TUTORIAL ON MULTIPHASE GAS-LIQUID PUMPING

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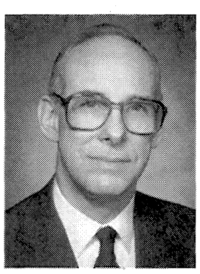
ABSTRACT

Pumping crude oil and gas without prior separation greatly reduces the machinery and platform space required. Multiphase pumps can be located at the surface or subsea and are designed primarily as a) two-screw rotary or b) multistage rotodynamic machines that can ingest 90 to 100 percent gas by volume. Each type has advantages and disadvantages, as can be seen from the growing body of test experience, both in the laboratory and the field. Six authors contributed the following five sections, in an effort to provide all facets of the subject in one tutorial.

SECTION I: OVERVIEW OF MULTIPHASE PUMPING



Paul Cooper is Director of Advanced Technology for Ingersoll-Dresser Pump Company, Phillipsburg, New Jersey. His responsibilities are the development of new pump products along with the hydraulic and acoustical improvement of centrifugal pumps. Dr. Cooper joined Ingersoll-Rand's research center in Princeton in 1977, where he conducted research on pumps and turbines. Prior to that, he had hydraulic design responsibility for aircraft fuel and oil field pumps at TRW Incorporated.

Dr. Cooper received a B.S. degree (Mechanical Engineering) (1957) from Drexel University, an M.S. degree (1959) from Massachusetts Institute of Technology, and a Ph.D. (1972) from Case Western Reserve University, Dr. Cooper is an ASME Fellow, having served as chairman of the ASME Fluids Engineering Division. He recently received ASME's Henry R. Worthington Medal for achievement in the field of pumping machinery.

SYNOPSIS

In many oil fields, the wells produce mixtures of gas and oil in varying proportions. Pumps that can handle crude oil with large and time-varying entrained gas, expressed in terms of the gas volume fraction (GVF), have been developed. These can replace a host of equipment currently utilized to a) separate the gas from the oil and b) compress the gas, and, c) pump the oil. Additionally, multiphase pumps can deliver the raw effluent over long distances, making them a possible cost-effective solution for developing small or remote fields. They can be deployed subsea or at the surface, the latter installations being on platforms or onshore.

Two fundamentally distinct pumping concepts are employed: 1) multistage rotodynamic helico-axial pumps, and 2) rotary positive displacement two-screw pumps. Both are capable of ingesting fluid mixtures that consist of well over 90 percent gas by volume and are briefly described as follows:

- *Rotodynamic Pumps (Concept 1)* have many stages that progressively compress the mixture dynamically; i.e., pump speed generates the required pressure, more speed being necessary to pump higher GVFs to the same pressure. The full range of GVF can be accommodated. Changes in GVF produce corresponding changes in torque, which are accommodated in the design of the machine.
- *Rotary Screw Pumps (Concept 2)* have two screws that are timed by external gearing, and they deliver whatever they ingest against a fixed exit pressure. They can pump 100 percent gas for indefinite

periods of time against this exit pressure. Being positive displacement machines, they produce very little torque fluctuation in response to GVF variations.

As alluded to above, each concept has operational and/or durability and maintenance advantages and disadvantages. In order to verify the operability of these concepts in a realistic multiphase pumping environment, pumps of various manufacturers have been extensively tested over a wide range of fluid conditions, including operation in a multiphase laboratory test loop and in the field. This testing encompasses 1) manufacturer's tests, 2) shakedown, acceptance, detailed performance evaluation in the multiphase loop; and 3) long term endurance testing in the field. Results provide a) operational experience re performance characteristics, durability and maintenance, b) confidence for future field deployments and c) direction for further pump development.

ECONOMICS, HISTORY, PRESENT STATUS, AND FUTURE PROSPECTS

Peter Lovie and Ken Barker, in their article entitled, "Multiphase pumping–Where to now?" summarize the contributions to the article by eight industrial experts as follows [1]:

- Development of concepts demonstrated over 20 years ago has gone on since that time, and they are now commercially available and can handle a wide range of production conditions; yet only a few units have so far been purchased and employed by the oil industry.
- Low oil prices have limited development of subsea multiphase pumping (MPP), which many oil companies regard as risky. Consequently, there is little MPP operating experience.
- Currently, two major evaluations of multiphase pumps in subsea service are underway, namely, Shell (Draugen - Norwegian North Sea) and Agip (Prezioso - Offshore Sicily).
- There are MPP units on surface field trials in many onshore and offshore platform locations. Testing is being done both independently and through joint industry projects.
- Turnkey approaches are being employed offshore, where contractors with the necessary skills take the responsibility for system selection, detailing and implementation. Subsea systems are an example of the need for this approach.
- Multiphase pumps must be packaged to address different operating conditions as to fluid mix, GVF, flowrate, pressure rise, etc. Changes in these conditions during the life of the field are accommodated by design and appropriate change-outs of pump components.

