout Pressure (PSIG)	244	73.3	13	200 - 280

From analyses of single samples from 10 wells in 1985 - 1986. Data courtesy of Magma Power Company.

[a]As CO_2

[b]As H_2S

[c]As NH_3

brine temperature ranges from 336°F to 365°F (169°C to 185°C), depending on which well is being considered.

In addition to well-to-well variation, the brines produced by each of the three zones have different chemistries. Unfortunately, though the flow from all three zones is required for economic power generation, the middle zone brine is incompatible with the other two zones. The mixing of these brines causes a thin film of carbonate scale to form on the well casing in the production zone. This scale builds to a thickness of a few mils (a fraction of a millimeter), then exfoliates, producing paper-thin flakes of scale, called "black paper scale" by the personnel at Magma. This mode of carbonate scale formation is distinct from that which occurs when the brine pressure is reduced enough to allow flashing of dissolved carbon dioxide. The latter is an operational problem which can be combatted by adjusting well drawdown or pump setting depth. The black paper scale phenomenon is an inherent characteristic of this resource. Magma has made repeated, and as yet unsuccessful, attempts to control this scale by chemical means. At this time, the black paper scale is a fact of life at this resource. It has a pernicious effect on pump life.

There are two distinct well designs at the Magma plant. The "standard" well is similar to most other geothermal power wells. It consists of 13 5/8-inch (casing sizes are nominal descriptors without exact metric equivalent) casing set to about 1250 feet (381 m). Next, 9 5/8-inch tieback casing extends down to the vicinity of the producing horizon. The standard wells are completed through the producing zones with a combination of slotted and gun-perforated 7-inch liners. Wells 44-7, 44-7A, 44-7B, 48-7 and 48-7A, and 88-7 were completed in this fashion.

The other well design, used only at Magma, is designated by Magma as the "slimhole". These wells are cased with 9 5/8-inch casing from the surface to between 1200 and 2000 feet (366 and 610 m). Next, 7-inch tieback casing extends to the top of the production zones. These slimhole wells are completed through the formation with slotted 4-inch liners. These wells are less expensive to drill, at a considerable penalty in production capacity. Wells 61-3, 63-7, 81-7, and 83-7 were completed as slimholes.

Four different basic lineshaft pump designs have been used by Magma. These have included the Peerless geothermal pump, the modified Peerless pump with a variety of pump bearing materials, and Johnston pumps, all in the nominal 12-inch (30 cm) diameter. Johnston pumps with a nominal 8-inch (20 cm) diameter were used in the slimhole wells. One of these pumps was also installed in a standard well. All of these pumps used oil-lubricated lineshafts. This survey begins with the first pumps with improved lineshafts. The design of these lineshafts was discussed previously (see the "lineshaft anatomy" portion of Section 3).

Although complete documentation of the design of each specific pump system was not available, Table 4-3 summarizes that information which was available. In this table, the pumps are identified with unique codes which are used throughout the rest of this section. These codes take the form "2-X-Y-N", where "2" indicates the second (current) survey; "X" is the pump manufacturer--"P" indicates Peerless, "JP" indicates a Johnston modification of a Peerless pump, and "J" indicates a Johnston raised throat pump, and "Y" is the resource designation--"EM" for Magma's East Mesa resource. "N" is an arbitrarily assigned sequence number. These sequence numbers were assigned in the approximate order that the pumps were installed.

The same basic wellhead design is apparently used for all wells. Of necessity, the wellhead is a tee with the lineshaft seal assembly and motor mounted over the well. Brine passes through the side arm of the wellhead, and then through check and manual control valves. Each wellhead is fitted with automatic high and low-pressure trips to protect the pump, but there is no automatic discharge pressure control. The brine pressure at the pump suction is monitored daily via a nitrogen bubble line, and flow is adjusted manually if needed to maintain the approximately 240 psig (1.8 MPa) suction pressure required to prevent brine flashing, and consequential carbonate scaling, at the pump suction. The wells are produced at the highest flow rate which will maintain the necessary pump suction pressure.

Different wellhead seal assemblies have been used at Magma. They currently use a combined tandem/double seal designed by Johnston, in conjunction with Johnston's seal box. The latter incorporates a sliding-sleeve device which allows speedy replacement of seal rings if needed. These seals appear to last about a calendar year. There is little documentation regarding earlier seals used by Magma.

Update of Operating Experience. At the close of the prior survey, Peerless pumps with lineshafts modified by Johnston had been installed in a number of the Magma wells, and were still running at the close of that survey. As can best be determined through somewhat conflicting reports, the pumps included in the current survey begin with those same "improved lineshaft" pumps. A total of 44 lineshaft pumps are included in the current data set.

Table 4-3

EAST MESA PUMP SYSTEM DESIGN DATA

	Radian ID No.				
Design Information	2-JP-EM-1	2-JP-EM-4	2-JP-EM-9	2-JP-EM-10	2-JP-EM-11
Pump					
Model	12 LB	12 LB	12 LB	12 LB	12 LB
Stages	16	16	16	16	16
Set Depth (ft)	906	906	1006	1006	1000
Rated GPM	1000	750	750	700	750
Actual GPM (startup)	Not available	Not available	675	650	630
Lineshaft					
Column dia (in)	8	8	8 5/8	8 5/8	8 5/8
Enclosing tube (dia)	Not available	3 1/2" top 200' 4"	3" & 3 1/2"	3" & 3 1/2"	3 1/2"/4"
Shaft diameter (in)	Not available	1 11/16	1 11/16	1 11/16	1 11/16
Topshaft diameter (in)	1 15/16	1 15/16	1 15/16	1 15/16	1 15/16
Shaft material	Steel	Steel	Carb. Steel	Carb. Steel	Steel
Bearing material	SAE 67	SAE 67	SAE 67	SAE 67	SAE 67
Brg. spacing (ft)	5	5	5	5	5
Seal type	Mech. Seal	Mech. Seal	Mech. Seal 241	Mech. Seal 241	Mech. Seal 241
Motor					
Make	General Elect.	General Elect.	General Elect.	General Elect.	General Elect.
Type	Vertical	Vertical	Vertical	Vertical	Vertical
HP	350	300	350	350	300
RPM	1770	Not available	1770	1770	1770
Volts	480	440	480	480	440
Phase	3	3	3	3	3
Hertz	60	60	60	60	60

Table 4-3 (Continued)

EAST MESA PUMP SYSTEM DESIGN DATA

Design Information	Radian ID No. 2-J-EM-13	2-JP-EM-14	2-J-EM-18	2-JP-EM-20	2-J-EM-21
<u>Pump</u>					
Model	8CC	12 LB	12 EMC	12 LB	12 LB
Stages	33	14	16	15	16
Set Depth (ft)	1000	1000	1000	1000	1000
Rated GPM	350	600	750	700	750
Actual GPM (startup)	292	540	800	Not Available	675
<u>Lineshaft</u>					
Column dia (in)	6 5/8	8 5/8	8 5/8	8 5/8	8 5/8
Enclosing tube (dia)	2 1/2"	3 1/2 & 4"	3 1/2" & 4"	2 1/2"	3 1/2" & 4"
Shaft diameter (in)	1 1/2	1 11/16 & 1 15/16	1 11/16/1 15/16	1 15/16	1 11/16
Topshaft diameter (in)	1 1/2	2	2	1 15/16	1 15/16
Shaft material	Carb. Steel	Steel	Steel	Steel	Steel
Bearing material	SAE 67	SAE 67	SAE 67	SAE 67	SAE 67
Brg. spacing (ft)	10	5	5	5	5
Seal type	Chesterton Seal 2"	Chesterton Seal	Mech. Seal 241	Mech. Seal 241	Mech. Seal 241
<u>Motor</u>					
Make	General Elect.	General Elect.	General Elect.	General Elect.	General Elect.
Type	Vertical	Vertical	Vertical	Vertical	Vertical
HP	125	350	300	400	350
RPM	1770	1770	1770	1770	1770
Volts	Not Available	440	440	440	460
Phase	3	3	3	3	3
Hertz	60	60	60	60	60

Table 4-3 (Continued)

EAST MESA PUMP SYSTEM DESIGN DATA

Design Information	Radian ID No. 2-JP-EM-22	2-J-EM-27	2-J-EM-28	2-J-EM-29	2-J-EM-30
Pump					
Model	12 EMC	8CC	12 EMC	8CC	8CC
Stages	16	38	16	38	38
Set Depth (ft)	1000	900	1006	936	936
Rated GPM	700	350	700	327	326
Actual GPM (startup)	500	300	Not Available	Not Available	Not Available
Lineshaft					
Column dia (in)	8 5/8	6 5/8	8 5/8	6 5/8	6 5/8
Enclosing tube (dia)	3 1/2"	2 1/2"	3"	2 1/2"	2 1/2"
Shaft diameter (in)	1 15/16	1 1/2	1 15/16	1 1/2	1 1/2
Topshaft diameter (in)	2	1 1/2	2	1 1/2	1 1/2
Shaft material	Steel	Steel	Stainless Steel	Steel	Steel
Bearing material	SAE 67	SAE 67	SAE 67	SAE 67	SAE 67
Brg. spacing (ft)	5	5	5	5	5
Seal type	Chesterton Seal	Mech. Seal	Chesterton 241, Mech. Seal	Mech. Seal	Mech. Seal
Motor					
Make	General Elect.	General Elect.	General Elect.	General Elect.	General Elect.
Type	Vertical	Vertical	Vertical	Vertical	Vertical
HP	350	150	300	150	150
RPM	1770	1770	1770	1770	1770
Volts	460	440	460	460	440
Phase	3	3	3	3	3
Hertz	60	60	60	60	60

Table 4-3 (Continued)

EAST MESA PUMP SYSTEM DESIGN DATA

Design Information	Radian ID No. 2-J-EM-31	2-J-EM-32	2-J-EM-33	2-J-EM-34	2-J-EM-35
Pump					
Model	8CC	8CC	12 EMC	8CC	8CC
Stages	38	38	16	38	38
Set Depth (ft)	936	936	990	936	936
Rated GPM	326	326	770	300	300
Actual GPM (startup)	Not Available	350	900	335	337
Lineshaft					
Column dia (in)	6 5/8	6 5/8	8 5/8	6 5/8 TC	6 5/8
Enclosing tube (dia)	2 1/2"	2 1/2"	3"	2 1/2"	2 1/2"
Shaft diameter (in)	1 1/2	1 1/2	1 15/16	1 1/2	1 1/2
Topshaft diameter (in)	1 1/2	1 1/2	2	1 1/2	1 1/2
Shaft material	Steel	Steel	1040 Steel	1040 Steel	1040 Steel
Bearing material	SAE 67	SAE 67	SAE 67	SAE 67	SAE 67
Brg. spacing (ft)	5	5	5	5	5
Seal type	Mech. Seal 241	Mech. Seal	Mech. Seal	Mech. Seal	Mech. Seal Chesterton
Motor					
Make	General Elect	General Elect.	General Elect.	General Elect.	General Elect.
Type	Vertical	Vertical	Vertical	Vertical	Vertical
HP	150	150	300	125	125
RPM	1770	1770	1770	1770	1770
Volts	440	440	460	460	440
Phase	3	3	3	3	3
Hertz	60	60	60	60	60

Table 4-3 (Continued)

EAST MESA PUMP SYSTEM DESIGN DATA

Design Information	Radian ID No. 2-J-EM-37	2-J-EM-38	2-J-EM-39	2-J-EM-40	2-J-EM-41
Pump					
Model	12EMC	8CC	12EMC	8CC	12EMC
Stages	16	38	16	38	16
Set Depth (ft)	1006	936	1006	930	1006
Rated GPM	700	325	700	325	700
Actual GPM (startup)	Not Available	Not Available	Not Available	Not Available	Not Available
Lineshaft					
Column dia (in)	8 5/8	6 5/8	8 5/8	6 5/8	8 5/8
Enclosing tube (dia)	3"	2 1/2"	3"	2 1/2"	3"
Shaft diameter (in)	1 15/16	1 1/2	1 15/16	1 1/2	1 15/16
Topshaft diameter (in)	2	1 1/2	2	1 1/2	2
Shaft material	1095 hard	Stainless Steel	1095 hard	Not Available	Not Available
Bearing material	SAE 67	SAE 67	SAE 67	SAE 67	SAE 67
Brg. spacing (ft)	5	5	5	Not Available	5
Seal type	Mech. Seal	Mech. Seal	Mech. Seal	Mech. Seal (Chest. 241)	Mech. Seal (241 Chest.)
Motor					
Make	General Elect.	General Elect.	General Elect.	General Elect.	General Elect.
Type	Vertical	Vertical	Vertical	Vertical	Vertical
HP	350	150	350	150	350
RPM	1770	1770	1770	1770	1770
Volts	460	460	460	440	460
Phase	3	3	3	3	3
Hertz	60	60	60	60	60

The pumps in this survey are uniquely identified by sequential Radian Identification Numbers. The omission of a pump identification number means that there is no available documentation of design particulars.

In reviewing the pump experience at Magma, several points must be remembered. First, most, if not all, of the pumps installed prior to late 1983 were to some degree experimental, as Johnston tried various bearing/shaft combinations and other modifications. Also, because of the pace of pump replacement during this period, multiple variations were frequently being tried at the same time.

A second major point is that pump development was not viewed as a "project" by Magma. Documentation of precisely what pump modifications were tried is not available. In a similar vein, the available reports on pump problems and causes of failure were taken from the plant logs. There has been no requirement that the manufacturer/repairer provide teardown reports. This means that though some "pump condition" data are available, these do not lend themselves to determination of the root causes of failure.

A third major point is the large difference often found between "rotating hours" and "total time in the well." During the interval covered by this survey, the Magma Plant experienced one or more prolonged outages unrelated to the production pumps. Of course, when the plant goes offline, the pumps are turned off and sit without rotation until the plant is brought back on line.

For the reasons discussed in Section 1, this survey measures pump endurance in terms of rotating hours, rather than total hours in the well. More recent experience has shown convergence of the rotating hours and the total hours in the well, reflecting improved performance of the above-ground equipment; in the ideal case, the pumps would never stand dormant. However, it is recognized that the manufacturers tend to report pump life in terms of total time in the well, and there are arguments for this approach also. This report cannot resolve this issue, but its recognition can clarify some apparent discrepancies between the results of this survey and some manufacturers' reports.

Table 4-4 tabulates the experiences for the 42 separate lineshaft pumps installed at the Magma Power Plant since late 1981. Of the 42 pumps, excluding those installed after March 1987, 11 were either running as of 31 March 1981, or had been removed from service for reasons unrelated to failure. These pumps are excluded from further discussion of failure causes.

Of the remaining (failed) pumps:

- 29% were logged as failed with no further discussion of their condition being available. An additional 10% were logged as "worn out".

Table 4-4

UPDATED LINESHAFT PUMP EXPERIENCE AT EAST MESA

Radian Ident. No[a]	Well Ident	Pump Model	Bearing Technology Shaft/bearing	Prod Temp °F	Date Install	Date Failed/ Removed	Approx. Endurance Rotating Hours	Condition
2-JP-EM-1	48-7	12LB[b,c]-16	T416/bronze[c]	350	10/81	5/83	3920	FAILED: plugged by scale/seized
2-JP-EM-2	44-7B	12LB[d]-16	T416/chromium oxide main bearings[d]	360	12/81	2/84	7327	FAILED: "worn out"
2-P-EM-3	44-7A	12LB[d]-16	undocumented	336	1/82	12/82	1440	FAILED: broken shaft collar
2-JP-EM-4	48-7A	12LB-16	plain T416/bronze[c]	350	9/82	3/84	9104	FAILED: broken lineshaft
2-JP-EM-5	44-7	12LB[d]-16	undocumented	365	11/82	2/83	675	FAILED: suction plugged by scale
2-JP-EM-6	44-7A	12LB[d]-16	undocumented	336	1/83	9/84	4728	NOT FAILED: well taken out of service. Pump spared.
2-JP-EM-7	44-7	12LB[d]-16	undocumented	365	3/83	5/83	720	FAILED: pump shaft broken at bearing
2-JP-EM-8	44-7	12LB[d]-16	undocumented	365	5/83	6/84	5811	FAILED: pump wore out
2-JP-EM-9	48-7	12LB-16	undocumented	350	5/83	11/83	1870	FAILED: pump seized by scale
2-JP-EM-10	48-7	12LB-16	undocumented	350	10/83	5/84	3696	FAILED: pump seized by scale
2-JP-EM-11	48-7A	12LB-16	T416/some "hard" bearings, most bronze	350	3/84	2/85	7020	NOT FAILED
2-J-EM-12	61-7 new well	8CC[e]-38	plain ss/BHS[d]	348	5/84	2/85	5098	Undocumented (presumed failed)
2-J-EM-13	63-7 new well	8CC-33	plain ss/BHS[d]	349	6/84	4/86	12,450	Undocumented (presumed failed)
2-JP-EM-14	48-7	12LB-14	plain ss/BHS[d]	350	7/84	2/85	3912	FAILED: undocumented causes
2-J-EM-15	44-7	12EMC[f]-14	plain ss/BHS[d]	365	7/84	5/85	5600	FAILED: seized by scale
2-JP-EM-16	44-7B	12LB[d]-14	plain ss/BHS[d]	360	8/84	6/85	6116	FAILED: pump "worn out"
2-J-EM-17	88-7 new well	12EMC-20	plain ss/BHS[d]	348	1/85	2/85	510	FAILED: undocumented causes
2-J-EM-18	48-7A	12EMC-16	plain ss/BHS[d]	350	3/85	4/85	720	FAILED: ingested debris (scale) from well

Table 4-4 (Continued)

UPDATED LINESHAFT PUMP EXPERIENCE AT EAST MESA

Radian Ident. No[a]	Well Ident	Pump Model	Bearing Technology Shaft/bearing	Prod Temp °F	Date Install	Date Failed/ Removed	Approx. Endurance Rotating Hours	Condition
2-J-EM-19	61-7	8CC-33	plain ss/BHS[d]	350	3/85	3/86	6795	FAILED: broken pump shaft
2-JP-EM-20	88-7	12LB-15 "junk pump"	plain ss/BHS[d]	350	5/85	2/86	6244	FAILED: "worn out". "Lots of sand from frac job"
2-JP-EM-21	48-7A	12LB-16	plain ss/BHS[d]	350	5/85	10/86	10,887	NOT FAILED: pump OK
2-J-EM-22	44-7	12EMC-16	plain ss/BHS[d]	365	5/85	7/85	778	FAILED: plugged with scale
2-J-EM-23	44-7B	12EMC-16	plain ss/BHS[d]	360	6/85	10/85	2371	FAILED: suction screen plugged
2-J-EM-24	44-7A	12EMC-16	plain ss/BHS[d]	336	6/85	12/85	2087	FAILED: bottom impeller collet came loose
2-J-EM-25	44-7B	12EMC-16	plain ss/BHS[d]	360	12/85	5/86	3595	FAILED: undocumented causes
2-J-EM-26	44-7A	12EMC-16	plain ss/BHS[d]	350	1/86	4/86	2574	FAILED: "Material failure" not further documented
2-J-EM-27	83-7 New well	8CC-38	plain ss/BHS[d]	347	1/86	12/86	6178	FAILED: Impellers topped out
2-J-EM-28	48-7	12EMC-16	plain ss/BHS[d]	350	2/86	7/86	3233	Undocumented (presumed failed)
2-J-EM-29	81-7 New well	8CC-38	plain ss/BHS[d]	345	2/86	2/86	-0-	FAILED: (Presumed) sand locked
2-J-EM-30	81-7	8CC-38	plain ss/BHS[d]	345	2/86	2/86	-0-	FAILED: broken pump shaft, sand locked.
2-J-EM-31	81-7	8CC-38	plain ss/BHS[d]	345	3/86	3/86	-0-	FAILED: broken pump shaft, sand locked.
2-J-EM-32	81-7	8CC-38	plain ss/BHS[c]	330	3/86	-	8891[g]	Running 6/30/87
2-J-EM-33	44-7A	12EMC-16	plain ss/BHS[c]	336	5/86	12/86	3735	FAILED: cause unknown. Pump not recovered, well abandoned.
2-J-EM-34	61-7	8CC-38	plain ss/BHS[c]	348	5/86	2/87	4633	FAILED: cause undocumented
2-J-EM-35	44-7B	8CC-38	plain ss/BHS[c]	360	6/86	-	7132[g]	Running 6/30/87
2-J-EM-36	88-7	12EMC-16	plain ss/BHS[d]	348	7/86	5/87	5731[g]	FAILED: undocumented causes

Table 4-4 (Continued)

UPDATED LINESHAFT PUMP EXPERIENCE AT EAST MESA

Radian Ident. No[a]	Well Ident	Pump Model	Bearing Technology Shaft/bearing	Prod Temp °F	Date Install	Date Failed/ Removed	Approx. Endurance Rotating Hours	Condition
2-J-EM-37	48-7	12EMC-16	plain ss/BHS[c]	350	8/86	1/87	931	FAILED: seized
2-J-EM-38	63-7	8CC-38	plain ss/BHS[c]	349	10/86	-	4915[g]	Running 6/30/87
2-J-EM-39	48-7A	12EMC-16	plain ss/BHS[c]	350	10/86	-	4492[g]	Running 6/30/87
2-J-EM-40	83-7	8CC-38	plain ss/BHS[d]	350	1/87	-	3653[g]	Running 6/30/87
2-J-EM-41	48-7	12EMC-16	plain ss/BHS[c]	350	2/87	6/87	2645	FAILED: Broken lineshaft bearings
2-J-EM-42	61-7	8CC-33	plain ss/BHS[d]	348	3/87	-	<1000[g]	Running 6/30/87
2-J-EM-43	44-7A	12CC[h]-24	plain ss/BHS[d]	365	5/87	-	841[g]	Running 6/30/87
2-J-EM-44	48-7	12EMC-16	plain ss/BHS	350	6/87	-	<400[g]	Running 6/30/87

[a]N-X-Y-NN N = 2 indicating second survey
X = Pump Mfg. P = Peerless. JP = Johnston Modified Peerless. J = Johnston

[b]Peerless Model 12LB

[c]Documented

[d]Inferred from best available documentation

[e]Johnston Model 8CC (slimhole pump)

[f]Johnston Model 12EMC

[g]Running as of 6/30/87

[h]Johnston Model 12CC

- Another 29% were listed as having been seized or plugged by scale. Based upon the best available information, this pluggage appears to be the result of the "black paper scale" phenomenon discussed earlier, rather than resulting from excessive drawdown. It is worth noting that some of the pumps with this attributed cause of failure achieved quite long run times. This phenomenon does not appear to be a clear cause of premature failure, though the ingestion of these scale flakes is certainly not good for the pumps.

- Sand damage was the most likely apparent cause of failure for 13% of the pumps, including three which apparently sand locked, and failed to rotate after installation in a new well. This well was then reworked prior to installation of a fourth pump which was running as of 31 March 1987.

- Apparent quality control problems could account for 19% of the failures. Included in this group are a pump in which one of the impeller collets came loose, and a pump in which the impellers were allowed to "top out". Two of the remaining four pump systems suffered broken lineshafts or lineshaft couplings. Lineshafts are reused. During disassembly, it is often necessary to use a cutting torch with a rosebud tip to free "frozen" lineshaft couplings. This damages the metallurgy, and the manufacturer requires that torch heated couplings and shafts not be reused. However, it is suggested that these failures may have resulted from reuse of such couplings and shafting.

  The remaining two failures were pump shaft partings at bearings. It is suggested that inadequate shaft/bearing clearance may have initiated these failures.

A statistical analysis was made to determine if any of these broad "causes of failure" could be correlated with pump endurance. Despite the stipulation of a low level of confidence (80%), no statistical correlation was found. This lack of finding was probably a result of the small pump population size and confounding of the various "independent" variables. It does appear that a certain lack of control over manufacture, assembly, and installation contributes significantly to the incidence of early failure at this site.

Magma Power uses Johnston-directed crews to pull and replace pumps. The amount of time required to pull and replace a pump varies depending on the amount of work required on the well itself (historically 35 to 40% of pump insertions have been preceded by well workovers). If there is no work to be done on the well itself, and if all pump parts are on site and ready to go, the average time to change out a pump is about four days rig time. After the rig is off the well, about one additional day of work is required to reconnect all of the instrumental, electrical, and mechanical services.

Magma uses a gradual pump heat-up sequence requiring several separate adjustments of axial end-play. Before the pump is turned on, Magma raises and lowers the shaft to determine the points at which the impellers "top out" and "bottom out". The next step is to raise the impellers to "just topped out" and then lower them to the position (spacing) recommended by the manufacturer. This is called "spacing out the pump". Brine is then produced at low flow rates until the production temperature reaches 250°F (121°C). The pump spacing is then readjusted. This procedure is repeated again at 275°F (135°C), 300°F (149°C), and 310°F (154°). The pump is then ready to deliver brine to the power plant. After the pump has operated for 24 hours, it is spaced out one final time.

Mammoth-Pacific Power at Mammoth Lakes, California

The Mammoth Power Plant, owned by Mammoth-Pacific Power, was constructed near Mammoth Lakes (Casa Diablo KGRA) in northern California. Construction began in 1983 and power was first generated in November 1984. Firm power generation began in February 1985. The plant is a binary design with air-cooled condensers, using isobutane as the secondary working fluid. Rated net power capacity is 7 MWe.

Resource Characteristics and Pump System Design. The plant is located at an elevation of 7300 feet (2225 m) above sea level on the eastern slopes of the Sierra Nevada mountain range. There are active surface manifestations of geothermal activity, i.e., steaming ground, on the plant site. The reservoir is fractured rhyolite with an estimated depth of 400 to 800 feet (122 to 244 m). Fracture flow results in essentially unlimited permeability, with virtually no well drawdown.

Table 4-5 summarizes the results of numerous brine analyses conducted by the plant operator. This table shows a pH of 6.1, with 237 to 370 ppm chloride. Total dissolved solids are only 1414 to 1730 ppm. The gas breakout pressure for each of the wells is about 120 psig (0.9 MPa).

Four production wells, designated MBP-1, MBP-2, MBP-4, and MBP-5, were originally completed. However, the latter three wells quickly developed communication to the surface as a result of cement failure. MBP-2 was abandoned, and a new well, MBP-3, was drilled as a replacement. MBP-4 and MBP-5 were packed with sand and redrilled. Currently, MBP-5 is used as a spare well, since it has limited capacity. MBP-1, MBP-3, and MBP-4 serve as the production wells. Flow is limited by the pump capacity, rather than by drawdown.

Table 4-5

PROPERTIES OF UNFLASHED BRINE AT MAMMOTH LAKES
(ppm except pH)

Component	Mean	SD	N	Range
pH	6.1	nil	4	6.1 - 6.1
Chloride	304	49.1	16	237 - 370
Sulfate	127	33.5	16	60 - 161
Total CO_2[a]	1011	194.6	7	821 - 1370
Total H_2S[b]	3.96	1.90	7	1.6 - 7.5
Total NH_3[c]	2.69	0.72	7	1.8 - 4.0
Calcium	4.21	2.15	16	1.2 - 6.73
TDS	1487	218.5	16	1414 - 1730
TSS		Not determined		
Gas breakout pressure	120 psig for all wells.			

Data provided by Mammoth Pacific Power Co.

[a]as CO_2

[b]as H_2S

[c]as NH_3

Table 4-6 summarizes the completion program for each of the current production wells. Though exact casing programs vary, all four wells have 13 3/8-inch liner from the surface to the bottom of the well. The 12-inch nominal diameter pumps are set to a depth of about 600 feet (183 m) within this liner. This places the pump suctions near the bottom of the wells, and well below the producing zones. This unusuàl configuration is necessary to maintain sufficient brine column head at the suction to prevent gas breakout and consequent carbonate scaling.

All of the production pumps used to date at this plant have been essentially identical Johnston lineshaft units with raised throat design and boride-hardened alloy steel bearing against uncoated 17-4PH stainless steel shafting. The lineshafts are carbon steel with SAE 67 bronze bearings 5 feet (1.5 m) apart. Table 4-7 provides further design details.

The lube oil is vented into the annular space around the pump discharge bowl. However, the unusual setting configuration means that some of this oil is entrained in the produced brine, passes through the plant, and is reinjected into the formation.

Pump control at the Mammoth-Pacific plant is simple. For wells MBP-1, MBP-3, and MBP-4, the pumps are operated at full capacity with the wellhead throttle valves wide open. The overall discharge pressure is controlled by a 120 psig (932 kPa) pressure control valve at the brine/hydrocarbon heat exchanger discharge. Well MPB-5 is less productive. Flow is manually throttled at the wellhead to maintain an adequate water column above the pump suction.

Pump Experience. Table 4-8 summarizes the pump experience as of 30 June 1987. The "condition" column of this table describes the condition of each pump upon its removal. Some of these pumps suffered multiple "fatal injuries", making it difficult to assign root causes of failure.

In the beginning, pumps at Mammoth Lakes were installed under the direction of Johnston, and used the standard 1.5 gpd (5.7 L/day) lubrication rate. However, by mid-1985, Mammoth-Pacific had trained their own personnel to direct the installation of pumps, using a service crew out of Bakersfield. They had also concluded that the standard lubrication rate was not adequate for their resource, and had doubled the oil feed rate.

Table 4-6

MAMMOTH-PACIFIC WELL COMPLETIONS

Casing	MBP-1	Well MBP-3	MBP-4	MBP-5 (Spare)
Conductor				
diameter (in)	20	22	22	22
top (ft)	-0-	-0-	-0-	-0-
bottom (ft)	approx. 100	118	46	62
Surface Casing				
diameter (in)	16	16	16	16
top (ft)	-0-	-0-	-0-	-0-
bottom (ft)	252	385	274	252
Liner				
diameter (in)	13-3/8	13-3/8	13-3/8	13-3/8
top (ft)	216	-0-	-0-	2
bottom (ft)	665	752	672	660
slotted interval (ft)	unknown	400-723	414-669	392-660
Other Information				
Total Depth (ft)	665	752	672	662
Probable Production Interval (ft)	625-665	691-752	413-443	not reported
Static Water Level (ft)	219	58	149	182
Pump Suction (ft)	612	612	602	597

Source: Mammoth Pacific Power Company.

Table 4-7

DESIGN CHARACTERISTICS OF THE PUMP USED AT MAMMOTH LAKES

Pump	Characteristic
Manufacturer	Johnston
Model	12 EMC
Length	13 feet
Diameter	11.75 inches
Stages	13
Setting Depth (nominal)	600 feet
Speed	1770 rpm
Power	250 hp
Rated Efficiency	77%
Rated Capacity	
- flow	800 gpm
- head	300 psig
Lineshaft	
Shaft Diameter	1.6875 inches
Shaft Material	Steel
Bearing Diameter (internal)	1.691 inches
Bearing Length	4.00 inches
Bearing Spacing	5 feet
Bearing Material	SAE67 bronze
Motor	
Type	HSRGZCH
Voltage	4160 V
Power Rating	250 hp

Source: Mammoth Pacific Power Company.

Table 4-8

SUMMARY OF LINESHAFT EXPERIENCE AT MAMMOTH LAKES

Radian Ident No[a]	Well Ident	Pump Model[b]	Setting Depth (ft)	Prod Temp (°F)	Date Installed	Date Failed/ Removed	Approx. Endurance Rotating Hours	Condition
2-J-ML-1	MPB-4	12EMC-11	600	350	3/16/84	12/20/84	303	FAILED: Pump pulled and well worked over Nov. and Dec. 1984. Pump reinstalled 12/13/84. Failed 12/18/84. Broken enclosing tube.
2-J-ML-2	MPB-5	12EMC-11	600	318	5/30/84	1/15/85	246	FAILED: Pump pulled and well worked over 1/85. Lineshaft parted at coupling during installation (over-tensioned lineshaft). Pump sand-locked.
2-J-ML-3	MPB-1	12EMC-11	600	331	7/30/84	5/7/85	20,185	FAILED: About 25% of bearings in bottom 1/4 of lineshaft severely worn. Three brigs parted. PUMP: Severe wear throughout pump. Three impellers loose on shaft, but pump produced to end. Surmised that wear allowed impeller/bowl interference.
2-J-ML-4	MPB-4	12EMC-11	600	345	1/13/85	9/13/85	4236	FAILED: Multiple parting of enclosing tube bearings. Some bearings split, some unscrewed. Pump full of sand and severely eroded.
2-J-ML-5	MPB-3	12EMC-11	600	341	2/12/85	11/24/86	5364	FAILED: Heavy wear and corrosion of bottom third of lineshaft and bearings. Pump severely eroded by sand.
2-J-ML-6	MPB-5	12EMC-11	600	332	2/21/85	3/11/86	2433	FAILED: Pump impellers damaged by falling centralizer pin. Severe sand erosion. Lineshaft OK.
2-J-ML-7	MPB-4	12EMC-11	600	345	9/25/85	2/26/86	3487	FAILED: Bottom half of lineshaft unlubricated. Severe wear. Nine of 11 impellers broken and loose from shaft.
2-J-ML-8	MPB-3	12EMC-11	600	340	12/17/85	-	12,387[c]	Running 6/30/87
2-J-ML-9	MPB-4	12EMC-11	600	346	3/11/86	-	10,934[c]	Running 6/30/87
2-J-ML-10	MPB-5	12EMC-11	600	337	3/13/86	-	1867[d]	In service as a spare

[a] 2-J-ML-N. 2 = second survey. J = Johnston pump. ML = Mammoth Lake. N = Sequence number

[b] All pumps have uncoated 17-4PH shafts and boride-hardened steel bearings.

[c] As of 6/30/87

[d] As of 5/31/87

Well MBP-1 was the only well not sand-packed and redrilled. The first pump was put into this well in July 1984. Upon startup, it failed to rotate. Inspection upon removal showed that one of the impellers was loose on the shaft. The pump was repaired and reinstalled and ran for a benchmark 20,185 rotating hours before finally failing on 7 May 1987. Failure was signaled by sudden onset of fluctuating amperage and lubrication oil injection pressures. This run was by far the longest single pump endurance reported at any domestic site to date.

Inspection of the lineshaft showed that about 25% of the bearing/bushings in the bottom seven joints of the 30-joint lineshaft were severely worn, and three had parted (fractured). The bearings in the upper lineshaft were in much better condition.

All parts of the pump were heavily worn, with marked erosion of the impeller vanes and shrouds, the diffusers, and wear rings. Several of the impellers were loose on the shaft, but the pump had operated normally right up to the moment of failure. It was surmised that bearing and wear ring wear had accumulated to the point that the impellers began to contact the bowls, knocking some of the impellers loose and triggering the symptomatic failure.

The first pump was installed in MPB-3 in February 1985. After 5364 hours of operation, the wellhead seal failed on 23 November 1985. The seal was replaced, but the pump was "noisy and vibrated". It was pulled on 5 December. Inspection showed that the bottom 160 feet (49 m) of the lineshaft bearings were corroded and the lineshaft was scored, apparently due to inadequate lubrication. The pump suffered severe sand erosion, requiring replacement of the pump shaft, four bearings, ten bowl wear rings and matching impeller wear rings, one bowl, and one impeller.

The next pump (2-J-ML-8) was installed in this well in December 1985 by a crew supervised by Mammoth-Pacific. This pump received 3 gpd (11.4 L/day) oil lubrication, twice the recommended rate. It was running as of 30 June 1987, and had accumulated 12,387 rotating hours.

Well MBP-4 perhaps best illustrates the early problems caused by the residual sand from the well workovers. Pump 2-J-ML-1 was installed in this well in March 1984, prior to the workover. It was removed, the well was repaired, and the pump was reinstalled. Brine production recommenced on 13 December 1984. High vibration and wellhead seal failure occurred on 18 December. The lube oil pressure tracked the brine production pressure, indicating a leak in the enclosing tube. Inspection showed that the enclosing tube had been pulled apart in two places, a failure which Mammoth-Pacific blamed on over-tensioning by the Johnston Pump installation crew. Brine had entered the enclosing tube, causing failure of about half of the lineshaft bearings. The pump showed erosion. The pumpshaft, one bearing, 12 wear rings, and 12 O-rings were replaced.

The next pump (2-J-ML-4) was installed in this well in January 1985, and ran 4236 hours prior to failure on 12 September 1985. The enclosing tube in joints 22 through 27 (from the surface) had parted. Some of the bearings, which also serve as couplings, had split while others had unscrewed. "Etching-like galvanic attack" was observed on the shaft under the bearings throughout the lineshaft. The pump, "full of sand" with severe erosion, was scrapped.

Pump 2-J-ML-7 was installed in this well on 25 September and ran 3487 hours. On 24 February 1986, the lube oil pressure rose sharply, causing failure of the wellhead seal. The seal was replaced the next day, but the pump ran only a few minutes before the discharge pressure dropped to zero. Inspection showed that the bearings in the bottom half of the lineshaft were unlubricated with excessive bearing wear and scoring of the shaft. In the pump, 9 of the 11 impellers had broken hubs. There was some sand erosion. While it is not possible to define the cause of the broken hubs, sudden sand-locking could explain the failure.

The next pump (2-J-ML-9) was installed in March 1986 by a crew directed by Mammoth-Pacific. This pump was operated with doubled lubrication (3 gpd or 11.4 L/day), and had logged 10,934 hours as of 30 June 1987.

The history in well MBP-5 is similar except that this well serves as a spare. The first pump (2-J-ML-2) was installed in May 1984 and apparently operated for 246 hours prior to removal for the well workover. The pump was reinstalled in January 1985. During installation, the Johnston-directed crew over-tensioned the enclosing tube, stripping the threads from one of the lineshaft bearing/couplings. The lineshaft was repaired and the pump was installed yet a third time, but would not rotate. Inspection showed that the pump was "choked with sand" and had nine broken hubs. It was scrapped.

The next pump (2-J-ML-6), installed in February 1985, operated for 2433 hours. In March 1986, the current began fluctuating and the lube oil pressure rose to 500 psig (3.6 MPa). The pump was removed on 13 March and was found to be severely eroded.

In addition, the vanes of the top two impellers had been chipped by a pin which had fallen from a centralizer.

Pump 2-J-ML-10 was installed in March 1986. Due to the spare duty of this well, this pump had accumulated only about 1867 hours as of 30 June 1987.

Comparison of the run-to-date performances of 2-J-ML-8 and 2-J-ML-9 with previous pumps in the same wells shows a major increase in pump endurance, though the magnitude of this increase will not be known until at least one of these pumps fails. Mammoth-Pacific attributes the improved longevity to:

- Reduced sand production, i.e., the workover sand has finally been pumped out of the wells;
- Greater care in pump installation, including their assumption of responsibility for directing the installing crew; and
- Increased lineshaft lubrication.

Chevron at the Heber Binary Plant, Heber, California

In this report, reference to Chevron and Heber always relates to the wells supplying the Heber Geothermal Binary Power Plant. Confusion can occur when considering this site, since Chevron is also the brine supplier for a flashed-steam geothermal plant also located on this resource.

Resource Characteristics and Pump System Design. Chevron declined to provide much detail about the resource at Heber. The producing formation is extremely thick and the continuing production of sand from certain wells suggests that the formation is composed of sandstone or mixed sandstone and shale. Chevron does indicate that brine flow results from inherent permeability rather than fracture flow.

The well field for the binary plant is being developed in phases. This update includes data generated by the Phase I development only. Any experience from Phase II became available too late to be included in this report. Chevron designates the Phase I wells as 101, 102, etc. There are seven Phase I wells. The phase I wells are cased with 13 3/8-inch casing to somewhat deeper than 1000 feet (305 m). The remaining depth is cased with 9 5/8-inch casing which is perforated adjacent to the targeted producing zones.

Chevron has divided the producing formation into different production zones, but states that this division is somewhat arbitrary since productivity and chemistry are homogeneous throughout the well field. Zone I is nominally the layer from 2000 through 4000 feet (610 through 1220 m) while zone II extends from 4000 feet to 6000 feet (1220 to 1830 m).

Table 4-9 summarizes the chemistry of the blended brine delivered to the binary plant. Chevron indicates that, with the exception of suspended solids, the brine chemistry is essentially identical from well to well. Well 106 produces large amounts of sand. Wells 103 and 105 produce minimal amounts of sand, and the other wells probably produce intermediate amounts. Chevron notes that there is a surge of sand production each time a pump is turned on.

All of the Phase I pumps began as standard Peerless lineshaft pumps, though they have been extensively modified as will be discussed below. These pumps have oil-lubricated lineshafts with SAE 67 bronze insert/steel sleeve bearing/couplings every five feet. The lubricating oil is vented into the pump suction rather than into the annular space between the casing and the production column.

Chevron began with two models of Peerless pumps, designated as 12HXHG and 12HXLG. Performance curves for these pumps are located in Appendix C. Chevron designated these as "Zone I" and "Zone II" pumps. Their characteristics are highlighted below:

- Zone I (12HXHG). Design flow of 1650 gpm (104 L/s) at a total discharge head (TDH) of 408 psia (2.8 MPa) at a rated efficiency of 77% from 20 stages. The high flow rates of these pumps were obtained by minimizing the thickness of all internal elements.

- Zone II (12HXLG). Design flow of 1320 gpm (83 L/s) at a TDH of 408 psia (2.8 MPa) at a rated efficiency of 73% from 18 stages. These pumps have more robust impellers and diffusers.

The original plan was that zone I pumps would be installed at 750 feet (229 m) in wells perforated in zone I, while zone II pumps would be set at 1000 feet (305 m) in wells perforated in zone II. However, this concept was abandoned fairly early, with subsequent pumps being set at 1000 feet (305 m) regardless of which zone was perforated.

Both types of pump are driven by the same type of motor: General Electric vertical TEFC 600 hp, 4160 V motors running at 1790 rpm.

Pump control is manual. Chevron attempts to maintain a wellhead pressure of 200 psig (1.5 MPa).

Table 4-9

PROPERTIES OF BLENDED[a] PRODUCED BRINE FROM HEBER BINARY PLANT PRODUCTION FIELD
(ppm except pH)

Component	Mean	SD	N	Range
pH	5.7	0.1	17	5.6 - 5.9
Chloride	8132	75	20	8000 - 8250
Sulfate	58.3	6.9	18	50 - 69.8
Total CO_2[b]	230	41.9	20	154 - 290
Total H_2S[c]	2.0	0.97	20	0.9 - 5.5
Total NH_3[d]		Not reported		
Calcium	894	14	20	876 - 928
TDS	14,000	510	19	13.160 - 15,272
TSS	10.6	8.4	9	5.8 - 26
Gas Breakout Pressure		Not available		

Data Provided by SDG&E

[a]Mixed brine from several wells

[b]As CO_2

[c]As H_2S

[d]As NH_3

Pump Modifications. The original pumps were configured like the "Peerless Geothermal Pump" described in Section 3. They had small sand lugs and bronze pump bearings against tungsten carbide hardfaced journals. The original pumps were assembled in two sections of nine or ten stages each, and coupled together in the field with a Peerless tandem adaptor. The tandem adaptor is a split case device housing a threaded coupling which joins the two pump shafts.

There have been three major design and materials selection modifications to the Phase I pumps at Heber, leading, in conjunction with the two general pump types listed above, to a large number of pump variations. Table 4-10 summarizes the experiences with these different pump modifications as of 31 March 1987. The pumps which were running as of that date were still running as of 30 June 1987. Theoretically they could have logged an additional 2184 hours, but the actual number of additional hours is less. Unfortunately, it was not practical to further update the actual rotating hours.

For clarity in presenting this information, these design variations are described in this table as defined below.

The three design modifications are:

- D1 (original design). Pump shop-assembled and transported in two sections. Sections coupled together in the field via threaded coupling housed in standard tandem adaptor. Small sand lugs with narrow gap between outboard face of lugs and mating bowl wall.