Future prospects, therefore, depend on the outcome of existing MPP projects, and on whether and when sufficient MPP experience can be accumulated to enable users to pursue with confidence the manifest economic benefits that reliable MPP can provide. Pump manufacturers now have the technology and are seeking enough applications to establish the needed field experience. It is the aim of these authors to help break this impasse by informing the user of a) the fundamental phenomena of MPP in general and how it has affected the design of pumps; b) experience with each of the two major pump types designed for oil field MPP, namely the rotodynamic helico-axial and the rotary two-screw machines; and c) experience gained in testing and operating these pumps.

MIXTURE MAKEUP AND OCCURRENCE

The raw product of oil fields is made up not only of the crude oil itself. Other liquids, including water are often present, requiring appropriate attention to the choice of materials of construction employed in multiphase pumps for pumping this raw effluent. However, the major influence on pump geometry is the volume flowrate that must be ingested and the fractions of this volume that are liquid and gas. The terms in use, their definitions and interrelationships are summarized here:

- Gas volume fraction (GVF) is the ratio of the volume of gas to that of the total volume of liquid plus gas, under the actual operating conditions of temperature and pressure. (The term ßs is used in the following section to denote the GVF at the pump suction or inlet connection.)
- Gas-liquid ratio (GLR) is the ratio of the volume of gas to that of the liquid only.
- Gas-oil ratio (GOR) is the mass of gas contained in the liquid (expressed as standard cubic feet per barrel of oil (scf/bbl)). The standard conditions are a temperature of 60°F (460 + 60 = 520 deg absolute) and a pressure of 14.7 psia.

Often it is the GOR that is quoted, as it involves the relative masses of the constituents. Changing temperature and pressure then lead to different volumes, expressed as GVR or GLR. The conversion from GOR is as follows:

$$GLR = GOR \times [(460 + \text{temperature in degrees Fahrenheit}) / 520] \times (14.7 / \text{absolute pressure, psia}) / (5.615 \text{ cu ft per bbl}); \text{ and}$$

$$GVF = GLR / (1 + GLR).$$

For example, if the GOR is 600; the temperature, 120°F; and the pressure, 1000 psig;

$$GLR = 600 \times [(460 + 120) / 520] \times [14.7 / (1000 + 14.7)] / 5.615 = 1.727, \text{ and}$$

$$GVF = 1.727 / (1 + 1.727) = 0.633.$$

INTRODUCTION TO THE FOLLOWING SECTIONS

With the foregoing development as background, the reader is equipped to deal with the remaining sections of this tutorial. These are briefly summarized as follows:

- Section II is an exposition of the fundamentals of MPP via rotodynamic means; including the fluid dynamical considerations that must be taken into account, depending on the value of GVF or $ß_s$, which can be anywhere in the range from zero to 1.0 (or zero percent to 100 percent gas.) There are different design approaches, each appropriate to a portion of this range. MPP of crude oil and substances other than crude oil is described in order to illustrate how the design conclusions are reached. *This section stands as a basic reference for MPP. The reader interested simply in the end result for oil field MPP should feel free to skip over to Sections III, IV and V.*
- Section III deals with the helico-axial design approach for subsea and surface MPP from oil fields. This configuration, as indicated earlier, is the appropriate rotodynamic geometry for the entire GVF range, and is essential for successful pumping rotodynamically at the upper end of this range. It is described in detail from both the hydraulic and mechanical design viewpoints.
- Section IV is an exposition of the rotary positive displacement two-screw approach to MPP. Applicable to the entire GVF range, it is described in detail in this section.
- Section V describes operation and testing procedures and experience in both the laboratory and the field of a major oil company.

SECTION II. EFFECT OF FLUID CONDITIONS ON PUMP TYPES AND DESIGN STRATEGY

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Mr. Schiavello was co-winner of the H. Worthington European Technical Award in 1979, with a paper on pump suction recirculation. He has written several significant papers and lectured at seminars in the area of pump recirculation. He is a member of ASME, AIAA, Société Hydrotechnique de France (SHF), and the International Association for Hydraulic Research (IAHR), he has served on the International Pump Users Symposium Advisory Committee since 1983.