- D2 (transition design). Pump assembled in two sections and coupled in the factory with threaded coupling housed in a "dummy bowl", a bowl from which the internals had been machined. Unitized pump inserted into length of casing for transport and lifting at the site. More robust sand lugs which extended flush with the mating bowl walls. These pumps also had much more rigorous assembly quality control than the D1 pumps. This design was strictly a matter of expediency while the final design was being brought into production.

- D3 (final design). Pump assembled in two sections and spliced without threaded coupling in the factory. This shaft splice maintains the even spacing of the bearings and leaves no segment of the shaft unsupported. The unitized pump is transported and lifted inside a length of casing. More robust sand lugs which extend flush with the mating bowl walls. These pumps also have much more rigorous quality control than did the D1 pumps.

There have also been three major materials selection changes.

- M1 (original specification). Type 416 pump shafts with tungsten carbide hardfaced journals running against SAE 67 bronze bearings. Type 410 or 420 (interchangeable) wear rings on the impeller skirts. No wear rings in the bowl throats.

Table 4-10

SUMMARY OF CHEVRON PUMP EXPERIENCE AT THE HEBER BINARY PLANT

Radian ID No.[a]	Chevron ID No.	Pump Type[b]	Setting Depth	Date Installed	Date Removed	Endurance Rotating Hrs	Condition/Cause of Failure
2-P-HB-1	104-1	I D1M1	720	3-13-85	1-22-86	2419	FAILED: Damage concentrated at suction and near tandem adaptor. Bearings missing from stages 5-15. Impellers 7-15 shattered. Severe impeller wear in vicinity of tandem adaptor. Severe sand erosion around sand lugs. Sand plugging stage 16 to outlet, also some gravel up to 0.5 inch (13 mm) dia.
2-P-HB-2	106-1	I D1M1	720/840[c]	4-19-85	11-20-85	926	FAILED: Severe spalling of shaft hardfacing. Severe bearing wear in vicinity of tandem adaptor. Bearings missing from first two stages. Severe sand erosion of collets, around sand lugs and into walls of matching bowls. Severe wear of 4 impeller rings and matching throats. Numerous impeller cracks, but impellers known to have been cracked prior to installation. Pump shafts were crooked when installed.
2-P-HB-3	102-1	II D1M1	740	5-11-85	7-3-86	4064	FAILURE IMMINENT: Shaft hardfacing spalling. Mild bearing wear. Erosion of collets. Pitting of shafts attributed to acidation of well. Impellers of upper shaft topped out damaging diffusers. Y-267 O-rings excellent. Inspector concluded that pump would have failed in a few hundred more hours.

Table 4-10 (Continued)

SUMMARY OF CHEVRON PUMP EXPERIENCE AT THE HEBER BINARY PLANT

Radian ID No.[a]	Chevron ID No.	Pump Type[b]	Setting Depth	Date Installed	Date Removed	Endurance Rotating Hrs	Condition/Cause of Failure
2-P-HB-4	103-1	II D1M1	740/1000[c]	5-28-85	1-9-86	2496	FAILED: Severe wear and secondary damage to impellers, bowls, bearings, and shaft in vicinity of tandem adaptor. Spalling of shaft hardfacing throughout. Bearings 3-13 missing. Fatigue fracture of pumpshaft coupling. Excessive shaft run-out at tandem adaptor and pump/lineshaft coupling. Severe wear of impeller rings and matching throats in vicinity of tandem adaptor. Impellers had topped out, several broken.
2-P-HB-5	105-1	I D1M1	720	6-1-85	7-5-86	6194	FAILURE IMMINENT: Shaft hardfacing spalled and worn. Wear rings worn. Little sand or erosion. Y-267 O-rings excellent. One sand lug broken from each of 3 bowls. Inspector concluded that pump would have failed in a few hundred more hours.
2-P-HB-6	101-1	II D1M1	740/1000[c]	6-6-85	2-6-87	2521	FAILURE: Similar to 2-P-HB-2, with most of the damage in the first three stages. The bearings were missing from these stages and the entire tailbearing and support were broken away. The first stage impeller was broken. Impeller wear rings and matching throats of first two stages severely worn. Little wear elsewhere. Shaft hardfacing poor throughout, with matching bearing wear concentrated in aligned 120^{o} segment. Impellers of upper shaft had topped out, damaging the diffuser suctions.

Table 4-10 (Continued)

SUMMARY OF CHEVRON PUMP EXPERIENCE AT THE HEBER BINARY PLANT

Radian ID No.[a]	Chevron ID No.	Pump Type[b]	Setting Depth	Date Installed	Date Removed	Endurance Rotating Hrs	Condition/Cause of Failure
2-JR-HB-7	106-2	I D1M1	1000	12-3-85	1-4-86	584	FAILED: Shaft hardfacing spalled and flaking, severe bearing wear with bearings 7-14 missing. Severe shaft wear (about 0.06 inch or 1 mm) in vicinity of tandem adaptor. Severe wear of impeller wear rings and matching bowl throats. Severe erosion of collets, around sand lugs. Deep grooving of bowl shirts adjacent to sand lugs. Massive secondary damage to impellers. Top of upper shaft bent with 0.007 inch (0.2mm) run-out.
2-P-HB-8	106-3	II D1M1	1000	1-8-86	2-18-86	719	FAILED: Condition similar to 2-P-HB-2 and 2-JR-HB-7
2-P-HB-9	103-2	II D2M1	1009	1-9-86	-	6925+[d]	Running 6-30-87.
2-PR-HB-10	104-2	I D2M2	1000	1-27-86	1-1-87	4389	FAILED: Coupling in lower lineshaft fractured due to fatigue. Bearings in vicinity worn away to steel sleeves.
2-JR-HB-11	101-2	II D3M3	1000	2-10-86	2-11-86	2	FAILED: "Tolerance accumulation" resulted in seizure of the top pump bearing/pump shaft.
2-JR-HB-12	106-4	I D3M3	1000	2-24-86	-	5950+[d]	Running as of 6-30-87.
2-PR-HB-13	101-3	II D2M3	1000	3-23-86	-	unknown[e]	Running 6-30-87.
2-JR-HB-14	105-2	I D2M2	1031	7-11-86	-	3081+[d]	Running 6-30-87.
2-PR-HB-15	107-1	II D3M3	1060	7-21-86	-	3269+[d]	Running 6-30-87.
2-JR-HB-16	102-2	II D3M3	1069	8-4-86	-	2779+[d]	Running 6-30-87.

Table 4-10

SUMMARY OF CHEVRON PUMP EXPERIENCE AT THE HEBER BINARY PLANT

Radian ID No.[a]	Chevron ID No.	Pump Type[b]	Setting Depth	Date Installed	Date Removed	Endurance Rotating Hrs	Condition/Cause of Failure
2-JR-HB-17	104-3	I D3M3	approx 1000[f]	1-7-87	3-31-87	1300	FAILED: Inspection showed that one of the bowls had been cracked by the installation tongs. There were deep tong marks "all over the bowl". After 1300 hrs. a piece of the bowl broke away. Pump showed <2 mils bearing wear.
2-JR-HB-18	104-4	I D3M3	approx 1000[f]	4-2-87	-	1545	Running 6-30-87.

[a]Pump Code
2 - X - HB - N
2 = second survey
X = Origin, P = Peerless, PR = repaired by Peerless, JR = repaired by Johnston
HB = Heber binary plant well field
N = Sequence number

[b]Pump reset to second depth.

[c]See text for explanation of descriptor code

[d]Hours shown are as of 3-31-87. However, pump was running as of 6-30-87 and could have accumulated as much as 2184 additional hours. (It was not possible to definitely update the run times to 6-30-87.)

[e]Run time not available

[f]Inferred

- M2 (transition specification). Uncoated 17-4PH shaft running against SAE 67 bronze bearings. Type 410 or 420 impeller wear rings against 17-4PH throat wear rings (liners).

- M3 (final specification). Uncoated 17-4PH shafting running against boride-hardened steel bearings. Type 410 or 420 impeller wear rings against 17-4PH throat wear rings.

Initial experiences. The first D1M1 pump was installed in well 105 on 13 March 1985. By the end of February 1986, six D1M1 pumps had failed, three of them in well 106 which produces lots of sand. From failure analyses of these pumps, Chevron determined the following failure mechanism.

- Sand enters the pump bearings and creates extreme point loads, resulting in spalling of the hardfacing. The spalled hardfacing then destroys the soft bronze bearings.

- As the bearings are worn away, the impeller skirts contact the bowl throats, resulting in rapid wear of both elements.

- If the pump is run long enough, the impellers strike the bowls, resulting in breakage of the impellers.

- Wholesale destruction of the pump results from fragments of the impellers.

Chevron also identified the following contributory factors:

- Poor alignment of the pump shafts by the tandem adaptor frequently resulted in severe bearing wear in that area.

- Some of the vanes of some of the original impellers were cracked during or prior to pump assembly.

- Engineering analysis indicated that under standard specifications and quality control practices, "tolerance accumulation" could lead to mechanical interference.

- The impellers can "drift" up or down in the pump over time. Topping or bottoming results in severe damage.

Pump Improvement Program. In the face of these poor early results, Chevron began a pump improvement program. Ultimately this program contained four main elements:

- Pump modification;

- Enhanced assembly quality control;

- Semi-continuous performance monitoring; and

- Improved installation practices.

The first element, <u>pump modification</u>, corrected certain problems specific to the originally supplied pumps. By early 1986, after consulting with both Peerless and Johnston, Chevron developed the D3 design and M3 materials selections, but the lead time required to make these modifications meant that less desirable transition designs (D2M2 and one D2M1) had to be used to fill the gap.

The remaining three elements constitute practical life-extension practices which Chevron indicates will be applied to all future pumps, regardless of manufacturer. These pump life extension practices will likely be of great benefit to other geothermal operations.

The first element of the pump life-extension practices is extremely <u>stringent quality control requirements on pump assembly</u>, requirements in some ways exceeding those for class II nuclear pumps. The Chevron QC protocol contains the following elements:

- Independent QC personnel present to witness and certify all pump disassemblies and assemblies.
- Chevron engineers and manufacturer agree prior to assembly regarding critical dimensions and tolerance limits. Chevron engineers confirm that tolerances are so controlled that "tolerance accumulation" interference is precluded.
- All parts, not just a statistical sample, are inspected.

<u>Pump repair</u>

- Clean all potentially reusable parts and dye penetrant test each impeller. REJECT any impeller with cracks.

<u>All pumps</u>

- Fit wear ring over impeller skirt. Wear ring will have 32 RMS or better surface finish. Interference fit will be not less than 0.003 inches (0.076 mm).
- Set up impellers on straight mandrel with match-marked collets. Check impeller run-out and correct as necessary.
- Dynamically balance each impeller in two planes to less than 1.0 gram-inch residual imbalance per plane. Inspect each impeller after balancing. REJECT any which were ground too thin during balancing.
- Repeat dye penetrant test of each impeller to verify that none were cracked by the above steps.
- Fabricate pump shaft, check shaft run-out and straighten if necessary to achieve straightness within 0.005 inch (0.127 mm) total indicated run-out (TIR) over the entire shaft length and not to exceed 0.0005 inch/ft (0.04 mm/m).

- Install shrink-fitted throat wear ring (liner). Wear ring must have 0.003 inch (0.076 mm) minimum interference fit.

- Measure all wear parts to ensure that they are within tolerances and that there is adequate clearance. Document all dimensions for future reference.

- Assemble pump. Dye penetrant test the impellers again after they are mounted on the shaft to verify that they were not cracked by the impact wedging of the collets. Check available axial end-play every three or four stages.

- Check run-out of spliced pump shaft. Reject if runout exceeds 0.0005 inch/ft (0.04 mm/m). Check for free rotation.

The second element of the pump life-extension practices is a novel pump performance monitoring program which allows diagnosis of pump problems before serious secondary damage has occurred. Chevron back-calculates the total discharge head from the wellhead conditions and uses this information in combination with the flow rate to determine the actual pump efficiency. The ratio of the actual vs. pump curve efficiency is the "performance index" or PI. Due to inaccuracies in the various measurements, the PI for a pump in good condition will fluctuate randomly from 95 to 105 percent. However, Chevron has found that this index is essentially stable over long periods of time, as long as the hydraulic parts of the pump are not damaged and the impellers are properly engaged (spaced-out) within the pump. Once the PI begins to deteriorate, it will fall rapidly until the pump is destroyed.

Chevron monitors the pump PI's approximately daily. Whenever there appears to be any downward trend in PI, Chevron takes the pump off line and performs backspin and axial end-play tests. After the pump is shut off, the water column in the production column falls and drives the pump into a backspin. The time required to complete the backspin and the maximum RPM produced are good indicators of the hydraulic and mechanical conditions of the pump. Chevron performs a backspin test shortly after pump installation to establish a baseline for that pump. They also try to perform backspin tests whenever there is an orderly shutdown of a pump.

The final test is determination of the axial spacing of the impellers. As discussed previously, Chevron has found that there is a tendency for the impellers to migrate upward in the bowls, perhaps as a result of further tightening of the lineshaft joints during operation.

The pump designated as 2-PR-HB-10 in Table 4-10 provides a fine example of the benefit of this monitoring program. The PI of this pump exhibited a downward trend after 2058 hours. The backspin time was only 115 seconds, compared to the 130 second baseline. The axial end-play test showed that the impellers were only 11/16 inches (1.75 cm) from the top, compared to the normal 2.0 inches (5.08 cm). After readjustment, the PI returned to normal and the pump ran for a total of 4389 hours. If this condition had remained uncorrected, it is reasonable to assume that failure would have occurred within a few hundred more hours (of the 2058 hours when the trouble was detected). Thus the monitoring program about doubled the life of this pump.

This performance monitoring program also provides strong evidence that standing by has little detrimental effect. Due to the shakedown nature of Heber binary plant operations during the time covered by this survey, several pumps experienced shutdowns of more than two months. Chevron was able to detect no change in PI as a result of even prolonged shutdown.

When a pump demonstrates falling PI and shortened backspin time, but has normal impeller spacing, mechanical difficulty is indicated. The operator must decide whether to pull the pump at that point or run to complete failure. Pulling the pump at the earlier point frequently results in much higher salvage value, since the pump does not reach the catastrophic breakup stage.

The final element of the pump life-extension practices is <u>particular care during pump installation</u>. Chevron uses their own personnel to oversee installation of pumps. Though it is possible to pull and install a pump in seven days, provided that a pump and rig are available, Chevron has added checks and tests which increase the turnaround to eight days.

The pumps are delivered to the site inside lengths of transport casing with welded-on handling tabs. The pump is lifted into place on the rig inside the transport casing to avoid bending or other damage to the pump. Chevron's design cannot accommodate a separate bubbler tube because Chevron runs 10 3/4-inch production column in the 13 3/8-inch casing. Chevron therefore uses 0.25-inch (6.35 mm) stainless steel tubing strapped to the production column as the pump is lowered into the well. Chevron experienced early problems with plugging of the inlet to the bubbler tube by scale derived from brine or kill fluid. Mechanical damage was rare. Chevron corrected this problem by switching to a 9 5/8-inch production column for the first section above the pump. They use 1-inch (25.4 mm) tubing in this area. They also purge the tubing as soon as the installation is completed to minimize scale formation in the bubbler tube.

Chevron applies tension to the enclosing tube equal to 230% of the weight of the enclosing tube and bearings. The manufacturer had recommended tension equal to 110%, but Chevron observed some eccentric lineshaft and bearing wear at this tension.

When installation is complete, Chevron produces the pump for at least two hours. They then perform a baseline backspin test and check the impeller setting within the bowls. After these tests are completed and the pump has demonstrated an acceptable PI, it is ready to supply brine to the plant.

Results of the Pump Improvement Program. The results of the pump improvement program are presented graphically in Figure 4-1. Two of the improved D3M3 pumps have failed. Pump 2-JR-HB-11, the first D3M3 installed, failed after two hours due to seizure of the top pump bearing. Chevron concluded that though the bearing and shaft were both within specification, the tolerance allowances were such that the shaft was bound. Chevron imposed tighter shaft and bearing tolerance limits on all future pumps. Pump 2-JR-HB-17 failed after 1300 hours. Inspection showed that one of the bowls had been broken by the tongs during installation. The bowl had numerous tong marks and a "sizable" hole in the wall.

No other D3M3 pump has failed as of 30 June 1987. The contrast in well 106 is the most dramatic. The first three pumps in this well were D1M1 units. All failed in less than 1000 hours. The first D3M3 in this well had run almost 6000 hours as of 31 March 1987, and was still running as of 30 June. Thus, this pump has accumulated close to 8000 hours. This clearly demonstrates that the D3M3 modification, combined with the pump life-extension practices, is extremely resistant to sand damage. Thus, this appears to be a major advance in pump durability, since sand production is a fact of life in many wells.

Figure 4-1 also shows that the original D1M1 design can give credible endurances in clean wells if all of the factors discussed above happen to come together correctly. Bronze bearing technology coupled with enhanced quality control and shop-aligned shaft couplings gave even better results in clean wells. However, this bearing technology appears highly vulnerable to sand.

In a broader context, early failure has continued to plague pump operations at other sites. However, the innovations brought together by Chevron would appear to offer a good chance of eliminating most, if not all, early failures. If this impression is correct, it will have a major beneficial effect on the average pump life.

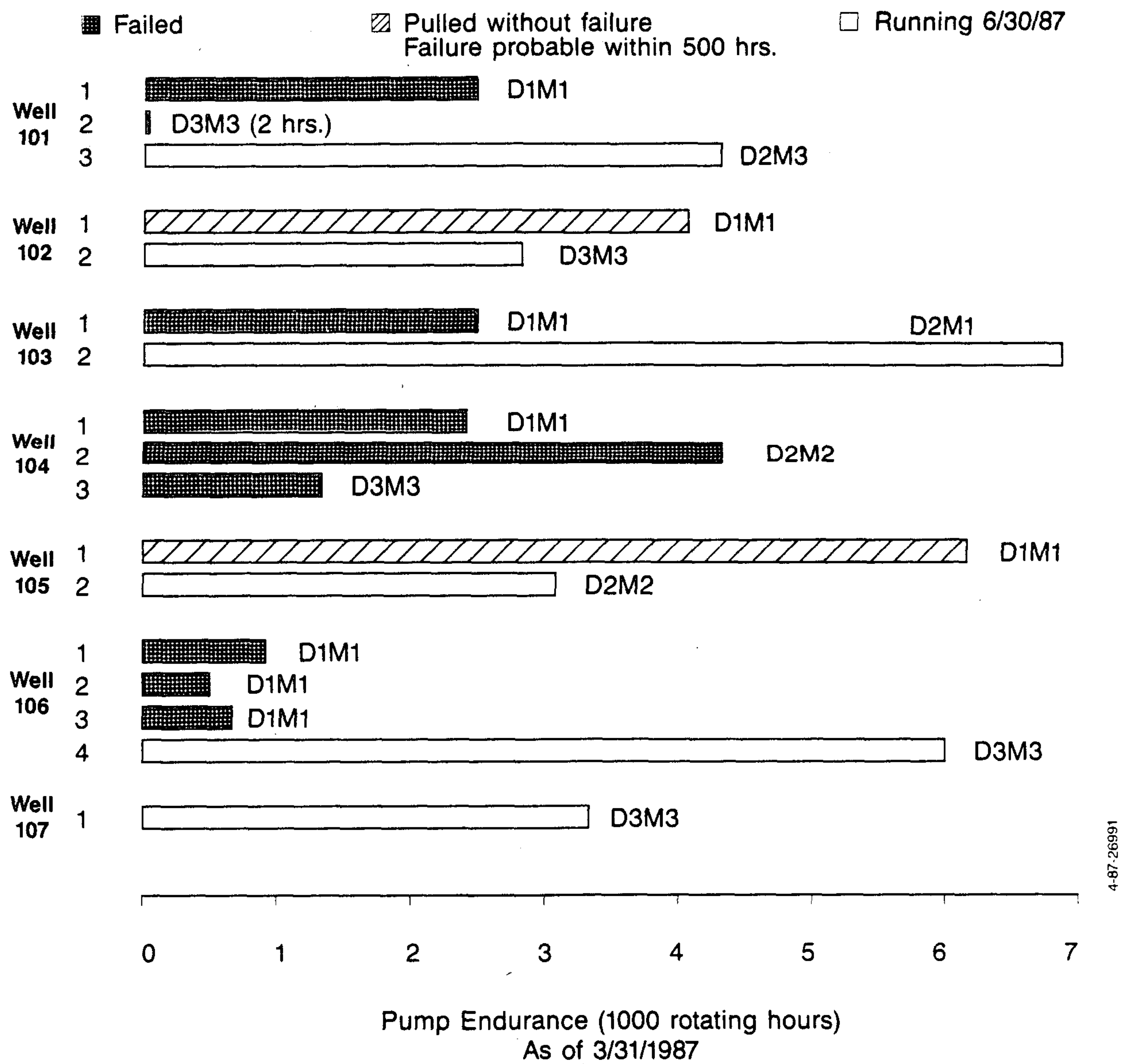


Figure 4-1. Endurances of brine production pumps at the Heber Binary Plant. (Refer to text for discussion of the Design "D", and Materials "M" designation associated with each pump).

Lineshaft and Seal Experience. Chevron has used oil lubricated enclosed lineshafts for all of its pumps. As is typical, the lineshaft is assembled in 20 foot (6 m) lengths with bearings every 5 feet (1.5 m). These bearings also serve as couplings for the enclosing tube. Chevron has opted for steel-sleeved bearings with SAE 67 bronze inserts.

Originally, Chevron specified Type 416 stainless steel for the lineshafts, but they concluded that the corrosion resistance was not needed. They graduated to 4130 and 4140 steel, which gave good results. However, Chevron concluded that these high strength alloys had more "memory" than was desired. They now prefer 1035 carbon steel, but will use the 4130/4140 if 1035 is not readily available.

Chevron has reported only one lineshaft failure. In this pump (2-PR-HB-10), a lineshaft coupling in the lower part of the lineshaft fractured by fatigue. The bearings in this vicinity were worn away, probably contributing to the stresses which caused this failure. At the time of this writing, Chevron was investigating the exact cause of this failure.

Chevron originally used John Crane 8B-1 cartridge mechanical seals to provide dynamic sealing of the lubricating-oil-filled annulus at the wellhead. Chevron experienced numerous prompt failures with this seal arrangement. They converted to a Borg-Warner single seal with a pumping ring in the seal assembly to promote flow around the seal cavity. This seal system has apparently been modified by the addition of a seal lip within the seal assembly. This lip prevents the eruption of hot oil in the event of seal failure. Chevron has reported success with this seal system and expects about one year seal life.

Chevron has used both Viton and EPDM Y-276 for the static bowl seals. EPDM Y-267 was developed by the Department of Energy specifically for high-temperature geothermal sealing service, and has demonstrated remarkable resistance to hot brine, oil, and isobutane. EPDM Y-267 bowl O-rings retained excellent properties after more than 10,000 hours in the 350^{o}F (177^{o}C) Heber brines.

Many of the Heber pumps had Viton O-rings (Viton is the industry standard O-ring material) due to difficulties in obtaining EPDM Y-267 O-rings. No failures of the Viton O-rings were reported.

LINESHAFT PUMP CONCLUSIONS

The interval since the original survey has seen a steady progression of pump developments, with numerous one-or-few-of-a-kind pump installations. Statistical examination of the data showed that any attempt to generate a quantitative cause of failure distribution by pump design variation would likely be misleading due to the small population of each discrete pump design or material selection modification. The difficulty in generating a frequency distribution of the causes-of-failure was further compounded by the fact that many of the pumps had been run to complete destruction, making assignment of specific causes of failure to each pump problematical. For these reasons, it was not possible to generate a cause-of-failure frequency distribution analogous to Table 4-1.

During the development of high-capacity, high-temperature geothermal lineshaft pumps, numerous problems have been encountered and surmounted. Table 4-11 highlights these problems and the solutions which have been employed to overcome them.

Section 5 will evaluate the lineshaft pump experiences for statistical trends, and conclusions concerning near-term pump life will be developed in that section. The following qualitative conclusions can be drawn from this and the prior survey.

- At the close of the original survey in 1983, the average endurance of lineshaft pumps was less than 1000 hours. Most failures involved the lineshaft and its bearings. At that time, improvements in lineshaft technology had been made which indicated that the major lineshaft problems had been mitigated. Available information indicated that pump bearing problems would be the next hurdle.

- Since the previous survey, lineshaft failures have continued to occur on a sporadic basis. Failure has generally been trackable to faulty installation or improper tensioning of the enclosing tube. Under- and over-tensioning can both cause failure. Plain 1035 carbon steel has emerged as the lineshaft material of choice.

- Modified Peerless pumps, and later Johnston pumps, were introduced in the early 1980's. Bronze pump bearing technology was identified as one of the weak points, and Johnston introduced proprietary boride-hardened steel bearings in 1983. These improvements have lead to an upward trend in longest pump endurances, but infant mortalities and short run lives have continued to plague operations, particularly in wells with high suspended solids.

- Recent pump development by Chevron at the Heber Binary Plant has shown that greatly enhanced quality control, coupled with the boride-hardened bearing technology, can greatly extend pump life, and perhaps more importantly, essentially eliminate infant mortality and other premature failures even in wells which are heavy sand producers.

Table 4-11

SUMMARY OF SIGNIFICANT HIGH CAPACITY GEOTHERMAL LINESHAFT PUMP DEVELOPMENT MILESTONES

Problem	Solution	Benefit
Lineshafts		
Open lineshaft bearings fail rapidly above about 320°F.	Open lineshafts replaced by enclosed lubricated lineshafts.	Mixed results.
Enclosed oil-lubricated lineshafts exhibit <1000 hr life due to excessive shaft whip, operation at critical speeds, and rapid wear of bronze bearings.	Bearings spacing and shaft design modified. Lineshaft tensioning increased to eliminate shaft whip. Oil lubrication rate set at 1.5 gpd.	Effective. Lineshaft failures greatly reduced. These modifications have become "standard."
Lineshaft assemblies damaged or broken during installation by manufacturer's crews.	Utility personnel assume supervision at pump installation	Effective where implemented.
Lower bearings in some lineshafts show inadequate lubrication at one site.	Lubrication rate increased to 3 gpd.	Effective where implemented
Fractures of reused lineshaft couplings.	Discard used lineshaft couplings.	Effective.
Parting (fracture) of solid bronze lineshaft bearings.	Problem not reported for steel-sleeved lineshaft bearings.	
Pumps		
Differential thermal expansion and lineshaft stretch cause impeller-bowl interference.	Axial end-play increased from 1.5 inches to 4 inches.	Effective. The revised axial end-play has become "industry standard."
Rapid erosion of parts in thrust-balanced impellers.	Use noncompensated impellers.	"Industry standard."
Bronze bearings with plain or hard-faced pumpshafts highly vulnerable to damage by sand.	Boride-hardened steel bearings introduced about 1983.	Effectively eliminated sand-induced bearing failure.

Table 4-11 (Continued)

SUMMARY OF SIGNIFICANT HIGH CAPACITY GEOTHERMAL LINESHAFT PUMP DEVELOPMENT MILESTONES

Problem	Solution	Benefit
Severe erosion of bowls by recirculating sand.	Recessed-throat pumps retrofitted with more robust sand lugs.	Effective.
	Raised-throat pumps introduced.	Effective.
Sand erosion and wear of impeller suction skirts and mating throats.	Wear-resistant wear rings or sleeves installed on both impellers and in throats.	Effective.
Extremely high sand loading "sand lock" pump.	No effective pump modification. Better well clean-up may prevent this problem.	
Field assembled inter-stage pump-shaft couplings provide inadequate alignment (problem limited to long pumps).	Shop-assembled interstage pump-shaft splices provide adequate alignment.	Effective.
Excessive drawdown results in gas breakout and consequential carbonate scaling at pump suction.	Deeper pumpsets and monitoring of drawdown to maintain adequate NPSH.	Effective.
Simultaneous production of incompatible brines produces "intrinsic" scaling.	No satisfactory solution as of 6/30/87.	
Factory specified tolerances inadequate for the severity of service, shown by "tolerance accumulation" failures and pump to pump endurance variation.	Much more stringent quality control including reduced allowances and 100% inspection of all components[a].	Effective where implemented.
"Migration" of impellers in pump bowls.	Semi-continuous monitoring of pump efficiency, allowing diagnosis of problem before significant damage to pump components occurs.[a]	Effective where implemented (doubled pump life).

[a]Part of Chevron's pump life-extension program.

- Standing by in a well appears to have little detrimental effect on pump endurance, provided that precautions are taken to keep brine from infiltrating into the enclosing tube. This can be done by maintaining oil pressure.

Section 5

STATISTICAL ANALYSIS OF LINESHAFT PUMP EXPERIENCE

Section 4 presented updated data on 72 lineshaft pumps, sufficient to justify a statistical evaluation. These 72 experiences were compiled into a database, presented in Table 5-1. This table is a combination of several tables in Section 4, with considerable condensation of some data. In Table 5-1, the pump experiences are grouped by pump model within each resource.

This section presents the results of a statistical evaluation of the updated lineshaft pump experiences. In this section, reference to pumps always includes the entire pump system. Likewise, failure refers to any event which forces removal of a pump from a well.

This analysis addresses the following issues:

- Search for trends;
- Estimation of pump life statistics based on recent operating practices;
- Estimated near-term pump life statistics based on implementation of the Chevron pump life-extension practices; and
- Effect on pump system availability and brine supply capacity.

ANALYSIS FOR TRENDS

A statistical analysis was made of the update experience from Magma Power, Mammoth-Pacific, and the OEM Peerless pumps at the Heber Binary Plant. Each of these sets of data was found to be statistically indistinguishable even at a relatively low level of confidence (80% interval). Note that selection of a low level of confidence increases the probability that the data sets will be shown to be significantly different.

For the remaining analyses, these three data sets were combined to form one larger set containing 64 endurances spanning the chronological interval from October 1981 through March 1987. This chronological interval was bisected and the endurances of the pumps installed in the earlier half were compared to the endurances of the pumps

Table 5-1

LINESHAFT PUMP DATABASE FOR STATISTICAL ANALYSIS

Site	Radian Ident. Number[a]	Well ID	Pump Manufacturer	Pump System Description			
				Pump Model-No. of Stages	Motor HP	Rated Capy (GPM)	Design Comments
Magma Power, East Mesa, CA	2-JP-EM-1	48-7	JRP[c]	12LB-16	350	1000	Developmental modifications of basic Peerless design. Evolving anti-sand geometry. Experimental brg materials.
	2-JP-EM-2	44-7B	JRP	12LB-16	-[d]	-	
	2-JP-EM-3	44-7A	JRP	12LB-16	-	-	
	2-JP-EM-4	48-7A	JRP	12LB-16	300	750	
	2-JP-EM-5	44-7	JRP	12LB-16	-	-	
	2-JP-EM-6	44-7A	JRP	12LB-16	-	-	
	2-JP-EM-7	44-7	JRP	12LB-16	-	-	
	2-JP-EM-8	44-7	JRP	12LB-16	-	-	
	2-JP-EM-9	48-7	JRP	12LB-16	350	750	
	2-JP-EM-10	48-7	JRP	12LB-16	350	700	
	2-JP-EM-11	48-7A	JRP	12LB-16	300	750	First trial of boride-hardened pump bearings
	2-JP-EM-14	48-7	JRP	12LB-14	350	600	Fully developed Johnston modifications of Peerless pump utilizing boride hardened steel bearing technology.
	2-JP-EM-16	44-7B	JRP	12LB-14	-	-	
	2-JP-EM-20	88-7	JRP	12LB-15	400	700	
	2-JP-EM-21	48-7A	JRP	12LB-16	350	750	
	2-JP-EM-12	61-7	Johnston	8CC-38	-	-	First true Johnston pump introduced at East Mesa.
	2-JP-EM-13	63-7	Johnston	8CC-33	125	350	"Slimline" with nominal 8-inch diameter. Features: boride hardened bearings; shop aligned interstage pumpshaft splice: "raised-throat" geometry.
	2-JP-EM-19	61-7	Johnston	8CC-33	-	-	
	2-JP-EM-27	83-7	Johnston	8CC-38	150	35	
	2-JP-EM-29	81-7	Johnston	8CC-38	150	327	
	2-JP-EM-30	81-7	Johnston	8CC-38	150	326	

(continued from left)

Operating Conditions						
Chemistry	Setting Depth (ft)	Prod. Temp (^oF)	Date Installed	Approx. Endurance[b] (hrs)	Condition Upon Removal	Radian Ident. Number
pH 5.5-5.80	906	350	10/81	3,920	FAILED: plugged by scale/seized	2-JP-EM-1
Cl 810-3657 ppm	-	360	12/81	7,321	FAILED: "worn out"	2-JP-EM-2
SO_4 56-112 ppm	-	336	1/82	1,440	FAILED: broken lineshaft collar	2-JP-EM-3
Total CO_2 194-323 ppm	906	350	9/82	9,104	FAILED: broken lineshaft	2-JP-EM-4
Total H_2S 0.5-1.5 ppm	-	365	11/82	675	FAILED: suction plugged by scale	2-JP-EM-5
	-	336	1/83	4,728	NOT FAILED: well taken out of service. Pump spared	2-JP-EM-6
Total NH_3 8-15 ppm	-	365	3/83	720	FAILED: pumpshaft broken at bearing	2-JP-EM-7
TDS 2160-8994 ppm	-	365	5/83	5,811	FAILED: pump wore out	2-JP-EM-8
	1006	350	5/83	1,870	FAILED: pump seized by scale	2-JP-EM-9
GBP[e] 200-280 psia	1006	350	10/83	3,696	FAILED: pump seized by scale	2-JP-EM-10
	1000	350	3/84	7,020	NOT FAILED: removed from service	2-JP-EM-11
	1000	350	7/84	3,912	FAILED: undocumented causes	2-JP-EM-14
Mixed brine flow from multiple formations produces intrinsic scaling	-	360	8/84	6,116	FAILED: pump "worn out"	2-JP-EM-16
	1000	350	5/85	6,244	FAILED: "worn out"	2-JP-EM-20
	1000	350	5/85	10,887	NOT FAILED: pump OK	2-JP-EM-21
	-	348	5/84	5,098	UNDOCUMENTED (presumed failed)	2-JP-EM-12
	1000	349	6/84	12,450	UNDOCUMENTED (presumed failed)	2-JP-EM-13
	-	350	3/85	6,796	FAILED: broken pumpshaft	2-JP-EM-19
	900	347	1/86	6,178	FAILED: Impellers topped out	2-JP-EM-27
	936	345	2/86	0	FAILED: (Presumed) sand locked	2-JP-EM-29
	936	345	2/86	0	FAILED: broken pumpshaft, sand locked	2-JP-EM-30

Table 5-1 (Continued)

LINESHAFT PUMP DATABASE FOR STATISTICAL ANALYSIS

Site	Radian Ident. Number	Well ID	Pump Manufacturer	Pump System Description: Pump Model-No of Stages	Motor HP	Rated Capy (GPM)	Design Comments
Magma Power, East Mesa, CA	2-JP-EM-31	81-7	Johnston	8CC-38	150	326	
	2-JP-EM-32	81-7	Johnston	8CC-38	150	326	
	2-JP-EM-34	81-7	Johnston	8CC-38	150	300	
	2-JP-EM-38	63-7	Johnston	8CC-38	150	325	
	2-JP-EM-35	44-7B	Johnston	8CC-38	125	300	
	2-JP-EM-40	83-7	Johnston	8CC-38	150	325	
	2-J-EM-42	61-7	Johnston	8CC-33	-	-	
	2-J-EM-15	44-7	Johnston	12 EMC-14	-	-	Johnston "standard" high-temp geothermal pump. 12-inch nominal diameter. Features: boride-hardened pump bearings; "raised-throat" geometry.
	2-J-EM-17	88-7	Johnston	12 EMC-20	-	-	
	2-J-EM-18	48-7A	Johnston	12 EMC-16	300	750	
	2-J-EM-22	44-7	Johnston	12 EMC-16	350	700	
	2-J-EM-23	44-7B	Johnston	12 EMC-16	-	-	
	2-J-EM-24	44-7A	Johnston	12 EMC-16	-	-	
	2-J-EM-25	44-7B	Johnston	12 EMC-16	-	-	
	2-J-EM-26	44-7A	Johnston	12 EMC-16	-	-	
	2-J-EM-28	48-7	Johnston	12 EMC-16	300	700	
	2-J-EM-33	44-7A	Johnston	12 EMC-16	350	770	
	2-J-EM-36	88-7	Johnston	12 EMC-16	-	-	
	2-J-EM-37	48-7	Johnston	12 EMC-16	350	700	
	2-J-EM-39	48-7A	Johnston	12 EMC-16	350	700	
	2-J-EM-41	48-7	Johnston	12 EMC-16	350	700	
	2-J-EM-43	44-7A	Johnston	12 CC-24	-	-	
	2-J-EM-44	48-7	Johnston	12 EMC-16	-	-	

(continued from left)

Operating Conditions						
Chemistry	Setting Depth (ft)	Prod. Temp (°F)	Date Installed	Approx. Endurance (hrs)	Condition Upon Removal	Radian Ident. Number
	936	345	3/86	0	FAILED: broken pumpshaft, sand locked	2-JP-EM-31
	936	345	3/86	8,891+	RUNNING: 6/30/87	2-JP-EM-32
	936	348	5/86	4,633	FAILED: cause undocumented	2-JP-EM-34
	900	349	10/86	4,915+	RUNNING: 6/30/87	2-JP-EM-38
	936	360	6/86	7,132+	RUNNING: 6/30/87	2-JP-EM-35
	930	350	1/87	3,653+	RUNNING: 6/30/87	2-JP-EM-40
	-	348	3/87	1,000+	RUNNING: 6/30/87	2-JP-EM-42
	-	365	7/84	5,600	FAILED: seized by scale	2-J-EM-15
	-	348	1/85	510	FAILED: undocumented causes	2-J-EM-17
	1000	350	3/85	720	FAILED: ingested debris (scale) from well	2-J-EM-18
	1000	365	5/85	778	FAILED: plugged with scale	2-J-EM-22
	-	360	6/85	2,371	FAILED: suction screen plugged	2-J-EM-23
	-	336	6/85	2,087	FAILED: bottom impeller collet came loose	2-J-EM-24
	-	360	12/86	3,595	FAILED: undocumented	2-J-EM-25
	-	350	1/86	2,574	FAILED: "Material failure" not documented	2-J-EM-26
	1000	350	2/86	3,233	Undocumented pressure failed	2-J-EM-28
	990	336	5/86	3,735	FAILED: cause unknown, pump not recovered, well abandoned	2-J-EM-33
	-	348	7/86	5,731+	FAILED: undocumented causes	2-J-EM-36
	1006	350	10/86	931	FAILED: seized	2-J-EM-37
	1006	350	1/86	4,492+	RUNNING: 6/30/87	2-J-EM-39
	1006	350	2/87	2,645+	FAILED: Broken lineshaft bearings	2-J-EM-41
	-	365	5/87	841+	RUNNING: 6/30/87	2-J-EM-43
	-	350	6/87	400+	RUNNING: 6/30/87	2-J-EM-44

Table 5-1 (Continued)

LINESHAFT PUMP DATABASE FOR STATISTICAL ANALYSIS

Site	Radian Ident. Number	Well ID	Pump Manufacturer	Pump System Description: Pump Model-No of Stages	Motor HP	Rated Capy (GPM)	Design Comments
Mammoth-Pacific Power, Mammoth Lakes, CA	2-J-ML-1	MPB-4	Johnston	12 EMC-11	250	800	First documented installation of Johnston 12EMC pump
	2-J-ML-2	MPB-5	Johston	12 EMC-11	250	800	All pumps have same features as 12EMC pumps described above
	2-J-ML-3	MPB-1	Johnston	12 EMC-11	250	800	
	2-J-ML-4	MPB-4	Johnston	12 EMC-11	250	800	Pumps installed by Johnston Lineshaft lubrication 1.5 gpd
	2-J-ML-5	MPB-3	Johnston	12 EMC-11	250	800	
	2-J-ML-6	MPB-5	Johnston	12 EMC-11	250	800	
	2-J-ML-7	MPB-4	Johnston	12 EMC-11	250	800	
	2-J-ML-8	MPB-3	Johnston	12 EMC-11	250	800	Pumps installed under M-P supervision. Lineshaft lubrication 3-gpd.
	2-J-ML-9	MPB-4	Johnston	12 EMC-11	250	800	
	2-J-ML-10	MPB-5	Johnston	12 EMC-11	250	800	
Chevron at Heber Binary Plant, Heber, CA	2-P-HB-1	104	Peerless	12HXHG-20	600	1650	Original equipment Peerless Pumps, or units rebuilt to original specifications: • Bronze pump bearings • WC-hardfaced shafts • Field assembled inter-stage pump shaft couplings • OEM quality control specifications
	2-P-HB-2	106	Peerless	12HXHG-20	600	1650	
	2-P-HB-3	102	Peerless	12HXLB-18	600	1320	
	2-P-HB-4	101	Peerless	12HXLB-18	600	1320	
	2-P-HB-5	105	Peerless	12HXHG-20	600	1650	
	2-P-HB-6	101	Peerless	12HXLG-18	600	1320	
	2-JR-HB-7	106	Peerless	12HXHG-20	600	1650	

(continued from left)

Operating Conditions						
Chemistry	Setting Depth (ft)	Prod Temp (°F)	Date Installed	Approx. Endurance (hrs)	Condition Upon Removal	Radian Ident. Number
pH 6.1	600	350	3/84	303	FAILED: Broken enclosing tube.	2-J-ML-1
Cl 237-370 ppm	600	318	5/84	246	FAILED: Lineshaft parted at coupling. Pump sand-locked.	2-J-ML-2
SO_4 60-161 ppm	600	331	7/84	20,185	FAILED: About 25% of bearings in bottom 1/4 lineshaft severely worn. Three brgs parted.	2-J-ML-3
Total CO_2 821-1370 ppm	600	345	1/85	4,236	FAILED: Multiple parting of enclosing tube bearings. Some bearings split, some unscrewed.	2-J-ML-4
Total H_2S 1.6-7.5 ppm	600	341	2/85	5,364	FAILED: Heavy wear and corrosion of bottom third of lineshaft and bearings. Pump severely eroded.	2-J-ML-5
Total NH_3 1.8-4.0 ppm	600	341	2/85	2,433	FAILED: Pump impellers damaged by falling centralizer pin. Severe sand erosion.	2-J-ML-6
TDS 1414-1730 ppm	600	345	9/85	3,487	FAILED: Bottom half of line-shaft unlubricated. Severe wear.	2-J-ML-7
GBP 120 psig	600	340	12/85	12,387+	RUNNING: 6/30/87	2-J-ML-8
	600	346	3/86	10,934+	RUNNING: 6/30/87	2-J-ML-9
	600	337	3/86	1,867+	In service as a spare	2-J-ML-10
pH 5.6-5.9; Cl 8000-8250 ppm	720	365	3/85	2,419	FAILED: Pump bearings failed; pumpshaft misaligned; severe sand erosion; secondary breakup of impellers.	2-P-HB-1
SO_4 50-69.8 ppm	720/[f] 840	365	4/85	926	FAILED: Pump bearings failed; severe sand erosion; impellers precracked during assembly.	2-P-HB-2
Total CO_2 154-290 ppm	740	365	5/85	4,060	FAILURE IMINENT: Pump bearing severely worn; impellers topped out; sand erosion.	2-P-HB-3
Total H_2S 0.9-5.5 ppm	740/ 1000	365	5/85	2,496	FAILED: Pump bearings failed; pumpshaft misaligned; fatigue-fractured pumpshaft coupling; impellers topped out; secondary breakup of impellers.	2-P-HB-4
Total NH_3 Not avail.	720	365	6/85	6,194	FAILURE IMMINENT: Pump bearings severely worn.	2-P-HB-5
TDS 13,200-15,300 ppm	740/ 1000	365	6/85	2,521	FAILED: Pump bearings failed; severe impeller wear; impellers topped out, broken.	2-P-HB-6
GBP Not avail.	1000	365	12/85	584	FAILED: Pump bearing failure; bent pumpshaft; severe pumpshaft misalignment, severe sand erosion; secondary breakup of impellers.	2-P-HB-7

Table 5-1 (Continued)

LINESHAFT PUMP DATABASE FOR STATISTICAL ANALYSIS

Site	Radian Ident. Number	Well ID	Pump Manufacturer	Pump System Description: Pump Model- No of Stages	Motor HP	Rated Capy (GPM)	Design Comments
Chevron at Heber Binary Plant, Heber, CA	2-P-HB-8	106	Peerless	12HXLG-18	600	1320	See note previous page
	2-P-HB-9	103	Peerless	12HBXLG-18	600	1320	Transitional modifications: • Large sand lugs • Bronze bearings • Shop aligned interstage pump shaft couplings • Interim enhanced quality control
	2-P-HB-10	104	Peerless	12HBXHG-20	600	1650	
	2-P-HB-14	105	Peerless	12HBXHG-20	600	1650	
	2-JR-HB-11	101	Peerless	12HXLG-18	600	1320	Same as transitional modifications, except boride-hardened pump bearings.
	2-JR-HB-12	106	Peerless	12HXHG-20	600	1650	Final modifications: • Large sand lugs • Boride-hardened • Shop aligned interstage pump shaft coupling/splice • Fully implemented QC and continuous performance monitoring life-extension programs
	2-JR-HB-13	101	Peerless	12HXHG-18	600	1320	
	2-JR-HB-15	107	Peerless	12HXHG-18	600	1320	
	2-JR-HB-16	102	Peerless	12HXHG-18	600	1320	
	2-JR-HB-17	104	Peerless	12HXHG-20	600	1650	
	2-JR-HB-18	104	Peerless	12HXHG-20	600	1650	

[a]Unique Radian-assigned identification number

[b]Rotating or operating hrs.