Mr. Schiavello started in the Research and Development Department of Worthington Nord (Italy) dealing with pump hydraulic research and design. In 1982, he joined the Central Research and Development of Worthington, McGraw Edison Company (U.S.A.), and continued with Dresser Pump Division as Manager of Fluid Dynamics. In 1993, he was appointed to his present position.

INTRODUCTION

Background

A thorough discussion of the pumping aspects (performance, design) with two-phase flow for rotodynamic pumps (having impellers with vanes) is presented in an earlier publication [2], from which this section is largely extracted.

When free gas is present in a liquid being pumped, the head, power and efficiency of rotodynamic pumps decreases. The decrease in head is greater than that which can be associated with the decrease in average density of the liquid-gas mixture. The decrease in efficiency suggests that some additional loss mechanisms arise when gassy liquids are pumped. The pump performance decreases continuously as the gas volume increases until at a certain critical gas content $ß_{s,cr}$ [where $ß_s = q_s / (q_s + Q)$; q_s = volumetric capacity of air at suction pressure and temperature; and Q = liquid capacity] the pump loses prime.

The above trend is common to radial-, mixed-, and axial-flow type pumps either in single- [3] or in multistage configurations [4]. In most cases of conventional design, the critical gas content $ß_{s,cr}$ ranges from 5.0 percent to 8.0 percent and rarely 10 percent, depending on pump type and design operating point (i.e., capacity as fraction of the best efficiency capacity) and suction pressure. It can be seen from experimental data [2] that as the gas content becomes high, the range of capacities in which the pump can operate continuously decreases. Basically the operating range appears to be limited by two phenomena: 1) gas locking or "choking" above bep capacity and 2) instability in the head-capacity curve (positive slope), which causes surge [3].

All the above phenomena can be detected for different two-phase flow media with noncondensable gas (e.g., air-water) characterized by a large difference in density between the gas and the liquid phase. Experiments on model pumps with condensable steam-water media showed performance degradation with increasing vapor content, but less marked than that with noncondensable gas-water media, especially at vapor content below 20 percent. With reference to rotary pumps, namely piston pumps and screw pumps [Section IV], the performance is in general less sensitive to gas volume, as fraction of a total volume.

Two-Phase Pumping Applications

Two-phase pumping covers a large spectrum of pump operation and applications. In certain situations the entrained gas is an unwanted by-product in the pumping process, e.g., due to inadequate stuffing box sealing, loose fitting of suction pipes, vortices in the suction pit, pulp or slurry as the pumped medium. In these cases usually the gas content is relatively small, up to about 10 percent and the suction pressure is also low, viz., around 1.0 bar. For rotodynamic pumps, the problem can be solved with appropriate installation or modification of the impeller [2, 4].

There are plants where the gas-phase is an intrinsic component of the process such as: a) chemical reactors operating in a highly oxygenated environment, b) biological water treatment plants, c) fermentation plants for bio-protein production. The pump operates as a circulation pump for cooling the reactor (fermenter) and is required to handle gas contents ranging from 15 percent (chemical reactions) up to 50 percent or even more (bio-industry). Suction pressure usually ranges from 2.0 to 20 bar. Moreover, multiphase media, which include gas and liquid and suspended solid particles (microcrystals) or micro-organisms (bacteria) must be softly pumped. The pump is of special design and assembly, using properly selected materials.

In the oil industry, a very special two-phase application is pumping the oil-gas mixtures coming out of production wells. The composition of the fluid in this case is difficult to define and varies over time, reaching gas volume percentages as high as 90 to 100 percent. The intake pressure is generally high, even up to 100 to 200 bar. The present conventional designs are not adequate, due to their inability to pump high gas volume fraction media.

In the past decade, studies have been carried out to develop multiphase pump sets of special design for very high gas content. The first step of the field exploration has been related to onshore installations, both with rotary screw pumps [5] and rotodynamic helico-axial pumps [2]. The objective here is first to move to offshore installation, the final objective being to develop pump modules for subsea installation.

The design and development of efficient and reliable rotodynamic and rotary pumps with acceptable operating range for the above duties is a technical challenge for the pump industry. This section is mostly focused on rotodynamic pumps with the aim of making a contribution in the area by a) outlining new theoretical insights and design approaches that have led to a successful multiphase flow pump for medium gas content, and b) discussing the design problems of pumps for high gas content.