[c]Peerless Pump modified and rebuilt by Johnston

[d]Information not available

[e]Gas breakout pressure

[f]Pump reset to second depth

(continued from left)

Operating Conditions						
Chemistry	Setting Depth (ft)	Prod. Temp (^{o}F)	Date Installed	Approx. Endurance (hrs)	Condition Upon Removal	Radian Ident. Number
	1000	365	1/86	719	FAILED: Pump bearing failure; severe pumpshaft misalignment, severe sand erosion; secondary breakup of impellers.	2-P-HB-8
	1009	365	1/86	6,925+	RUNNING: 6/30/87; hrs shown as of 3/31/87	2-P-HB-9
	1000	365	1/86	4,389	FAILED: Coupling in lower lineshaft fractured due to fatigue. Bearings in vicinity worn away to steel sleeves.	2-P-HB-10
	1031	365	7/86	3,081	RUNNING: 6/30/87; hrs shown as of 3/31/87	2-P-HB-14
	1000	365	2/86	2	FAILED: "Tolerance accumulation" resulted in seizure of the top pump bearing.	2-JR-HB-11
	1000	365	2/86	5,950+	RUNNING: 6/30/87; hrs as of 3/31/87	2-JR-HB-12
	1000	365	3/86	Not available	RUNNING: 6/30/87	2-JR-HB-13
	1031	365	7/86	3,269+	RUNNING: 6/30/87; hrs as of 3/31/87.	2-JR-HB-15
	1069	365	8/86	2,779+		2-JR-HB-16
	1000	365	1/87	1,300	FAILED: Inspection showed that one of the bowls had been cracked by the installation tongs.	2-JR-HB-17
	1000	365	4/87	1,545+	RUNNING: 6/30/87	2-JR-HB-18

installed in the latter half. No significant difference between the endurances of the two groups was found.

A third analysis made of the combined Magma Power and Mammoth-Pacific data looked for significant differences in pump endurance by pump model. This analysis showed a general trend toward increasing longest pump endurances but little if any increase in the median endurance. No significant difference was found between the pump types.

These analyses lead to the following conclusions concerning trends:

- The pumps installed at Magma Power since October 1981, the Mammoth-Pacific pumps, and the OEM pumps installed at the Heber Binary Plant can be taken as a single group representing what can be expected of "recently applied practices" at a wide range of resource conditions.
- The data are highly confounded and of such quality that no meaningful correlations between endurance and variables such as operating conditions can be drawn.
- There has been an upward trend in the longest pump endurances, but the recently-applied-practices data exhibit no evident upward trend in the median endurance.

As an illustration, Figure 5-1 shows endurance as a function of installation date for the combined Magma Power and Mammoth-Pacific experiences. In this figure, the pumps are differentiated by type. This figure shows an increase in extreme values which would likely have resulted in an increasing median endurance but for the continuing predominance of early failures.

ESTIMATED PUMP ENDURANCE UNDER RECENTLY APPLIED PRACTICE

A problem which had to be overcome in the estimation of any time-to-failure statistics from the available data was the presence of running pumps. Simple elimination of these pumps from the limited data set would give falsely low values. Therefore, a statistical method was developed to estimate the times to failure of the running pumps.

Examination of the pooled data, including both running and failed pumps, showed that the endurances were highly skewed toward low values, a frequent characteristic of time-to-failure data. As a first order approximation, the pooled data were found to have a log-normal distribution (the logarithms of the endurances were approximately normally distributed). For this analysis, the three "zero run" pumps at Magma Power were arbitrarily assigned lives of one hour, since zero can not be used in calculations involving logarithms.

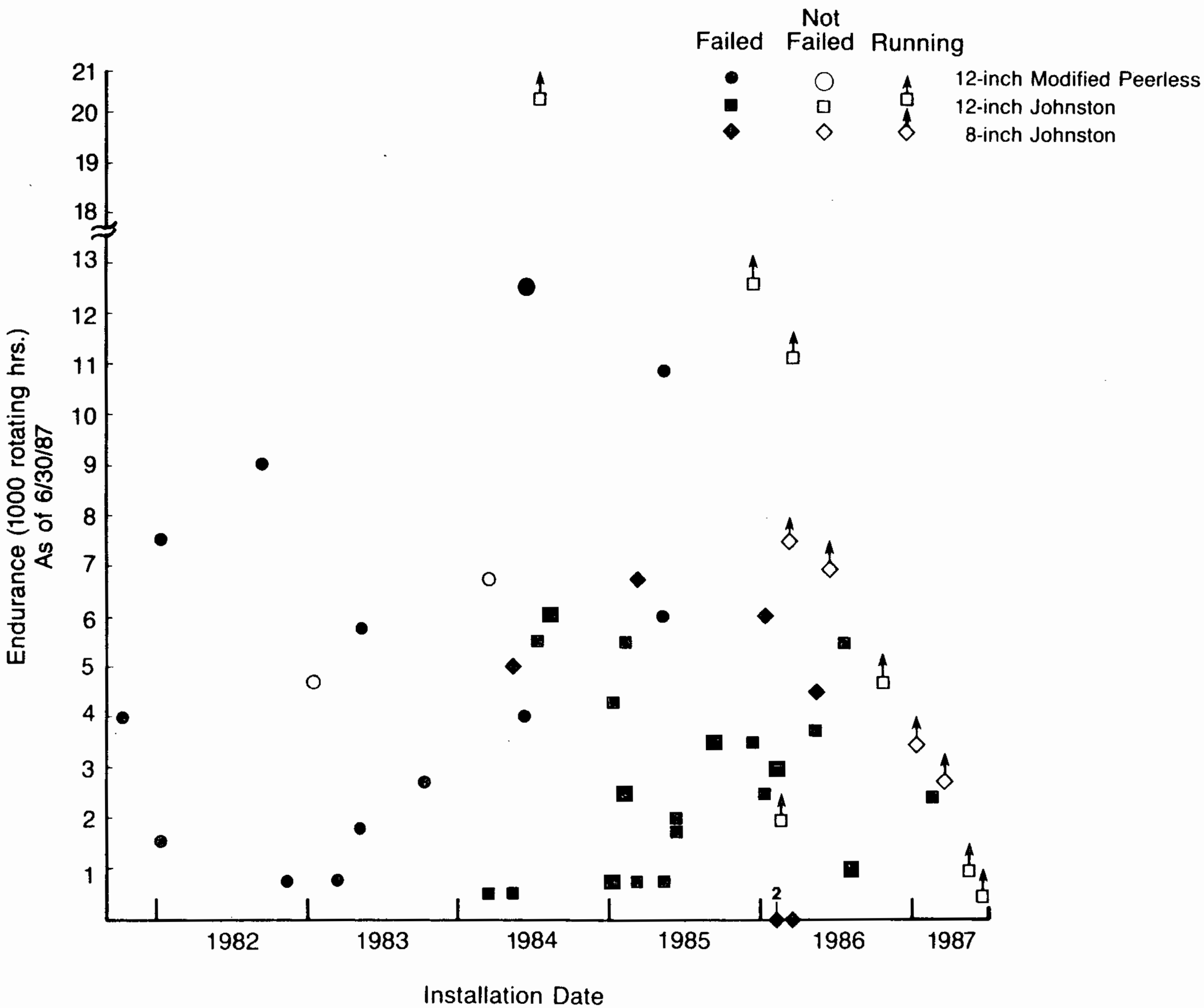


Figure 5-1. Endurance as a Function of Installation Date for Lineshaft Pumps Installed at East Mesa and Mammoth Lakes Since October 1981

The area under the frequency distribution curve is defined as the probability density. For a running pump, the area to the right (toward endurances higher than the time the pump has already run) is the residual density. The pump could fail at any endurance greater that its current running time, but some fraction of this residual density represents the remaining probable life.

The pooled data set contained three pumps that were running as of 31 March 1987, but had failed as of 30 June. Using the 31 March running time as a starting point, endurances were calculated using different fractions of the residual density until the fraction which gave the closest fit to the actual time to failure was found. This fraction (20%) of the residual density, along with each running pump's endurance to date, was then used to estimate a time to failure for that pump. These estimated times to failure for running pumps combined with the known times to failure of deceased pumps formed the basis for calculating pump life statistics.

Based on this rationale, the (arithmetic) mean time-to-failure was estimated to be approximately 5100 hours ($\pm$750 hours at the 80% confidence level). While the mean time-to-failure of these pump systems was about 5100 hours, one-fourth of them failed in less than 1400 hours, and half failed in less than 4100 hours. At the other extreme, one-fourth of the pumps ran more than 7100 hours.

From these statistics, it is clear that under recently applied practices of manufacture, installation, and operation, lineshaft pump systems can run more than 7000 hours under a wide range of resource conditions, but that a high frequency of early failures from a variety of causes has kept the mean time-to-failure low.

IMPACT OF THE CHEVRON PUMP LIFE-EXTENSION PRACTICES

As was discussed in Section 4, Chevron at the Heber Binary Plant has implemented a program of pump life-extension practices with the following elements distinct from those recently applied at other sites:

- Enhanced assembly quality control;
- Improved installation practices; and
- Semi-continuous monitoring of pump efficiency.

There is no way to directly quantify the effect of these practices on pump life, since all of the pumps which have benefitted from the full implementation of the Chevron program were still running as of 30 June 1987.

However, the first four pumps to benefit from the full program of life-extension practices were each running without trouble after more than 4000 hours. The hypothesis that these pumps are from a population with an endurance distribution similar to the set defined as "recently applied practice" (i.e., the null hypothesis) was tested. The probability of randomly selecting four successive pumps with more than 4000-hour endurances from that distribution was less than 2.5 percent. Thus, it is very likely that the success of these four pumps represents a real improvement rather than a statistical fluke.

In order to form a rough estimate of near-term endurances, it was assumed that:

- General implementation of the Chevron pump life-extension practices will essentially eliminate failures at less than 4000 hours.
- The implementation of these practices will have relatively little effect on the high end of the endurance distribution.

Using these assumptions, the failures at less than 4000 hours were eliminated from the recently-applied-practices endurance distribution, and new statistics were calculated from the remaining data. The results, which must be considered as rough estimates, are shown in Table 5-2. If these assumptions are even a close approximation of reality, they indicate that the implementation of the Chevron pump life-extension practices is likely to result in a near-term mean time-to-fail of at least 8000 hours (8600 ± 950 hours at the 80% confidence level).

PUMP AVAILABILITY AND BRINE SUPPLY CAPACITY

The ratio of the time-to-fail (TTF) of a particular kind of pump to the sum of the TTF plus the time-to-repair (TTR) is the pointwise availability; the probability that a given pump will be operational at any randomly selected time. Over a relatively long period of time, such as a year, this ratio also represents the fraction of time that the pump will be available to produce brine. Thus, the individual pump availability factor (A%) is given by:

$$A(\%) = 100[TTF/(TTF + TTR)]$$

Estimates of pump availability factors require that the TTR be known. Table 5-3 contains a number of estimates of TTR. This table indicates that, if everything goes well, a pump can be pulled, replaced, and brought on-line in about five to eight days. The TTR could be greater than 30 days if no pump was readily available, if well work repair was required, or if other problems occur. Table 5-3 indicates that 13 days may be a reasonable average TTR which allows for the various problems

Table 5-2

EFFECT OF ELIMINATING EARLY FAILURES

Statistics	Recently Applied Practice	Estimated Near-Term[a]	Nominal Improvement
Mean			
Nominal	5100 hrs	8,600 hrs	69%
Limits[b]	±750 hrs	±950 hrs	-
Distribution			
First Q-tile	1400 hrs	5,700 hrs	300%
Median	4100 hrs	7,100 hrs	73%
Third Q-tile	7100 hrs	11,000 hrs	55%

[a]Assumes no failures at less than 4000 hrs.

[b]Limits of mean at 80% confidence level.

Table 5-3

TIME TO REPLACE LINESHAFT PUMPS

Estimates	Calendar Days[a]
Prior Survey	
- Manufacturers' best estimate	5
- Typical good case	7
- Worst case[b]	31.5
Current Survey	
- Magma Power Typical[c]	5
- Mammoth-Pacific Typical[c]	8
- Heber Binary Typical[c]	8
- Heber Binary avg. of eight installations	13

[a]Includes moving rig to site, pulling pump, delivering new pump, installation, and checkouts and start up.

[b]Assumed that no pump was on site and that replacement was not off-the-shelf.

[c]Assuming that rig and replacement pump are available within a few hours of the decision to pull and replace.

which may occur. This value for the TTR was used throughout the rest of this analysis.

In practice, the impact of the pump availability factor on the brine supplier's ability to deliver the required brine is not intuitively obvious because the supplier will in most cases use a number of wells to meet the plant's requirements.

In order to explore the effect of the pump availability factor on the brine supplier's ability to meet the plant's brine requirements, a relatively simple probability model, using the binomial distribution to simulate a hypothetical binary power plant brine supply system, was created during the prior survey (1). The hypothetical brine supply system model had the following characteristics, definitions, and assumptions:

- The amount of brine required to operate the plant at 100% of its rated output was defined as 100% brine supply.
- Operation of the plant at 100% of rated capacity required the brine production of 10 equally productive wells. Therefore, 100% brine supply capacity is defined as the production from 10 wells.
- The brine supply system could have some number of spare wells with pumps installed. The model considered no spare wells, two spare wells, and five spare wells.
- The power plant could not accept more than 100% brine supply, as defined above. Therefore, it was assumed that if more than 10 wells are operational at any time, then either production from all wells would be reduced, excess brine would be diverted into the production casing of the wells, or excess wells would be shut down. In any case, delivery of more than 100% brine supply was not allowed by the model.
- It was assumed that there was no difference in the availability factor for pumps on standby, as opposed to pumps in production, an assumption which appears justified by the results of the present survey.
- Because only the pumping system was being investigated, it was assumed that no factor other than pump availability affected the capacity of the brine production system (more realistically other factors such as well failure may also intrude).

It was judged that this model fairly represents the physical system described above and provides an adequate set of output quantities to characterize the effect of availability of individual pumps on the performance of the brine supply system. The mathematical basis of this model, along with a variety of general graphical solutions, can be found in reference 1. These graphical solutions allow ready determination of a number of brine supply statistics given only the individual pump availability factor.

Table 5-4 summarizes the individual pump and brine supply system statistics generated by this model using the estimated mean time-to-failure resulting from "recently applied practices" and the near-term estimated time-to-failure derived above. For the sake of historical completeness, the same statistics for the high-capacity lineshaft pumps documented in the prior survey are also presented.

REFERENCE

1. P. F. Ellis, T. F. Green, and H. J. Williamson. Geothermal Downwell Pump Reliability: State-of-the-Art Assessment, AP-3572, Electric Power Research Institute, Palo Alto, CA, June 1984.

Table 5-4

EFFECTS OF LINESHAFT PUMP IMPROVEMENTS

Variable	Prior Survey (pre 10/81)	Recently Applied Practice	Near-term[a] Estimate
Individual Pumps			
Mean life (hrs)	900	5100	8600
A (%)[b]	74	94	96
Hypothetical wellfield[c]			
Avg. Brine Supply Capacity Factor (%)			
10 wells (no spares)	74	94	96
10 wells + 2 spares	86	99	>99
10 wells + 5 spares	96	>99	>99
% of year that Brine Supply Capacity will be <100% due to pump problems			
10 wells (no spares)	94	45	30
10 wells + 2 spares	61	3	<1
10 wells + 5 spares	15	nil	nil

[a]Assumes that general implementation of Chevron's pump life-extension practices will essentially eliminate failures at less than 4000 hrs.

[b]Individual pump system availability factor assuming an average time-to-replace of 13 days.

[c]Refer to text and reference 1 for discussion of model used.

Section 6

ELECTRIC SUBMERSIBLE AND NOVEL PUMP TECHNOLOGY STATUS

Electric submersible pumps (ESPs) are fundamentally different from lineshaft pumps in that the electric motor is close-coupled to the pump unit and is therefore located downwell. The novel technology pumps are centrifugal pumps driven by a close-coupled hydraulic turbine which is in turn driven by pressurized water from a surface motive pump. Each of these technologies eliminates the long lineshaft connecting the downwell pump to the surface motor. Each technology offers the possibility of deeper pump setting depths and/or settings beyond deviations in crooked wells. There has been very little high temperature geothermal experience with either technology since the original survey.

ELECTRIC SUBMERSIBLE PUMPS (ESPs)

Technology Overview

Figure 6-1 illustrates a typical ESP installation, though there are no typical high-temperature installations at this time. The electric submersible pump system consists of a multi-stage downwell centrifugal pump, a downwell motor, a seal section (also called a protector) between the pump and motor, all supported within the well by a production column which also conducts the pumped brine to the wellhead, as well as a metal-armored power cable to conduct electricity from the surface to the motor. The seal section's function is to allow expansion of the oil which fills the motor while keeping brine out. The pothead seal at the motor performs the same function, while the packoff seals the cable where it exits the wellhead.

Electric submersible pumps for high-temperature, high-capacity geothermal brine production do not exist at this time, though the various "pieces" of the necessary technology appear to have been developed. When, or if, such pumps are deployed, it is likely that the 12-inch nominal diameter pump will be the optimum compromise between well cost and brine productivity.

There is no theoretical limit to the setting depth of ESPs, but 2000 feet (610 m) may be a practical limit for 12-inch pumps with the desired high flow rates.

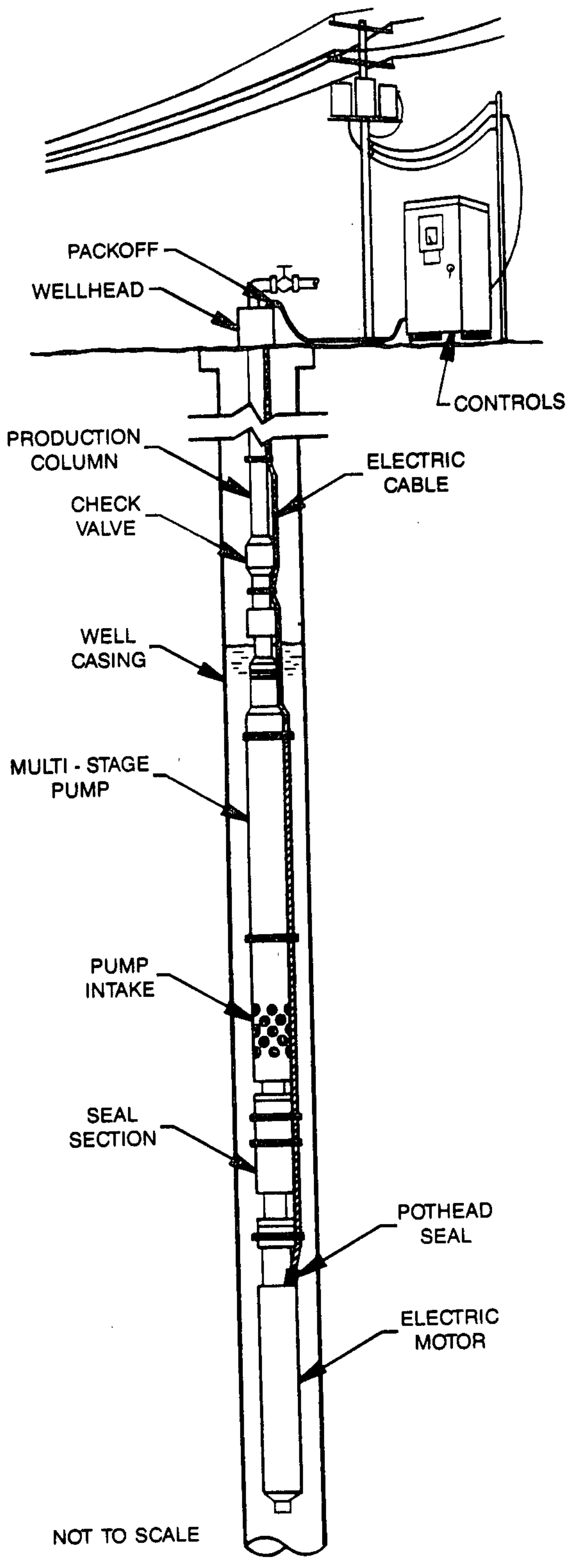


Figure 6-1. Typical Electric Submersible Pump System Installation

Prior Status of Electric Submersible Pumps

The electric submersible pump system has the apparent advantage of eliminating the complicated lineshaft assembly which limits lineshaft settings to less than 2000 feet (610 m). However, attempts to operate such pumps in the 268° to $350^{\circ}F$ (131° to $177^{\circ}C$) range showed that these pumps suffered from a number of weak points.