The major issues of both design and performance for screw pumps, which have emerged as the rotary pumps most suitable for pumping highly gas volume fraction media, are discussed in Section IV [5].

THEORETICAL INSIGHTS ON TWO-PHASE BUBBLY FLOW PHYSICS

Head Degradation Mechanism at Low Gas Content

The first rational attempt to explain the mechanism of head degradation in centrifugal pumps on the basis of flow visualization and to develop a two-phase head equation based on the first principles of thermo-fluid dynamics in turbomachinery was proposed by Murakami and Minemura [3]. A semi-empirical approach for predicting the head degradation from low to high gas (steam) content is described by Muench [6].

Simple analytical models for the low gas content and homogeneous bubbly flow regimes can be found in the literature [3, 4]. In summary, the following parameters have a key influence on the pump performance in two-phase flow:

• Gas content

• Gas bubble dimensions and distribution

• Pressure rise across the pump

• Suction pressure

• Rotational speed

• Impeller size

• Nature of the thermodynamic transformation of the two-phase flow media (isothermal, adiabatic, other)

• Geometry of the impeller, both the meridional shape (centrifugal, mixed, axial flow) and the blading

• Operating capacity as a fraction of the bep flow

Behavior at High Gas Content. Influence of Speed of Sound

Flow visualization of open centrifugal impellers, which are described by Murakami and Minemura [3] show that, as the gas content is increased, the gas bubbles tend to concentrate at the impeller eye on the blade suction surface (at part capacities) or the blade pressure surface (at high capacities). This behavior of the gas phase can be related to high local pressure gradients, which result from both the incidence angle (positive below the best efficiency capacity and negative at high flowrates above the best efficiency point) of the relative flow and the local blockage of the bubbles. With gas content of about 10 percent, the bubbles tend to agglomerate in a large gas pocket that clogs the blade inlet throat and causes the loss of pumping action. Therefore, the pump choking in this case is related to physical blockage of the inlet throat by a gas-filled cavity. Gas phase separation is the critical phenomenon. When the gas content is higher than 5.0 percent, the effect of the compressibility of the gas/liquid medium on the internal flow field pattern cannot be neglected.

A very peculiar property of two-phase flow media is the large influence of the gas content on the speed of sound as it is shown by theoretical analysis [7], and experimental data [9]. It can be seen [4, 7, 8] that the wave velocity in air/water mixture drops far below its value for either the liquid (1000 m/s or 3280 ft/s at 20ºC) or gas (340 m/s or 1115 ft/s at standard pressure and temperature conditions) phase and reaches values of about 25 m/s (82 ft/s) at $\beta s = 10$ percent. Normal values of peak relative velocities around the blade leading edge are in this range, especially at off-design conditions (high positive/negative incidence). Therefore, pumps operate at transonic or supersonic local flow. It is not surprising that a blade designed for incompressible single-phase flow is not very effective and produces choking for relatively high gas volume fraction (steamy) liquids.

Theory shows that the dramatic variation of the speed of sound at low percentages is very much related to the large difference in density between the two phases. The sonic velocity in two-phase mixtures is also pressure dependent. Therefore, one would expect that head performance of pumps handling two-phase mixtures depends on the suction pressure. According to calculations for isothermal homogeneous bubbly flow [4], a pump operating at a suction pressure of one bar and eye speed of 20-25 m/s (65-82 ft/s) would be close to or at choking conditions for air content around 10 percent. In fact, the peak velocity on the blade can be 20 percent or more higher than the inlet tip speed, depending on the flow incidence and blade geometry and thickness, where the local blade surface pressure is lower than the suction pressure.

Therefore, the fluid dynamical physics related to a very low speed of sound cannot be ignored in pumps operating with medium to high gas content, i.e., 10 percent or above, depending on the suction pressure. The effect would mainly manifest itself at off design conditions with low suction pressure/high speed and small blade throat areas (i.e., small stagger angle and thick blades), causing

• "Sonic" blockage of the throat (choking).

• Narrow range of operating conditions, depending on the blade geometry.

• Additional shock loss, which also depends on the blade geometry and increases with the inlet relative velocity.

DESIGN CRITERIA FOR TWO-PHASE FLOW PUMPS AND APPLICATIONS

Design Objectives and Approach

For gas content β_s below 10 percent (approximately) standard design pumps can be used if appropriately modified [2, 4]. Let us now consider pump applications characterized by gas content above 10 percent. Two-phase flow performance maps (head and efficiency *vs* capacity at different rotational speeds and suction pressures) are mainly characterized by four characteristic lines: 1) single-phase performance, 2) "critical gas content" producing full loss of priming, 3) the surge line, and 4) the choking line [2]. The surge capacity and the choking capacity define the operating range.

Essentially, the design objectives for two-phase pumps can be stated as follows:

• Move the loss of priming line to high gas volume fraction and/or minimize the head and efficiency degradation with moderate gas content.

• Control the surge line and choking line in such a way that either a broad range is achievable for flexible duties or a high efficiency is obtained for narrow range operation.

• The main mechanisms that seem to control the surge and choking phenomena are

  • separation of the gas phase from the liquid phase and tendency to coalescence in a large gas pocket at the blade entry throat, and

  • sonic choking due to the reduction of speed of sound.

The various pressure fields which operate inside the impeller blades play a critical role in the above two mechanisms [2]. Basically, the pressure fields are generated by a) centrifugal and Coriolis forces, b) aerodynamic or blade forces and c) inertial forces that are associated with the acceleration/deceleration of the fluid particles in the stream direction.

Flow visualization shows that centrifugal pumps (specific speed up to 3500 US approx) are not suitable for pumping high gas volume fraction media because of mechanism (a) (centrifugal and Coriolis forces dominant).

In order to generate blade forces that are gradually applied to liquid and gas particles, high solidity blade rows, i.e., with long chord compared to the pitch, should be selected. Care has to be given also to the blade shape and thickness.

The streamwise diffusion rate or fluid deceleration can be correctly controlled by appropriate shape of the impeller hub and shroud contours. It can be expected that open impellers work better than shrouded ones, as the differential pressure across the blade tip section, which is the most critical one, is reduced by the leakage flow.

The pressure rise through the impeller is partially controlled by the pressure field in the volute (vaned diffuser), especially at off-design operation. There are no data available showing the effect of volute or vaned diffuser geometry on two-phase performance. But, one would expect that special designs of volutes or vaned diffusers with low diffusion rates are more beneficial for broadening the operating range than high pressure recovery configurations.

In order to move the sonic choking line to higher capacity, the relative velocity at impeller eye should be reduced. Therefore, stationary (guide vanes) or rotating (inducers or other similar rotors) devices are instrumental in improving two-phase performance, when they generate at impeller eye a positive prerotation, i.e., a tangential component of the absolute velocity in the same direction of the peripheral speed. They also permit reduction of the relative velocity without reducing the rotational speed and so exploit its positive effect on the bubble average size.

Opening up of the blade inlet throat area is quite effective in delaying the gas locking effect and sonic choking as well. This can be achieved by increasing the blade stagger angle. Therefore axial-flow impellers (high specific speed, Ns) are superior to mixed-flow impellers (medium Ns) and radial-flow ones (low Ns) [2, 4].

Aerodynamic pressure gradients are too high for two-phase flow performance optimization in conventionally designed blades of mixed- and axial-flow rotors. Thus, specially designed impellers and casings must be developed.

Wide operating range and high efficiency are difficult to match, especially for two-phase flow pumps. Design strategy must rank priority between range and efficiency. On the other hand, the designer is reminded that the suction pressure PS has a large impact on the gas/liquid performance, so that this also becomes a design parameter. The selection of optimum design geometrical parameters is different for low (1.0 to 5.0 bar), medium (5.0 to 15 bar), or high (15 bar or above) suction pressure duty. From a fluid dynamic design standpoint, the duties can be categorized as follows:

- low $ß_s$ (10 to 20percent) and low-medium P_s (1.0 to 15 bar)
- high $ß_s$ (20 to 60percent) and low P_s (1.0 to 5.0 bar)
- high $ß_s$ (20 to 90percent or more) and high P_s (above 15 bar).

Low β_s and Low/Medium P_s Applications

Performance maps for conventionally designed pumps are shown in the literature [2, 3]. Some design considerations and application guidelines for open impellers are given in [9].

A special two-phase flow pump was designed by the author in 1981. This pump, which was developed for external recirculation from the reactor to the heat exchanger of a process plant operating on polyphase media, i.e., gas-liquid-suspended particles, is presented in the literature [2]. Limit values for the process variables were as follows:

temperature	=	150ºC (302ºF)
suction pressure	=	12 ata (180 psia)
gas content	=	15 percent
suspensions (organic and inorganic)	=	20 percent (by weight)
liquid velocity	=	min 5.0 to 6.0 m/s (16 to 20 ft/s)
		max 20 to 25 m/s (65 to 82 ft/s)
mixture corrosion degree	=	high

The pump can handle fluids with either inorganic suspensions (chemical plants for inorganic products, e.g., phtalic acid) or organic suspensions (biological and/or pharmaceutical plants, e.g., micro-organisms, bacteria) in sterile/not-sterile versions. Materials should be properly selected, depending on the application.