AP-3572 documented 30 actual geothermal trials of ESPs, as well as an additional 16 tests in a portable DOE test facility. Table 6-1 (modified from Table 3-4 of AP-3572) summarizes the causes of failure and degradation for the 30 electric submersible pumps actually field tested during the prior survey. Assignment of cause of failure for these pumps was complicated by the fact that many suffered multiple "fatal injuries," any one of which would have rendered the pump system inoperable. Therefore, it was not possible to assign a single cause of failure to many of these pumps. This complication of the data manifests itself in Table 6-1, where the columns do not "add up."

One of the weakest elements in the electric submersible system was the conductor cable. Numerous cases of failure due to mechanical damage during installation were documented. Gradual degradation of the insulation over time was also noted. Additionally, four of the failures resulted from inadequate quality control during the installation of packoffs.

The electric motors suffered from several factors. The most common difficulty was brine inleakage through the pothead or seal section. In some of these cases it was not possible to determine the origin of the brine. Brine inleakage generally shorts out the motor.

Another prevalent cause of failure was inadequate quality control during the fabrication of the motor itself. Cases of cracked solder connections and spalled winding encapsulation epoxy were documented.

Another motor problem was excessive wear of the thrust bearings. These bearings present an unusual challenge due to the temperature and confined geometries of electric submersible motors.

Problems with the pumps themselves were relatively rare, probably mostly due to the generally short run times for the pumps.

Table 6-1

COMPONENT DEGRADATION DISTRIBUTION FOR ELECTRIC SUBMERSIBLE PUMPS
(AP3572 Data Only)

Incidence of Failure/Distress	Instances as Fraction of Pumps Surveyed			
	Failure[a]		Distress[b]	
	Total	Operational[c]	Total	Operational[c]
Overall Pump Condition	25/30	20/30	3/30	0/30
By Cause/Location				
Cable				
Insulation breakdown	1/30	0/30	4/30	0/30
Mechanical damage	5/30	5/30	4/30	4/30
Motor				
Brine Inleakage[d]	8/30	4/30	6/30	0/30
Bearing Wear	4/30	0/30	2/30	0/30
Winding Insulation[e]	8/30	7/30	4/30	0/30
Pump				
Bearing Wear	0/30	0/30	2/30	0/30
Impeller/Diffuser Wear	2/30	1/30	1/30	0/30
Pump Scaled	2/30	2/30	0/30	0/30
Packoff	4/30	4/30	0/30	0/30

Notes: Insufficient data to assess the cause(s) of failure of 3 of the 30 pumps surveyed.

Columns will not "add up" due to multiple causes of failure in many pumps.

[a]Deemed to be the component(s) whose degradation made the pump inoperable.

[b]Degradation was not sufficient to render the pump inoperable, or damage was judged consequential to the component(s) which had failed. Distressed components would likely have produced their own failures given longer pump life.

[c]Indicates that some identifiable error in installation and/or operation was a key contributor to the observed degradation.

[d]Indicates pothead and/or seal section failure.

[e]Generally indicates an internal short.

With the exclusion of three experimental surface-pressurized pumps discussed below, the average run for the electric submersibles was about 560 hours with a maximum of 4052 hours and ten runs of less than 24 hours.

Under DOE funding, Barber-Nichols Engineering modified three small (80 hp) REDA electric submersible pumps so that they could be continuously pressurized with oil conducted from the surface via a separate small diameter oil tube. The underlying premise was that this would prevent brine intrusion, the leading cause of motor failure.

Three attempts were made in East Mesa wells yielding 350^{o}F (177^{o}C) brine. In the first, the oil tube was damaged during installation and the pump ran for only 136 hours. Installation of the second pump unit went well and this pump ran 7900 hours! The pump was finally pulled because of falling production and evidence that failure of the power cable was imminent. The pump was found to have been severely eroded.

The third pump was run only 1152 hours, prior to being shut down because the brine was no longer needed. This pump remained in the well for several months after shutdown. When an attempt was made to recover the pump, it was determined that the well casing had collapsed, making recovery impossible.

At the time of the initial survey, REDA indicated plans to develop and test full-sized (i.e., 350 hp) surface pressurized electric submersible pumps. Radian estimated that given sufficient control over manufacturing and installation, such pumps held the prospect of 6-to-12-month service lives.

Experience Update

Barber-Nichols (DOE). At the close of the prior survey, Barber-Nichols Engineering was testing surface-pressurized modifications of TRW-Reda's ESP under DOE sponsorship at the 500 kW direct contact binary demonstration project at DOE's East Mesa site. The 500 kW DCBD was shut down at the end of 1982, and no further tests were conducted at this site.

The small, 80-hp, prototype pumps tested by Barber-Nichols utilized both an oil tube and a separate nitrogen bubble tube from the wellhead to control oil pressure within the pump motor chamber. Though excellent for experimental purposes, this system had two distinct disadvantages for a commercial downwell lubrication system:

- In addition to the oil flow line it also required a separate nitrogen bubble tube set to the pump setting depth. Bubble tubes frequently plug, which necessitates pulling and cleaning. Additionally, it represents one more relatively fragile component being put into the well, one more risk of installation damage.

- The system could not be used on deepset pumps where the hydrostatic head of the oil column would exceed the desired oil pressure in the motor housing.

Barber-Nichols designed, built, and bench-tested a downwell motor oil pressure regulator valve to solve these problems. The valve body was designed to fit below the pump motor, and contained separate differential pressure regulator and relief valves. In operation, if the motor oil pressure dropped to within 3 psi (20.7 kPa) above the brine pressure, the regulator would open, allowing pressurized oil from the surface supply to enter the motor. At 5 psi (34.5 kPa) the valve closed, stopping the oil flow. If oil pressure exceeded 10 psi (69.0 kPa) above brine pressure, the relief valve opened and vented oil until the pressure dropped to 7 psi (48.8 kPa) over the brine pressure. Thus, this regulator controlled oil pressure within the motor to 3 to 10 psi (20.7 to 69.0 kPa) above the brine pressure regardless of the pump setting depth, and without continuous venting of oil.

Apparently this regulator was bench tested at 380^{O}F (193^{O}C) with good results. Barber-Nichols also indicates that one of these regulators was tested on a pump in DOE's mobile geothermal pump test facility in late 1984. This skid-mounted facility simulated geothermal temperatures and pressures, but not the aggressive chemistry. The pump ran for 180 hours at 375 ^{O}F (190^{O}C) without failure of the regulator, though the 5 micrometer filter on the oil feed line did plug. Testing was discontinued for lack of funds.

<u>TRW-Reda at East Mesa</u>. In March 1983, TRW-Reda installed a full-scale, surface-pressurized ESP in Magma Power's well 44-7 at East Mesa. This pump system had two 150 hp series 540 motors in tandem. The motor section was fitted with a surface-pressurization system designated by TRW-Reda as "Posi-life." A surface mounted metering pump provided a continuous flow of oil to the downwell motors, where a pressure relief valve allowed a constant bleed of oil to the external brine.

Installation of the pump began on 6 March 1983, and a number of minor, but very annoying, problems occurred which were the result of inexperience with the new system. The greatest aggravation occurred when vacuum filling the motors with the extremely viscous synthetic motor oil while the ambient temperature was low. This was a slow process despite the preheating of the oil to 220^{O}F (105^{O}C) before filling.

During installation, monitoring indicated a loss of continuity in one leg of the three-phase cable. The pump was pulled, the oil drained, and the pothead (the device through which the cable connects to the motors) was removed. When the pothead was removed, the connections between the pothead and the motor leads separated with very little effort. Resistance reading of the cable and motor showed that the loss of continuity was apparently a loose lead connection.

The leads were crimped together during reassembly to assure tight connections. After being refilled with oil, the unit was reinstalled without incident.

The unit operated normally for about 336 hours over the next 16 days, delivering over 600 gpm (38 L/s) at 275 psig (2 MPa). On 22 March a power interruption occurred. When the pump was restarted, it tripped on overload and further attempts to restart were unsuccessful. The pump was pulled from the well and surface inspection revealed that a massive "short" had occurred in the pothead region, destroying the pothead.

Teardown inspection at the TRW-Reda plant showed clearly that the failure mode was an electrical short in the pothead/motor lead wire area. The resulting arc was intense enough to explode the pothead housing, leaving the upper motor assembly open to the brine. It was concluded that the repaired motor lead connectors were partially or totally responsible for the electrical short.

The teardown inspection found that the general condition of most of the critical elements of the motors, protector, pump and cable splice were in "excellent" condition. However, some sand erosion of the impellers was noted.

To the best of Radian's knowledge, no further field testing of high temperature geothermal ESPs has been conducted by TRW-Reda.

<u>Other manufacturers</u>. In addition to TRW-Reda, there were two other electric submersible pump manufacturers active in the geothermal field at the time of the previous survey, Centrilift-Hughes and Kobe (now Trico Industries). Centrilift-Hughes had accumulated both field experience and test experience with the DOE portable test facility at the time of the prior survey; this experience was documented in AP-3572. Trico Industries had begun development of a geothermal pump line using the DOE portable test facility. Both companies indicated ongoing development programs at the close of the prior survey.

However, the cessation of DOE support of pump development efforts, coupled with subsequent economic factors and the inherent small size of the geothermal power market, resulted in the termination of these development projects shortly after the prior survey was completed. There has been little, if any, further development since the prior survey.

Based on their geothermal experience, Centrilift-Hughes has developed a 300^{O}F (149^{O}C) ESP for the oil industry. Development of this pump required a major improvement in the understanding of thermal growth and its impact on the required precision of fit and tolerences. This development work has also lead to the development of power cable insulated with a proprietary EPDM formulation rated to 400^{O}F (204^{O}C). Centrilift-Hughes opines that all of the necessary technology is available to create a 375^{O}F (191^{O}C) geothermal ESP with an anticipated run life of 6 to 12 months, but that considerable development work would be required to bring such a pump to fruition.

Conclusions Concerning Electric Submersible Pumps

The status of electric submersible pumps appears to have changed little since the original survey, primarily because little additional development work has been done. However, it appears likely that this has been the result of market and economic factors rather than an industry consensus that the technology is not viable.

The experience accumulated in these two surveys indicates that the necessary pieces to create a durable electric submersible pump with a ceiling temperature in the 375 to 400^{O}F (190 to 204^{O}) range probably do exist, but that a considerable development effort may be required to bring them together with sufficient refinement to give consistent high endurances.

The following areas, among others, will have to be improved.

- Cable. The power cable is one of the components most vulnerable to damage. Existing cables rely on wrapped wire armor or flexible conduit, both of which allow brine to contact the elastomeric insulation. For these cables, development of insulation capable of retaining resistance for extremely long times in hot brine is a must, as is development of stouter cable armor to protect the cable during installation.

  Before DOE funding was reduced, Halpen Engineering was developing a solid-metal-sheath conductor cable for geothermal pumps. This conductor had a continuous metal sheath which excluded all brine, and used high-temperature, non-elastomeric insulation. Commercialization of such cable would probably solve both the insulation degradation and cable mechanical damage problems.

- Pothead and Motor Lead Connectors. The pothead, the device which terminates the downwell end of the power cable, has been implicated in more high-temperature ESP failures than any other component. It must form a brine-tight termination for the cable insulation, as well as a brine-tight seal against the motor housing. Existing potheads rely on large elastomeric seals, their weak point. If continuous solid-metal-sheath cable is commercialized, then potheads with metal-to-metal seals at both the cable and the motor housing should be possible, and would likely be more reliable.

  A surprising number of pothead failures appear to actually relate to the motor lead connectors, suggesting that there is some basic problem with these components as well.

- Surface-pressurized Motor Protection System. The work by Barber-Nichols has shown that such systems may be critical to the durability of high-temperature ESPs. Demonstration of a reliable system, capable of operating at any depth and requiring a minimum of hardware, particularly conductors to the surface, will be required.

- Seal Section. The use of surface-pressurized motor protection systems should relieve the seal section of its duty to accommodate the expansion and contraction of the motor oil. This should simplify its design. However, the seal section must still house the thrust bearings in a small space. The difficulty with thrust bearings may be increased if uncompensated impellers must be used (lineshaft experience has shown that thrust-balanced impellers are subject to many problems in geothermal service).

  Development of reliable seal sections that can accommodate the thermal expansion of the motor oil while adequately excluding brine moisture would be valuable, since the surface-pressurization system itself is vulnerable to damage during pump installation.

- Motor. The downwell motor operates considerably above the ambient brine temperature. Motor insulations capable of prolonged service at well above 400°F, even in oil with some brine contamination, will be required.

- Pump. As was the case for lineshaft pumps, the electric submersible pump bearings are likely to be the weakest point with respect to degradation by sand. The boride-hardened steel bearing/uncoated 17-4PH stainless steel shaft technology, already demonstrated in lineshaft pumps, should be applicable to this problem.

  Electric submersible designs frequently rely on thrust-compensated impellers. Such impellers are susceptible to severe erosion of the compensation ports, resulting in lost efficiency, or to plugging of these same ports by foreign solids, resulting in higher-than-design bearing loads. Their use has been abandoned by the lineshaft pump manufacturers. It may be necessary to develop ESPs with uncompensated impellers.

- Quality Control. Both the original survey experience with ESPs and the more recent experience with lineshaft pumps have shown that reliable performance will require much more exacting quality control of manufacture and installation than is commonly practiced for oilfield ESPs. In particular, 100% inspection of all components, as opposed to statistical process control, is likely to be required.

NOVEL TECHNOLOGIES

Technology Overview

Figure 6-2 is a conceptual drawing of a close-coupled hydraulic turbopump for geothermal operations. Such pumps have already been demonstrated in the oil production industry. In this pumping concept the centrifugal pump is close-coupled to a hydraulically driven turbine just above the pump. Furthermore, the pump is packed off (sealed) against the cemented casing, allowing the brine to be produced through the casing rather than through a separate production column. The turbopump unit is suspended in the well on a relatively small diameter tubing which also serves as the energized brine, or charge fluid, conductor.

During operation, a small sidestream of the produced brine is pressure-boosted in a "motive pump" and directed downwell through the drive fluid tube to the hydraulic turbine where its pressure is converted to torque to drive the pump as the pressurized brine is discharged back into the well.

The surface motive pump/downwell hydraulic turbine imposes an unavoidable efficiency penalty. Both Weir and Anderson indicate greater than 80% efficiency for each component: surface motive pump, downwell turbine, pump unit. The resulting overall efficiency is 50 to 60%, compared to 70 to 80% for lineshaft pumps.

The proponents of such pumps feel that they will be much more rugged and easily installed than either lineshaft or electric submersible pumps. The pump unit itself operates at much higher speed than is available to the other technologies, resulting in a pump of only a few stages. This makes fabrication from "exotic" alloys economical.

Also in their favor is the fact that a close-coupled hydraulic turbopump which will fit in 9 5/8-inch casing can produce as much brine as a 12-inch lineshaft pump. Thus, these pumps could be installed in smaller (cheaper) wells, and could be set very deep. Despite the drawbacks, this technology does appear to offer the best probability of success if or when downwell pumping is extended to resources hotter than 400^{o}F (204^{o}C), or to the much more aggressive resource chemistries, such as those of the Brawley and Salton Sea areas.

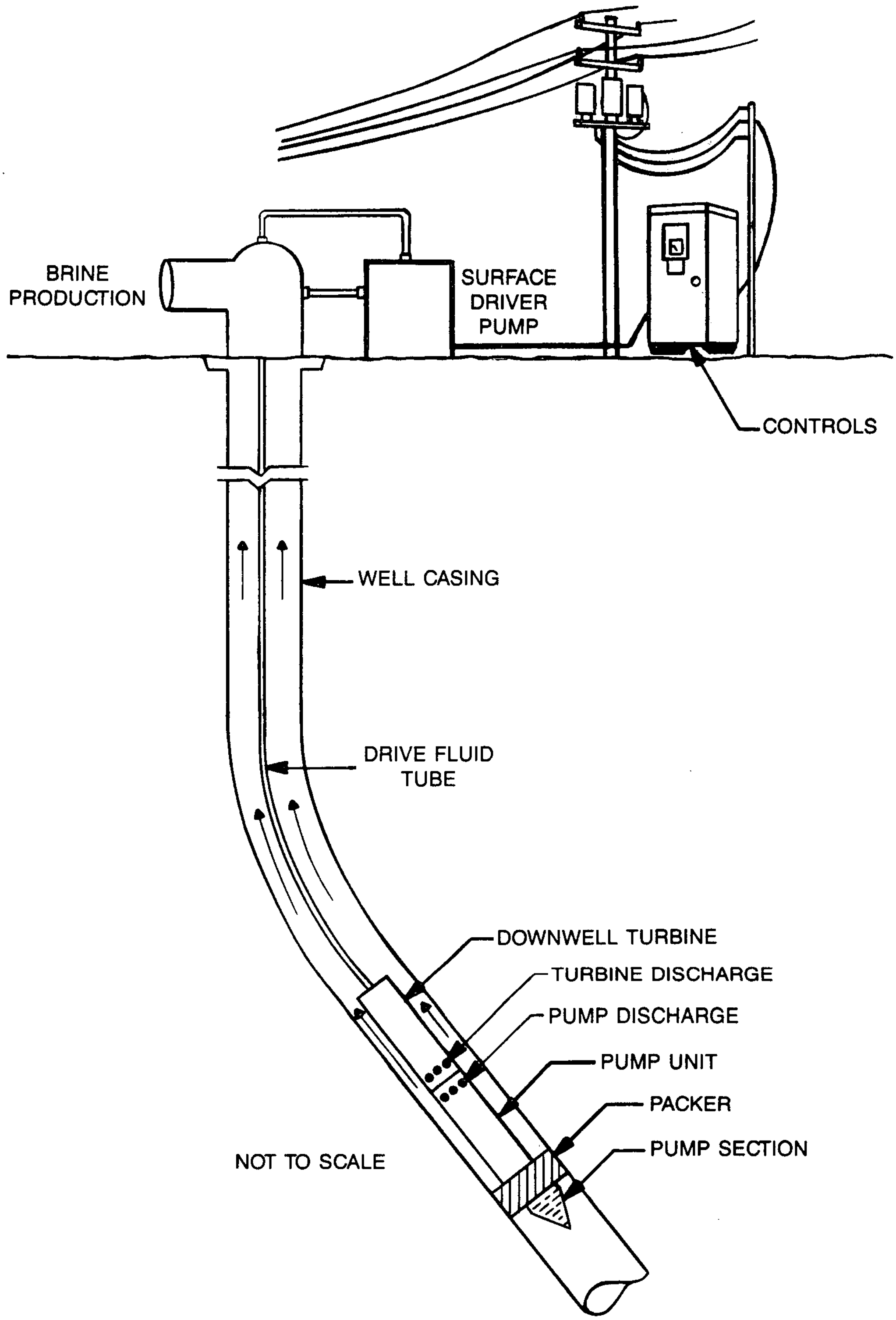


Figure 6-2. Conceptual Close-coupled Hydraulic Turbopump Installation in a Deviated Well

Prior Status Of Novel Technologies

At the time of the previous survey, Weir Pumps, Ltd., of Scotland and J. Hilbert Anderson, Inc., were each developing close-coupled downwell hydraulic turbopumps for high-temperature, high-capacity geothermal applications.

Weir and Anderson both estimated that their pump systems could operate at temperatures of at least 390°F (200°C) and setting depths of 2000 feet (610 m) even in deviated wells. They argued that these various advantages offset the inherent lower overall efficiency of the close-coupled hydraulic turbine concept. Measured from the shaft of the surface motive pump, the peak efficiency of these systems is about 50 percent.

In the spring of 1983, Weir had completed 2500 hours of testing of a 320-hp unit set at 2000 feet (610 m) in a low temperature well. The Weir pump was constructed almost entirely of cobalt-based super-alloys ("Stellites"). Weir considered their system to be commercial at that time, and were anticipating the opportunity to test similar pumps in Magma Power Company's wells at East Mesa.

Anderson had completed successful room temperature bench tests of a prototype in 1981. Funding limitations had prevented further development.

Experience Update

In March 1985, a Weir pump with a four-stage turbine and two-stage pump was installed in Magma Power Company's well 48-7. This pump accumulated 1119 rotating hours prior to failure. It was removed in June 1985. Magma Power's logs indicate that the pump was "destroyed."

A second Weir pump with a two-stage turbine was installed in the same well in July 1985. After 380 rotating hours, this pump was removed in August to allow repair of the packer. The repaired packer and the Weir pump were reinstalled in September. The pump accumulated 1109 rotating hours prior to failure, and was removed in January 1986. Inspection showed that "the inlet check valve [was] plugged with very fine scale and [a] small amount of mud."

In June 1986, a third Weir pump was installed in well 88-7. This well had a prior history of high sand production. The Weir pump ran six hours. Magma Power's logs list it as "destroyed."

There has been no further testing of the Weir pump in U.S. geothermal applications. Likewise, there have been no trials of the Anderson pump. Lack of funding for field tests appears to be the constraint.

Conclusions Concerning Novel Technologies

There has been little change in the status of the novel pump technologies since the initial survey. An indeterminate amount of development will be required before these pumps can be considered commercial, at least for application in high-temperature geothermal wells.

The surface motive pump/downwell hydraulic turbine imposes an unavoidable efficiency penalty. Both Weir and Anderson indicate greater than 80% efficiency for each component: surface motive pump, downwell turbine, pump unit. The resulting overall efficiency is 50 to 60%, compared to 70 to 80% for lineshaft pumps.