The pump has inlet guide vanes, an open mixed-flow impeller of special design, a vaned diffuser, and a fabricated volute. The leading design concepts for the guide vanes, impeller, vaned diffuser and volute and their matching were consistent with the principles enumerated above and are described by Schiavello and DeMichele [10].

Shop tests, which were carried out at low suction pressure using air and water, basically confirmed that the pump was not sensitive to the gas content up to approximately 10 percent. The pumps (four units) were installed in the field (May 1982) and started to operate at a suction pressure of 12 to 15 bar with a gas content of 15 percent to 20 percent. No head and efficiency degradation were noted between pure water and multiphase operation.

High β_s and Low P_s Applications

In the biochemical and pharmaceutical industries there are many processes that require a more or less intense aeration of the process medium and stringent regulation of the process temperature in a narrow range of a few degrees. To meet such needs, a cooling circulation pump is required, which must handle highly aerated media ($ß_s$ from 20 percent to 80 percent). The pump suction pressure is close that in the fermenter (reactor) and in many cases, is in the range of 1.0 to 5.0 bar. Typical fields of application are:

- Single cell protein production from various raw materials, e.g., hydrocarbons (such as n-paraffins, methanol, and ethanol), and carbohydrates such as molasses, sulfite liquor, whey, and hydrolyzed substances).
- Microbiological fermentation for the production of antibiotics, enzymes and organic acids, drinking and process water treatments.
- Biological waste water treatments.

The duty capacity can range from 100 to 10,000 m^3/h (500 to 50,000 gpm, approx), while the head range is much narrower, typically 30 to 50 m (100 to 170 ft). Centrifugal and mixed-flow pumps are very suitable for such duties. The operating range of the pump is narrow.

In many cases, the process medium is a polyphase flow which looks like an homogeneous foam, even at large gas content, the gas bubbles being finely distributed. The suspended particles are a small percentage in terms of volume.

Currently, there is no single stage pump that is able to circulate such high gas volume fraction media at the available relatively low suction pressure. A special impeller design, which operates with a partial extraction of the gas at the impeller hub and subsequent reinjection of the gas in the fermenter, was applied in conjunction with an inducer upstream of the main impeller in a volute pump family. Then a complete family of pumps with good two-phase performance was developed and several pumps have been installed since the mid 70s. These pumps are able to handle polyphase media with gas content up to 60 percent and even 70 percent.

Experience and experimental research (Ingersoll-Dresser Pump, Italy, and Vniighidromash, Russia) has indicated that for gas content above 20 percent (approx.) a partial removal of the gas at the impeller inlet along with the use of an inducer is needed in order to maintain high the performance with two-phase media.

High β_s and High P_s Applications

A very particular two-phase pump application for medium-high gas content (from 20 percent to 80 percent and even more) and high suction pressure (usually from 100 to 200 bar) is for direct pumping of gas/oil mixtures from oil well mouths either onshore or offshore. Downhole pumps capable of operating efficiently and reliably would permit the elimination of the current solutions that require antieconomic gas separation equipment and that are faced with serious safety problems, especially in offshore installations.

The relative limits of two lift systems, i.e., a) gas lift (GL) and b) submersible electric pumping (SEP) for offshore high volume production were examined by Cline and Garford [11]. One of the conclusions is that SEP is a method worthy of serious consideration for high volume producing wells in the remote offshore environment, provided that some current operational limitations are eliminated.

One serious limitation of a submersible, centrifugal pump is its inability to pump highly gassy oils. At a certain critical gas content the pump is locked and stops pumping. This causes the motor to unload and shut down due to excessive underload (control protection).

An experimental investigation of submersible multistage pumps with two different gas-liquid mixtures, i.e., air-water and carbon dioxide-diesel fuel, is described by Lea [12], indicating that the conditions of service at oil well mouths would not be prohibitive for a two-phase pump of a specific advanced design.

Several design alternatives were suggested in [2], whic included:

- Special geometry for the first stage (or more than one).
- Special inducer.
- Multistage geometry with mixed - or axial flow type stage.
- Variable speed.

A new multistage pump, using a helico-axial type design, optimized for two-phase flow through experimental investigation, is presented in Section III [13]. Another pump type that appears to be suitable for this type of multiphase pump application is the rotary screw pump described in Section IV.

Apart from the performance problem, other crucial design issues must be addressed for such pumps, especially from reliability and maintenance standpoints. Proper materials have to be selected by taking into consideration the presence in the fluid of a) sand (a few ppm), which can cause abrasion (mainly) and clogging (less critical), these are possible problems to solve in advance; b) sulfuric acid, which has very high chemical attack; and (c) water, as it can combine with hydrocarbons and produce hydrates, which in a cold sea, tend to freeze and clog the pump.

CONCLUSIONS FOR SECTION II

There are many applications in which pumps are required to handle a mixture of liquid and gas (condensable or noncondensable) and sometimes suspended solid particles. Conventional pump design with some particular modifications can be used for gas content up to 5.0 percent (low suction pressure) and even 10 percent (moderate suction pressure).

Special pump designs are requested for gas content from 10 percent to 20 percent and low to medium suction pressure.

A special pump design with an inducer and partial extraction of the gas is suitable for fermentation reactors operating with gas content from 20 percent up to 60 percent or more.

Finally, for pumping multiphase flow with gas content variable from zero percent to 100 percent in the oil production industry (onshore, offshore, subsea pump installation) two types of special pumps have emerged from the laboratory and field investigation conducted in the period 1985 to 1995, namely the rotodynamic helico-axial pump and the rotary screw pump. These two special pumps are discussed in detail in the two following sections of this tutorial.

SECTION III. HELICO-AXIAL MULTIPHASE PUMPS–PRODUCT DEVELOPMENT AND EXPERIENCE

Charles de Marolles graduated (1979) from the Federal Institute of Technology in Zurich, Switzerland, with a degree (Mechanical Engineering). In 1980, he joined the Engineering Department for Nuclear and Thermal Power Plants in Sulzer Pompes France. He worked on the design, manufacturing and installation of boiler feed pumps. In 1985, he was appointed head of the Engineering Group for Energy, Water Transport and Oil ans Gas. In 1989, he transferred to the Sulzer Pumps headquarters in Switzerland, serving as Product Development Manager, with activities in the mechanical development, calculation codes and design of high power boiler feed pumps, injection pumps, large water transport pumps. Since 1992, he has been responsible for the multiphase pump mechanical development (onshore, offshore and subsea). In 1995 he relocated to France where he is Technical Coordinator for Sulzer Pumps Europe.

Jacques de Salis graduated (1984) from the Federal Institute of Technology in Zurich, Switzerland, with a degree (Mechanical Engineering) and where he subsequently worked as a research assistant. He also holds an MBA from the School of Business Leadership, University of South Africa. In 1984, he joined the Pump Division of Sulzer South Africa, where he became head of engineering and design. He was then given responsibility for projects, sales, and contracts in the mining and water segments in Southern Africa. In 1993, he relocated to Sulzer Pompes France, where he is currently Program Manager for the multiphase pump product.

ABSTRACT

Multiphase boosting of raw effluent over long distances is a possible solution for cost-effective developments of small or remote oil fields. This technology is of interest not only for deep water and marginal fields but also for onshore applications. A complete range of helico-axial rotodynamic multiphase pumps (eight frame sizes) has been developed covering total volumetric flowrates from about 22,000 bbl/d (150 m^3/h) up to about 220,000 bbl/d (1500 m^3/h) at pump suction conditions. The pump mechanical design is detailed, while the instrumentation and pump packages are briefly described. Operational experience gained on the Pécorade field (onshore France) operated by Elf Aquitaine is also presented in this section.

INTRODUCTION

Multiphase pumping (MPP) finds most of its applications in marginal oil fields onshore and offshore, either subsea or on the surface. The idea is to transport the unprocessed effluent (water, oil, and gas) from the wells to an existing terminal having excess separation capacity. Subsea applications are specific and mainly dedicated to deep offshore regions [14]. Surface applications that are typically characterized by a lack of well eruptivity do not allow a conventional production method by natural depletion. MPP can then be used to add energy to the effluent at the well head, provided the wells are eruptive or activated (e.g., using downhole pumps or gas lift) in case insufficient reservoir energy [15]. Generally, surface applications require low well head pressures, below 150 psia (10 bars abs).