In their favor is that a Weir (and presumably Anderson) pump which will fit in 9 5/8 casing can produce as much brine as a 12-inch lineshaft pump. Thus, these pumps could be installed in smaller (cheaper) wells, and could be set very deep. Despite the drawbacks, this technology does appear to offer the best probability of success if or when downwell pumping is extended to resources hotter than 400°F (204°C), or to the much more aggressive resource chemistries, such as those of the Brawley and Salton Sea areas.

Section 7

PUMP DEVELOPMENT PROJECTS

At the close of the prior survey, several firms had active programs to develop high temperature geothermal brine production pumps.

- Johnston Pump had entered the market and was developing a lineshaft pump of their own design.
- Centrilift-Hughes was developing a line of electric submersible pumps for 375°F (190°C) geothermal service.
- Kobe (now Trico) was developing a separate line of high-temperature geothermal electric submersible pumps.
- TRW-Reda was preparing to test the first in what was to be a line of surface-pressurized electric submersible pumps.
- J. Hilbert Anderson had a close-coupled hydraulic turbopump at the prototype stage.
- Weir Pumps Ltd was preparing to field test a close-coupled hydraulic turbopump.
- Barber-Nichols Engineering was developing improved oil pressure regulators for surface pressurized electric submersible pumps.

The results of these various projects have been discussed in Sections 3, 4, and 6.

In 1982 and 1983 most of these development projects came to an end, primarily due to market pressures and the withdrawal of third-party research and development funds. The position of most pump manufacturers is that the size of the geothermal market does not justify unilateral expenditure of the funds necessary to further develop their products for this market. Several indicated that some form of cost-sharing would be required to promote further development. This survey identified only two ongoing pump development programs.

Johnston-Pump continued pump research, as was discussed in Section 3, culminating in the introduction of their unique raised-throat pump line in late 1983 or early 1984. Johnston has continued to expand and improve this product line.

Gould Pump (Vertical Product Division) has recently entered the geothermal market with a lineshaft pump line of their own. Gould indicates that they have an R&D budget of $5.2M, largely for geothermal development. Gould indicates that they have had two pumps in 340°F (171°C) wells near Reno (NV) for "almost a year". Gould indicates that they expect to install pumps at Ormesa (at East Mesa) and the Heber Binary Plant in 1987 or 1988.

Gould indicates that their pumps have design and materials modifications which will, in Gould's opinion, make their pumps more reliable than the products now on the market. Gould declined to provide any details concerning these modifications.

Section 8

CONCLUSIONS

This survey is an update of a prior state-of-the-art assessment of high-temperature geothermal downwell brine production pumps conducted for the Electric Power Research Institute (EPRI) in 1982 and 1983 and published as EPRI report AP-3572. The conclusions presented below reflect the findings of both the original survey and the current survey.

THE DOWNWELL PUMP ENVIRONMENT

- The range of high-temperature (greater than 250°F, 121°C) downwell pumping conditions which may be encountered in the U.S. is broad. Temperatures may range from approximately 280°F (138°C) up to more than 400°F (204°C). However, approximately 80% of the potential applications have resource temperatures of less than 365°F (185°C), and the resources likely to be of greatest interest in the near term will probably have production temperatures in the range of 330° to 365°F (165° to 185°C).

- The chemistry of geothermal resources varies widely. The total key corrosive species (TKS), a first-order indicator of corrosivity, may vary from less than 500 ppm to more than 100,000 ppm. However, the resources pumped to date, and those likely to be pumped in the near term, all fall into Class IV of the Geothermal Corrosivity Classification System developed and published by Radian. The corrosivity of this class of brines is relatively mild to the materials commonly used in geothermal pumps, and corrosion per se should be a relatively minor problem. This conclusion is supported by the finding of this study that prolonged standby in a well has little effect on ultimate pump system life, provided that steps are taken to protect the lineshaft from brine intrusion.

- Sand is a fact of life in many geothermal wells. It may come from the formation or result from well workover operations. High sand production can severely damage downwell pumps in several ways. If there is enough sand, it can simply clog the pump, or become so wedged between the impeller skirt and bowl throat that the pump "locks up" and won't turn. Sand produces severe erosion of some pump bowl geometries, and is particularly damaging to the bearing technologies used in earlier pumps.

- Calcium carbonate scale will rapidly plug pumps if the drawdown is sufficient to allow gas breakout in the pump suction. Maintenance of sufficient suction pressure will prevent this form of scaling. Production of brine from multiple zones in a formation can produce scaling problems which cannot be controlled by this method.

LINESHAFT PUMPS

- At the close of the prior survey the mean life of high-temperature lineshaft pumps was less than 1000 hours, with most failures being attributed to lineshaft, rather than pump, problems. By late 1981, it was clear that water-lubricated lineshafts could not provide adequate service, and that the then existing oil-lubricated lineshafts had severe but surmountable problems as well.

- Since late 1981, several distinct improvements in lineshaft pump technology have occurred, including:

  --improved oil-lubricated lineshaft design and installation practices;

  --modifications of bowl geometry to minimize sand damage;

  --use of mating wear-resistant impeller skirt and bowl throat wear rings; and

  --the introduction, in late 1983, of metallurgically hardened steel pump bearings which are much more resistant to sand than the prior bronze bearing technology.

- These technological improvements, along with the generally used installation and operating practices, can be defined as "recently applied practices". The longest pump endurance to date under these practices was just over 20,000 hours (all reference to hours means rotating or pumping hours), while one-fourth of the pumps lasted at least 7100 hours, demonstrating that current technology is capable of producing pumps that can run for one year under a wide range of resource conditions. However, one-fourth of the pumps lasted less than 1400 hours, and the median life was 4100 hours. The predominance of early failures has kept the mean time-to-failure low. The estimated mean time-to-failure (MTTF) under recently applied practices is about 5000 hours.

- In 1985, Chevron began a pump life-extension program at the Heber Binary Plant. They concluded that the levels of quality control then being applied to high-temperature geothermal lineshaft pumps were not adequate, given the severity of the operating conditions; i.e., that the pumps operate with very little margin for error, so that minor imperfections that would be harmless in less severe service result in premature failure in high-temperature geothermal service. Chevron implemented pump life-extension practices which include the following elements:

  --enhanced assembly quality control;

  --improved installation practices; and

  --semi-continuous monitoring of pump efficiency allowing detection of pump problems while damage is minimal.

  In at least one case, the last practice alone almost doubled the life of a pump.

- The first four pumps to receive the full benefit of these pump life-extension practices were each running without detected problems after more than 4000 hours. It is highly likely that this portends a new breakthrough in lineshaft pump reliability resulting from the elimination of early failures as a major factor in pump life statistics. The estimated near-term mean time-to-failure, predicated on the virtual elimination of failures at less than 4000 hours, is on the order of 8000 hours.

- Lineshaft problems have been rare in pumps with high endurances, but were a significant contributor to early failures. Realization of the full benefits of the pump life-extension program will also likely require improved quality control of lineshaft components and assembly.

- Modeling of a hypothetical multi-well (10 well-no spares, 10 well plus 2 spares, and 10 well plus 5 spares) brine supply system showed that the recently-applied-practices MTTF provides adequate individual pump availability (about 94 percent) as well as adequate brine capacity. This is not surprising since two commercial power plants have been operating for several years using these practices.

  The model indicated that the improvement in estimated near-term MTTF will have little effect on either individual pump availability or brine supply capacity. However, the economic benefit, reflected in reduced pump and maintenance costs, as well as reduction in the number of spare wells required to maintain a reliable brine supply, will likely be a significant factor in the economic viability of existing and future geothermal binary plants.

ELECTRIC SUBMERSIBLE PUMPS

- At the time of the prior survey, several manufacturers of electric submersible pumps had active programs to develop reliable pumps for 365°F (165°C) geothermal service. Developments at that time indicated that such pumps would be available in the near term. However, the cessation of DOE support of pump development, coupled with subsequent economic factors and the perceived small size of the potential market, resulted in the termination of these projects shortly after the conclusion of the prior survey. This was apparently an economic decision rather than an industry consensus that the technology was not viable. The current survey failed to identify any active development work since 1983.

- The experience accumulated in these two surveys indicates that the pieces needed to overcome several serious problems and create a durable electric submersible pump with a ceiling temperature of at least 375°F (190°C) probably do exist, but that a considerable development program may be required to bring them together with sufficient refinement to give consistent high endurances. It appears unlikely that electric submersible pump manufacturers will proceed unilaterally with such development programs. High-temperature geothermal electric submersible pumps with demonstrated reliability comparable to existing lineshaft pump technology do not appear likely in the near term.

DOWNWELL HYDRAULIC TURBOPUMPS

- Close-coupled hydraulic turbopumps (CCTPs) can be set extremely deep, even in highly deviated wells. Their high speed means that high flow rates at high pressure boosts can be generated by smaller diameter pumps, offering the potential for high production rates from smaller diameter wells, resulting in much lower wellfield costs. They appear to be the only downwell pump technology which is likely to be readily extendable to temperatures above 400°F (204°C). Their "wire to pump" efficiency is only about 35 to 50%, compared to the nominal 70 to 75% associated with lineshaft pumps.

- At the close of the prior survey, a Weir, Ltd. CCTP was awaiting trials in the United States. J. Hilbert Anderson, Inc. had bench-tested a prototype CCTP of their own design. The Weir pump was tested repeatedly at East Mesa in 1985 and 1986, with each trial resulting in fairly rapid plugging of the pump by scale. Weir concluded that they were not able to get an adequate setting depth and the testing was concluded. Weir considers the pump to be commercial, and they indicate no further development work is planned. Considerable demonstration effort will be required before any estimate of its high-temperature geothermal reliability will be possible. Lack of funds prevented testing of the J. Hilbert Anderson pump.

- A high-temperature geothermal close-coupled hydraulic turbopump with demonstrated reliability comparable to existing lineshaft pump technology does not appear likely in the near term.

Appendix A

GLOSSARY

This appendix presents a glossary of pump technology and other terms as they are used in this report. The definitions presented here are intended to help the reader who is unfamiliar with downwell pumping technology, but should not be considered as formal definitions. Some of the terminology is specific to particular pump technologies. This will be indicated by the notations "LSP", "ESP", or "CCTP", depending on whether the term is specific to lineshaft pumps, electric submersible pumps, or close-coupled hydraulic turbopumps, respectively.

axial end-play -- (LSP) the vertical (axial) range through which the impellers can be moved, by pulling or pushing on the lineshaft, without contact between the impellers and stationary pump components.

availability factor -- see pump availability factor

backspin test -- a method of assessing the condition of a pump in-situ by measuring the length of time required after pump shutdown for the water in the production column to drain back through the pump.

black paper scale -- a scaling phenomenon at the East Mesa resource resulting from simultaneous production of incompatible brines from different formations. The mixing of these brines produces thin sheets or flakes of black carbonate scale.

boride-hardened bearing -- (LSP) state-of-the-art pump bearing fabricated from low alloy steel metallurgically hardened with boron. Johnston Pump states a proprietary claim to the specific details of the manufacturing process and design of these bearings.

bottom out -- downward motion of the pump impellers which results in their contact with stationary elements of the pump. Bottoming out will lead to rapid failure of the pump.

bowl -- the stationary casting which constitutes the body of the pump.

bowl discharge flange -- the upper flange of the bowl. This flange is precisely machined to provide a register to assure proper alignment of successive bowls.

bowl neck -- taper in the bowl diameter above the diffuser (qv).

bowl shirt -- the lower part of the bowl below the diffuser (qv), which houses the impeller.

bubble tube -- small diameter tube inserted into well along with the pump to allow measurement of the height of the brine column above the pump suction (qv).

cable -- (ESP) the armored conductor which transmits electricity from the wellhead to the downwell electric motor. The cable consists of three conductor wires, elastomeric insulation, and some form of external metallic armor.

centralizer -- (LSP) stationary device inserted about every 20 feet in the lineshaft assembly (qv) to support the enclosing tube (qv) and keep it concentric within the production column (qv).

charge fluid tube -- (CCTP) relatively small diameter tube which supports the close-coupled hydraulic turbopump (qv) in the well and conducts pressurized drive fluid (qv) from the surface charge pump (qv) to the downwell turbine (qv).

close-coupled hydraulic turbopump -- novel downwell pumping system consisting of a high speed centrifugal pump driven by a close-coupled hydraulic turbine. The hydraulic turbine is driven by pressurized drive fluid from the surface charge pump (qv).

collet -- metal ring pressed into the annular gap between the pump impeller (qv) and the pumpshaft (qv) to attach or fix the impeller to the shaft.

diffuser -- stationary arrangement of flow channels in the bowl above the impeller. The diffuser redirects the impeller discharge inward and upward into the suction of the next impeller, simultaneously converting velocity head to pressure head.

diffuser vanes -- the vanes of the diffuser.

discharge bowl -- (LSP) the top bowl in a lineshaft pump, containing the lower terminations of the lineshaft (qv), enclosing tube (qv), and production column (qv).

drive fluid tube -- syn. charge fluid tube

downwell differential pressure regulator -- (ESP) R&D stage component of the surface-pressurized motor protection system (qv) which maintains motor oil pressure slightly above the ambient brine pressure.

downwell turbine -- (CCTP) the close-coupled motive unit for the pump system.

enclosed impeller -- impeller so configured that the vanes and shrouds (qv) form enclosed channels through which the pumped fluid is accelerated by the centrifugal forces created by impeller rotation.

enclosed lineshaft -- (LSP) lineshaft design in which the actual lineshaft (qv) is housed within a concentric tube which excludes geothermal brine from the lineshaft bearings (qv).

enclosing tube -- (LSP) in the enclosed lineshaft design, the enclosing tube houses the actual lineshaft and its bearings, and conducts lubricant (oil) from the wellhead to the discharge bowl (qv).

enclosing tube couplings -- see lineshaft bearings

endurance -- the number of rotating hours that were realized or may be expected from a given pump.

failure -- is defined as that point in the operating life of a pump system when it is decided to remove the pump from the well, either because it is inoperable or for refurbishment. In the initial survey, failure was defined as having occurred when a pump system was no longer capable of operation. However, the recent trend is toward preventive intervention, where the pump system is removed and repaired at the first signs of distress, rather than being run to the actual point of inoperability. Thus a pump which has enjoyed a long and successful run is still considered to have failed at the point in time that it is removed from the well for maintenance reasons.

gas breakout pressure -- the pressure below which gases dissolved in the geothermal fluid come out of solution, forming bubbles.

impeller -- the rotating element in a centrifugal pump.

impeller drift -- (LSP) the gradual upward or downward movement of the impellers over time. Impeller drift is suspected to result from gradual tightening of the lineshaft couplings during pump operation.

impeller shrouds -- the solid faces of the impeller which hold and support the impeller vanes (qv) and form two faces of the flow channels within the impeller.

impeller skirt -- the cylindrical downward extension of the lower impeller shroud which extends downward into the mating bowl throat (qv).

impeller suction -- the face of the impeller through which brine enters, sometimes referred to as the "eye of the impeller" because of its appearance.

impeller wear sleeve -- a sleeve of wear resistant alloy shrink-fitted to the outside surface of the impeller skirt (qv) to combat wear and erosion.

lateral motion -- syn. axial endplay

life-extension practices -- practices including greatly enhanced quality control, more careful installation practices, and semi-continuous monitoring of pump performance index (qv) developed by Chevron at the Heber Binary Plant, which appear to greatly extend the life expectancy of lineshaft pumps.

lineshaft -- (LSP) 1) the actual shaft which transmits torque from the surface mounted motor to the downwell pump. 2) collectively, the lineshaft assembly (qv). 3) adj. relating some term to lineshaft pump technology.

lineshaft assembly -- (LSP) the collective lineshaft component of a lineshaft pump system, including the actual lineshaft (qv), the lineshaft bearings (qv) and enclosing tube (qv), centralizers (qv), and production column (qv).

lineshaft bearing -- (LSP) simple journal bearing which supports the lineshaft and keeps it concentric within the enclosing tube. Lineshaft bearings are externally threaded and also serve as enclosing tube couplings.

lineshaft coupling -- (LSP) threaded fitting which joins two sections of lineshaft.

lineshaft seal -- (LSP) dynamic seal at the wellhead which seals the annulus between the rotating lineshaft and the stationary enclosing tube. This seal must contain the hot, pressurized oil which fills this annulus.

lubrication tube -- syn. enclosing tube

mean time to failure (MTTF) -- the arithmetic mean (average) endurance of a particular group or class of pumps.

motive pump -- (CCTP) the aboveground pump which pressurizes the drive or charge fluid (qv) which drives the downwell turbine (qv).

motor lead connectors -- (ESP) the device(s) by which the power cable conductors and electric motor leads are joined inside the motor housing.

open lineshaft -- (LSP) lineshaft design in which the lineshaft bearings (qv) are exposed to and lubricated by the pumped fluid. Not used in modern high-temperature, high-capacity geothermal lineshaft pumps.

packer -- mechanical device installed in a well to seal one part off from another. Close-coupled hydraulic turbopumps are set into packers to isolate the pump suction (qv) from the pump discharge.

packoff -- (ESP) the sealing device at the wellhead through which the power cable passes into the wellbore.

performance index (PI) -- ratio of actual pump efficiency, calculated from wellhead pressure and flow data, and the theoretical (pump curve) efficiency. Sudden drops in performance index can indicate the onset of pump adjustment problems before serious damage has occurred.

pothead -- (ESP) the termination of the power cable at the electric motor housing.

production column -- (LSP and ESP) large diameter pipe which supports the weight of the downwell pump and conducts the pump discharge to the surface.

production column elongation -- (LSP) the elastic elongation of the production column (qv) while the pump is operating due to the weight of the water column contained within the production column.

production tube -- syn. production column

protector -- (ESP) the component between the electric motor and pump sections of an ESP. It seals the motor shaft against brine intrusion and allows for thermal expansion of the oil which fills the motor section.

pump -- 1) the actual downwell centrifugal unit which increases brine pressure. 2) the entire pump system including all ancillary components.

pump availability factor (A%) -- the fraction of the time which a single downwell pump is operable, relative to the total length of time in which the service of the pump was desired. The pump availability factor is defined as

$$A\% = 100[TTF/(TTF + TTR)]$$

where: TTF = Time to fail; and
TTR = Time to repair or replace.

pumpshaft -- the shaft within the pump unit.

pump bearing -- journal bearing within the pump bowl. The pumpshaft runs against the pump bearings.

raised throat -- (LSP) bowl geometry in which the throat (qv) is machined into an elevated neck which is an integral part of the bowl casting. Thus, the throat is raised above the bowl discharge flange (qv). To Radian's knowledge, this term is unique to this report.

recessed throat -- (LSP) bowl geometry in which the throat (qv) is machined downward into the bowl neck (qv), and is thus recessed below the bowl discharge flange (qv). To Radian's knowledge this term is unique to this report.

relative shaft elongation -- the difference between the shaft elongation (qv) and the production column elongation (qv).

reliability -- this term has a precise statistical meaning as well as a more common, strictly qualitative, usage. Unless expressly indicated otherwise, the term reliability as used in this report will indicate the latter usage, rather than the more strict statistical meaning.

run time -- see endurance

sand lock -- cause of failure in which the pump unit becomes so clogged or packed with sand that the impellers will not rotate.

sand lugs -- protrusions on the top face of the bowl discharge flange (qv) which serve to break up cyclonic sand recirculation on the underside of the impeller lower or suction shroud

seal section -- syn. protector

setting depth -- distance from the wellhead to the pump suction (qv).

shaft elongation -- (LSP) the elastic elongation of the lineshaft (qv) during pump operation due to dynamic thrust and the weight of the water column supported by the impellers.

specific speed -- non-dimensional index used to classify impeller types and geometry. Specific speed may be expressed as

$$N_s = (RPM)(GPM)^{0.5}(Head)^{-0.75}$$

where:
- N_s = specific speed;
- RPM = Full-load pump speed;
- GPM = GPM at best efficiency point on the pump performance curve; and
- Head = Pressure head per stage at the best efficiency point on the pump performance curve.

spider -- (LSP) in open lineshaft (qv) design, the stationary device which houses the lineshaft bearings (qv).

stick -- (LSP) production columns and lineshafts come in 20 foot lengths, while enclosing tubes are five feet long. At the factory, four sections of enclosing tube are coupled together by lineshaft bearings (qv), which serve the dual role of enclosing tube couplings, and the lineshaft is inserted. The resulting assembly is commonly called a "stick".

suction bowl -- the bottom-most bowl in a multi-stage pump assembly.

suction screen -- slotted or perforated sieve or screen installed across the intake of the suction bowl to prevent large objects from entering the pump.

surface-pressurized motor protection system -- (ESP) R&D stage system by which the downwell electric motor is pressurized with oil continuously supplied from the surface to prevent brine intrusion into the motor.

tandem adaptor -- (LSP) split-case device allowing field coupling of pumpshafts of pump assemblies which have more stages than can be assembled on a single pumpshaft.

tensioning -- (LSP) the practice of applying tensile load to the enclosing tube (qv) of a lineshaft assembly in order to keep this tube as straight as possible, reducing lateral loading and wear on the lineshaft bearings (qv).

top out -- upward motion of the pump impellers which results in their contact with stationary elements of the pump. Topping out will lead to rapid failure of the pump.

throat -- machined receptacle above the diffuser (qv) into which the impeller skirt (qv) is inserted.

throat wear liner -- wear-resistant alloy sleeve or ring shrink-fitted into the throat (qv) to combat wear and erosion.

thrust bearing -- motor bearing which receives the thrust transmitted from the impellers along the connecting shaft(s).

thrust-compensated impeller -- impeller design in which the impeller contains perforations or ports which serve to transmit thrust-balancing pressure to the backside of the impeller, reducing the loading on the motor thrust bearing (qv). Thrust-compensated impellers are more susceptible to erosion.

tolerance accumulation -- the stochastic phenomenon whereby, if the tolerance limits on the individual components of a complex assembly are too large, then on some occasions the assembled device may be out of tolerance, may even experience mechanical interference, despite the fact that each individual component was within tolerances.

unitized pump -- (LSP) pump assembly in which two (or more) pumpshafts, each with assembled bowls and impellers, are shop-aligned and spliced to create a single very long pump. Unitized pumps are inserted into lengths of casing to provide rigidity and support during transport and installation.

wear rings -- see impeller wear sleeve and throat wear liner

wellhead seal -- syn. lineshaft seal

Appendix B

CONTRIBUTORS TO THE CURRENT SURVEY

Numerous people voluntarily contributed to this updated downwell pump database. Some of these individuals requested that their contributions be used without direct attribution. Others provided copies of internal memoranda and/or material photocopied from other documents, again with the request that this information be used without direct attribution. In accord with these requests, the information presented in Sections 3 and 4 was not referenced. The contributors are listed below.

CONTACT	COMPANY OR AFFILIATION	LOCATION
J. H. Anderson	J. Hilbert Anderson, Inc.	York, PA
Bill Asper	Mammoth-Pacific Power Co.	Mammoth Lakes, CA
Keith Auyeung	Chevron Resources Co.	San Francisco, CA
Jonne Berning	Electric Power Research Institute	Palo Alto, CA
John Bigger	Electric Power Research Institute	Palo Alto, CA
Bruce Brookbank	TRW-Reda Pumps	Bartlesville, OK
Neil Elliot	Byron Jackson Pumps	Tulsa, OK
Bob Fisher	Trico Industries (formerly Kobe Pump)	Huntington Park, CA
Jack Frost	Johnston Geothermal Pump Div.	Azusa, CA
Robert Hanold	Los Alamos National Laboratory	Los Alamos, NM
Tom Hinricks	Magma Power Co.	Escondido, CA
Rubin Menashian	Peabody-Floway	Fresno, CA
Douglas Miller	Ormat Systems, Inc.	Sparks, NV
Michael Pierce	Magma Power Co.	East Mesa, CA
David Riddet	Weir Pumps Ltd.	Glasgow, Scotland
Rick Saranda	Gould Pumps, Inc.	City of Industry, CA
Randy Shetler	Oil Dynamics, Inc.	Tulsa, OK
Douglas Werner	Barber-Nichols Engineering	Arvada, CA
Joe Vandervier	Centrilift-Hughes	Claremore, OK
Yona Yahalom	Ormesa	Holtville, CA

In addition, the following published articles provided much useful background information.

1. J. Frost. "Introduction to Geothermal Lineshaft Production Pumps", Course on Pumping of Geothermal Brines, 21-23 March 1984, Geothermal Resources Council, Davis, CA, March 1984.

2. A.A. Grant and A.G. Shell. "Development, Field Experience, and Application of a New High Reliability Hydraulically Powered Downhole Pumping System", SPE 11694, March 1983.

3. A.A. Grant and D.A. Riddet. "High Reliability Hydraulic Turbine Powered Downhole Pump for the Geothermal Industry", Course on Pumping of Geothermal Brines, Geothermal Resources Council, Davis, CA, March 1984.

4. R.J. Hanold. "Geothermal Pumping Systems", (Proceedings) DOE Geothermal Technology Conference (CONF-8410179-1), El Centro, CA, 16 October 1984.

Appendix C

DOWNWELL LINESHAFT PRODUCTION PUMP PERFORMANCE CURVES

This appendix contains performance curves for the two Peerless lineshaft pump models used during the Phase I development of the Heber Binary Plant wellfield. These curves were provided by Chevron, the wellfield operator. Johnston Pump declined to provide pump curves.

The curves presented in this appendix were reproduced from the best copies available to Radian.

Peerless Pump
A Sterling Company

1200 Sycamore Street
Montebello, CA 90640

2005 Northwestern Avenue
Box 7026
Indianapolis, IN 46206

HYDRAULIC PERFORMANCE WARRANTY

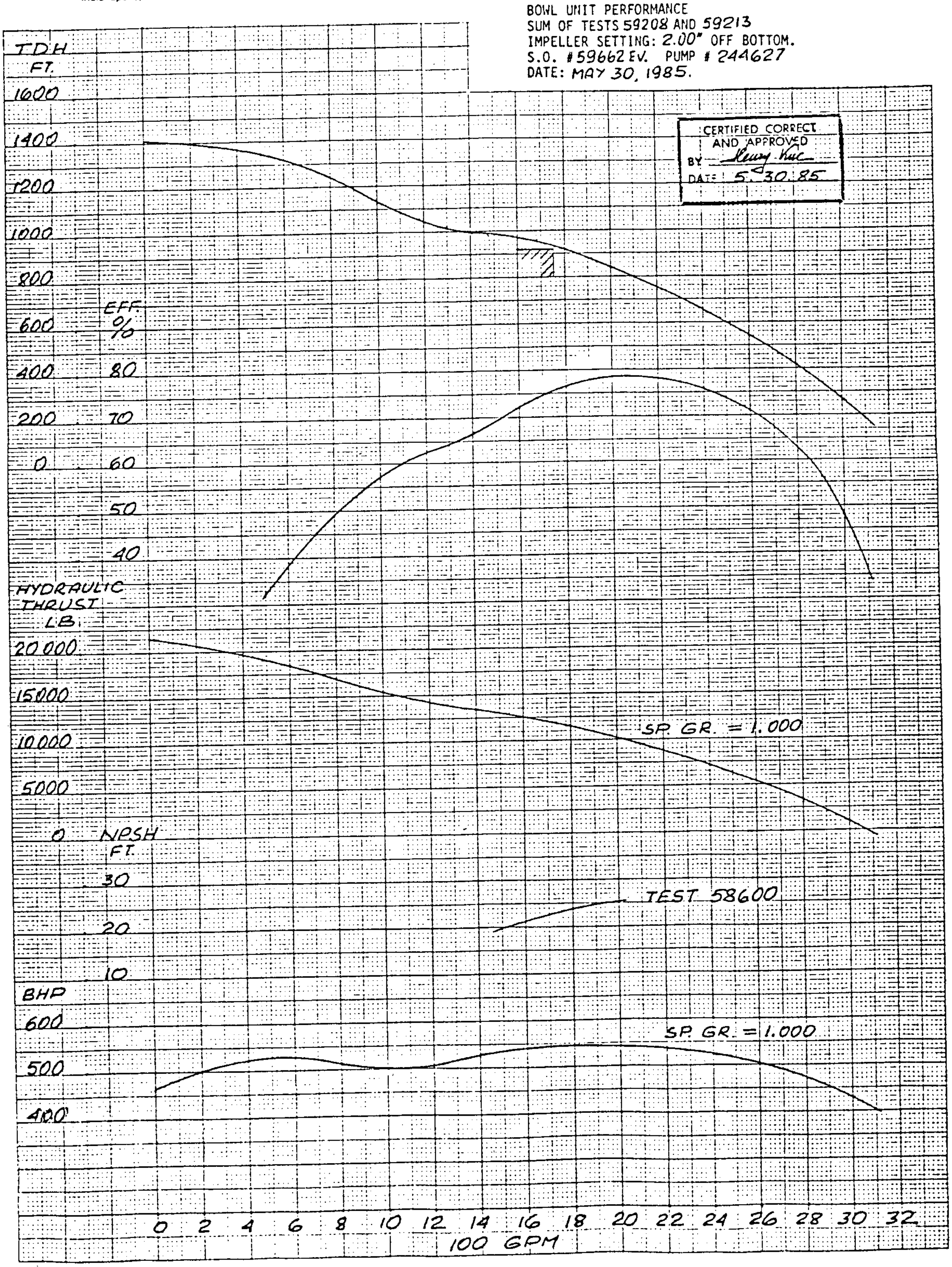
Guaranteed at designated point only and is contingent on:

1. Proper and adequate flow to pump suction.
2. Proper submergence and NPSH available.
3. Fluid free of gas, air and abrasive matter.
4. Impeller with proper lateral adjustment.

PUMP TYPE	12 HXHG
IMPELLER	4600879
RPM	1790
BOWL	4600877
STAGES	20

CUSTOMER
CHEVRON GEOTHERMAL
CO. OF CALIFORNIA
P.O. # GTC 761280 BK
WP ~~102~~ 105
ZONE I

BOWL UNIT PERFORMANCE
SUM OF TESTS 59208 AND 59213
IMPELLER SETTING: 2.00" OFF BOTTOM.
S.O. #59662 EV. PUMP # 244627
DATE: MAY 30, 1985.



Peerless Pump
A Sterling Company
1200 Sycamore Street
Montebello, CA 90640
2005 Northwestern Avenue
Box 7026
Indianapolis, IN 46206

HYDRAULIC PERFORMANCE WARRANTY

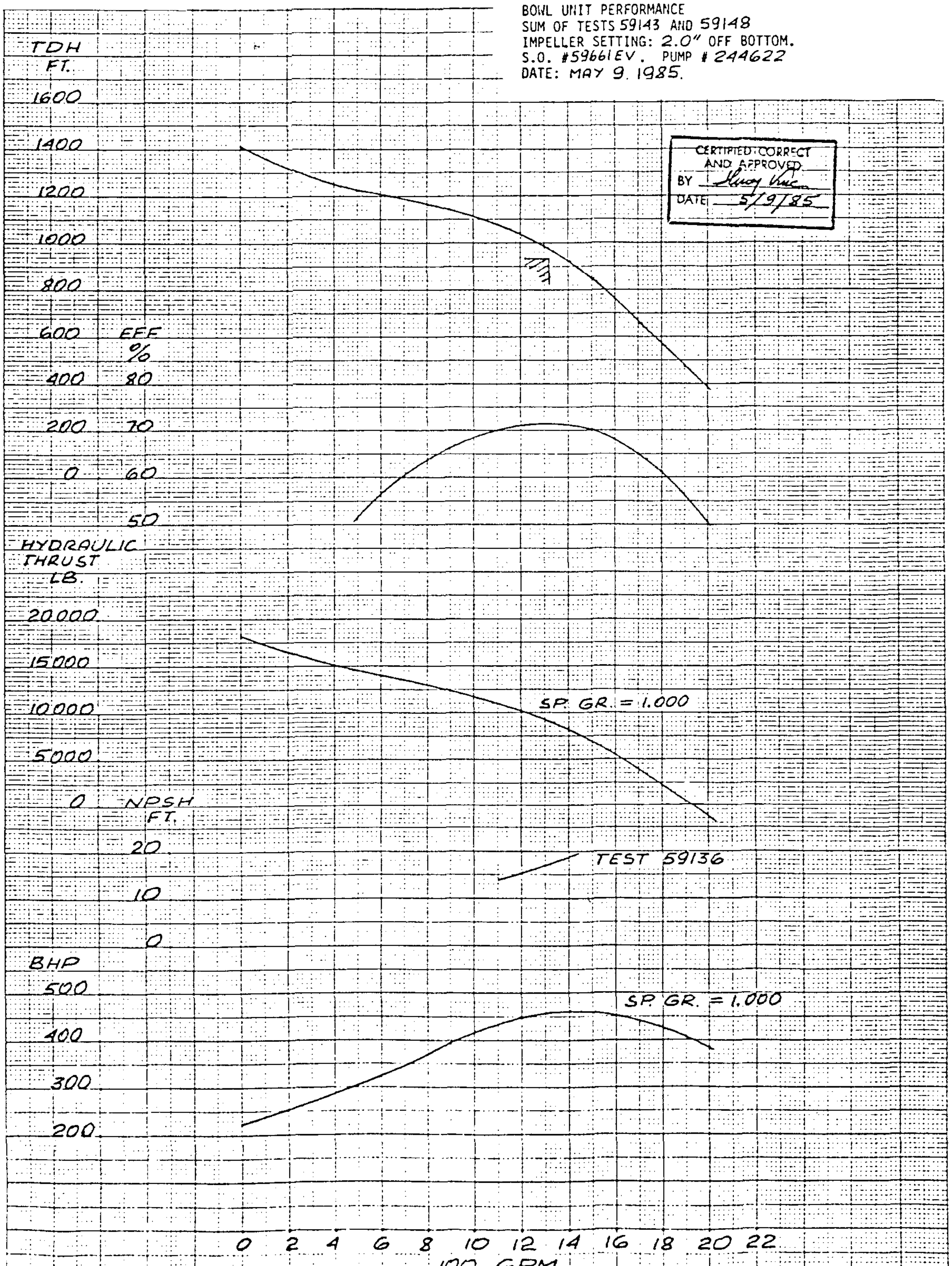
Guaranteed at designated point only and is contingent on:

1. Proper and adequate flow to pump suction.
2. Proper submergence and NPSH available.
3. Fluid free of gas, air and abrasive matter.
4. Impeller with proper lateral adjustment.

PUMP TYPE 12 HXLG
IMPELLER 4600885 RPM 1790
BOWL 4600883 STAGES 18

CUSTOMER
CHEVRON GEOTHERMAL
CO. OF CALIFORNIA
P.O. # GTC 761280 BK
WP ~~104~~ 102
ZONE II

BOWL UNIT PERFORMANCE
SUM OF TESTS 59143 AND 59148
IMPELLER SETTING: 2.0" OFF BOTTOM.
S.O. #59661EV. PUMP # 244622
DATE: MAY 9. 1985.



Appendix D

BIBLIOGRAPHY

PREFACE

As a part of this project, Radian conducted an automated literature search for information relating to the reliability of high-temperature geothermal brine production pumps. This search used the following terms: downwell, downhole, pumps, failure causes, availability, reliability, testing, geothermal wells, failure mode analysis, and high-temperature. This search concentrated on the period 1983 and later, the time not covered in EPRI report AP3572.

The search explored the following databases: APLIT, APILAT, EPIA, Ei Engineering Meetings, NTIS, DOE ENERGY, COMPENDIX, and ENERGYLINE.

This search yielded approximately 30 citations with abstracts. The abstracts were reviewed and those documents which were clearly not relevant were rejected, leaving ten citations. The small number of possibly relevant documents reflects the relatively small amount of research since the early 1980's.

BIBLIOGRAPHY

Anon. "Listing of EPRI Geothermal Power Systems Research & Development Projects", Proceedings: Eighth Annual Geothermal Conference and Workshop (EPRI AP-3686), Seattle, WA, June 25-29, 1984, 4p.

This paper lists 27 different projects, both completed and underway, on geothermal power systems. These projects include geothermal exploration methods, control of scaling in geothermal systems, geothermal chemistry analysis, waste heat rejection, hydrocarbon expander turbines, rotary separator turbines, binary-cycle demonstration plants, removal of hydrogen sulfide from geothermal systems, wellhead conversion systems, and geothermal downwell pump reliability.

Ellis, P.F., T.F. Green, and H.J. Williamson. Geothermal Downwell Pump Reliability: State-of-the-Art Assessment, Final report. EPRI AP-3572, Radian Corporation, Austin, TX, June 1984, 104p.

The current and near-term state of the art of high-temperature (250-365°F, 121-185°C) geothermal downwell pumps was assessed by investigating the causes of failure of more than 70 lineshaft and electric submersible pumps. Recent developments in novel pump technology were also reviewed. The data were used to estimate the likely near-term availability factor of high-temperature downwell pump systems. In an industry survey completed in April 1983, it was found that both lineshaft and electric submersible pumps had encountered repeated and severe difficulties when pumping fluids above 250°F (120°C) in geothermal wells. Sixty-three percent of the lineshaft and 83% of the electric submersible pumps failed. The average run life was less than two months. However, both lineshaft and electric submersible pumps had undergone recent developments that strongly suggest that a significant improvement has occurred. Near-term lives were estimated based on engineering judgment in light of recent developments. It was concluded that near term lineshaft and electric submersible pumps will have a probable average run life of 6-12 months in resources up to 365°F (185°C). Statistical modeling showed that if average run lives reach 6-12 months as expected, then the near-term pumping technology is capable of providing reliable brine supply to a large geothermal binary power plant.

Ellis, P.F., C.C. Smith, R.C. Keeney, D.K. Kirk, and M.F. Conover. Corrosion Reference for Geothermal Downhole Materials Selection, DOE/SF/11503-1, Radian Corporation, Austin, TX, March 1983, 332p.

Geothermal downhole conditions that may affect the performance and reliability of selected materials and components used in the drilling, completion, logging, and production of geothermal wells are reviewed. The results of specific research and development efforts aimed at improvement of materials and components for downhole contact with the hostile physicochemical conditions of the geothermal reservoir are discussed. Materials and components covered are tubular goods, stainless steels and non-ferrous metals for high-temperature downhole service, cements for geothermal wells, high-temperature elastomers, drilling and completion tools, and downhole pumps.

Frost, J., "Introduction to Geothermal Lineshaft Production Pumps", Course on Pumping of Geothermal Brines, 21-23 March 1984, Geothermal Resources Council, Davis, CA, March 1984, 18p.

An introduction to the basic concepts, terms, and formulas used to select and apply geothermal lineshaft production pumps. Since geothermal installations vary greatly, this discussion is limited to general considerations common to most applications.

Grant, A.A. and D.A. Riddet. "High Reliability Hydraulic Turbine Powered Downhole Pump for the Geothermal Industry", Course on Pumping of Geothermal Brines, Geothermal Resources Council, Davis, CA, March 1984, 18p.

Artificial lifting of brine by means of deepset downhole pumps is a widely used method of extracting the maximum output from the well. Pumping of hot brine requires high downhole horsepowers with a need to operate in a relatively narrow bore well. Various types of pumps have been operating in geothermal wells, but until recently pump configurations used have had application and reliability limitations which can have a marked effect on plant viability and performance. The authors describe the concepts incorporated to overcome these limitations in an entirely new hydraulically powered downhole pumpset patented by Weir Pumps Ltd., Glasgow, UK. Development and field operating experience are discussed in relation to reliability objectives.

Grant, A.A., and A.G. Shell. "Development, Field Experience, and Application of a New High Reliability Hydraulically Powered Downhole Pumping System", SPE 11694, March 1983, pp275-284.

Expansion of secondary oil recovery operations both offshore and onshore and geothermal energy recovery has lead to an increasing requirement for artificial lift systems. A large proportion of these systems incorporate submersible pumpsets downhole either for pumping of oil or for the lifting of water from subterranean aquifers. Many of these applications are characterized by the requirement for high horsepower installed downhole and/or, particularly offshore, installation of the pumpsets in highly deviated wells, frequently at high temperatures. Experience has shown that the technical difficulties encountered in the extrapolation of existing submersible electric motor pump technology are severe, resulting in poor service reliability and high maintenance cost. The authors describe the concepts incorporated to overcome these limitations in an entirely new hydraulically powered downhole pumpset patented by Weir Pumps Ltd, Glasgow, UK. Development and field operating experience are discussed in relation to reliability objectives. The paper reviews the range of applications for the new technology as well as the types of systems and completions which will take full advantage of the features of this new concept.

Hanold, R.J. "Geothermal Pumping Systems", (Proceedings) DOE Geothermal Technology Conference (CONF-8410179-1), El Centro, CA, 16 October 1984, 15p.

After successful field testing of a prototype pressurized lubrication system designed to prevent brine intrusion and loss of lubricating oil from the motor and protector sections of electric submersible pumps, a second-generation lubrication system has been designed, fabricated, and laboratory tested. Based on a sensitive downhole pressured regulator, this system is not depth limited and it accurately controls the differential pressure between the motor oil and the external brine. The first production lengths of metal-sheathed power cable have been fabricated by Halpen Engineering and delivered to REDA for testing and evaluation. Laboratory tests performed on prototype metal-sheathed cable samples have demonstrated the durability of this power cable design. The East Mesa Pump Test Facility is currently being activated for high-horsepower pumping system tests that are scheduled to commence during the first quarter of FY 85. A 300-hp REDA pumping system equipped with a pressure regulator controlled lubrication system and and a metal sheathed power cable is being fabricated for testing in this unique facility.

Harvey, C., W. McBee, and H.B. Mathews. Sperry Low Temperature Geothermal Conversion Systems, Phase I and Phase II, Volume V, Component Development, Final report. DOE/ET/27125-T2-V.5, Sperry Research Center, Sudburry, MA, 1984, 253p.

The fundamental inventions which motivate this program are system concepts centered on a novel heat engine cycle and the use of downwell heat exchange. This document concentrates on the component development for the Sperry Gravity-head Generator. Components discussed include downwell heat exchangers, conduits, instrumentation and the turbine-pump unit (TPU). The TPU for the gravity-head generator requires a different kind of turbine because of the large flow-rate and small pressure drop through the TPU. The design study of a Francis turbine to meet these requirements is reported, along with an extensive investigation of the integrity of the downwell heat exchanger which is an integral part of this novel system.

Hosang, G.W., A.R. Stetson, "Development of Bushings and Bearings for Use in Hot Brine Pumps". Addendum to Materials Selection Guidelines For Geothermal Energy Utilization Systems, Part II: Proceedings of the Geothermal Engineering and Materials (GEM) Conference, San Diego, CA, 6-8 October 1982, DOE/ET/27026-2, Radian Corporation, Austin, TX, March 1983, pp176-186.

The exploitation of geothermal resources often requires that naturally heated subterranean brines be pumped to the surface from depths of up to 6000 feet underground while minimizing heat losses and maintaining sufficient fluid pressure to prevent boiling. To accomplish this requires the use of downhole brine pumps capable of months of uninterrupted operation. Significant problems have occurred with pump lineshaft bearings in the geothermal wells. The objective of this research program was to determine the nature of the problems associated with commonly reported premature failures of water-lubricated downhole pump bearings. A series of bearing endurance tests was performed on a variety of candidate bearing materials. These tests were accomplished using test rigs specially developed to simulate actual geothermal field conditions and to isolate specific bearing wear problems.

Pascual, W., J. Brady, A. Crumley, and C. Nelson. "Development of High Temperature Electric Submersible Pump", SPE 11736, March 1983, pp661-666.

Today's oil well production is exposing artificial lift equipment to higher and higher downhole operating temperatures as a result of deeper wells and enhanced recovery methods such as steam floods and insitu combustion. KOBE is pursuing a development program that will result in an ESP capable of operation in these high-temperature applications. Tests under simulated conditions of 350°F continuous service are being conducted and have already resulted in design and material developments that make this operation practical. This paper reports on the progress achieved and the future testing that is to be done.