The function of MPP is to tie in newly discovered satellite fields to an existing production center or to boost the effluent in an existing pipeline from an older declining field, often in hostile or remote environments. Basically, MPP has been developed to simplify both the field process and operations by means of a single rotating machine on a single pipeline allowing unmanned operation of the multiphase pump at the remote site.

A complete range of helico-axial multiphase pumps has been developed for such applications. The potential benefits are to increase oil production by reducing the well head pressure; to restore the production of low pressure wells or old declining fields, and to extend the economical access to small or medium satellite fields in a cost-effective manner [16].

BACKGROUND

Design Conditions

Helico-axial pumps are capable of handling the raw effluent, i.e., a mixture of oil, gas, formations water possibly containing H_2S, CO_2, and a quantity of sand. Unlike conventional (single-phase) centrifugal pumps that normally have well defined operating conditions, multiphase pumps generally have to cover a wide working envelope in order to suit changing field parameters, due to reservoir evolution. With time, well head pressure and oil flows tend to decline, while the volumetric gas flow and possibly the water cut will increase. Furthermore, helico-axial pumps are fully capable of coping with unsteady flow conditions (reservoir or pipeline induced), which may consist of a series of gas pockets and liquid slugs, resulting in wide fluctuations of the gas volume fraction GVF which can reach 100 percent gas at the pump inlet.

Field Parameters and Pump Selection

In order to check the technical feasibility of MPP for a particular application, the following data are required as a minimum:

- Required oil flowrate
- Standard Gas Oil Ratio (GOR at standard conditions)
- Water cut
- Pump suction pressure
- Required discharge pressure

The GVF at pump suction can be determined from the first four parameters. A numerical model (described hereunder) is used to calculate the multiphase compression of the Poseidon helico-axial hydraulics and to plot the pump performance (Figure 1) for the given suction conditions (GVF, pressure level).

The capabilities of the current Sulzer multiphase pump range are:

- Total volumetric flowrate at suction: 22,000 to 220,000 bbl/d.
- Suction pressure (absolute): 50 to 600 psia
- Gas volume fraction at suction: 50 percent to 92 percent (min. down to 0 percent, max. up to 0.95, dry-running in 100 percent gas)
- Variable speed range: up to 6800 rpm

The pump pressure rise in multiphase flow depends on many thermodynamic, geometrical and operational parameters that are best calculated by the numerical model. However, a simplified procedure (Figure 2) is proposed here. It enables one to determine the

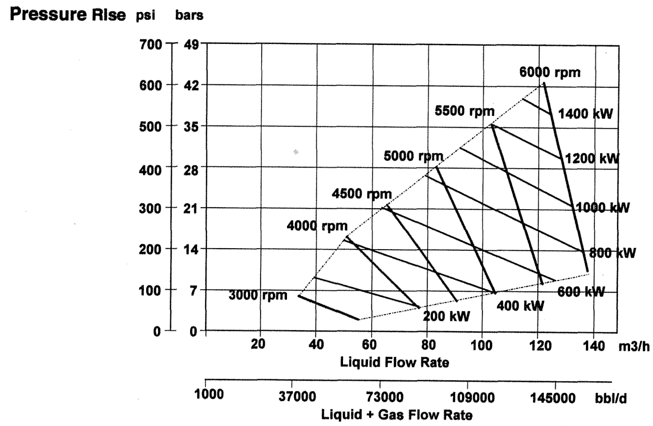


Figure 1. Typical Helicoaxial Pump Performance Curve for Given Suction Conditions.

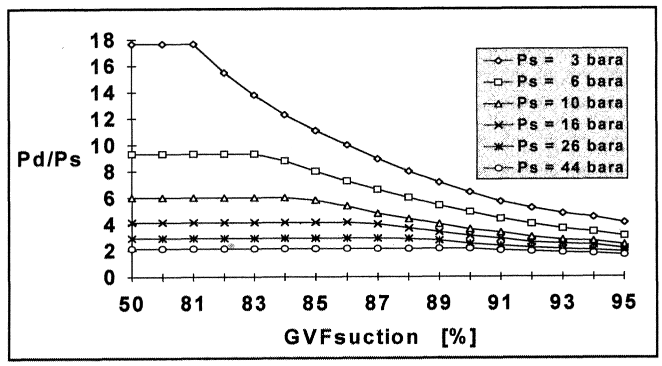


Figure 2. Achievable Pressure Ratio $P_{discharge}/P_{suction}$.

approximate maximum pressure ratios, which can be currently achieved within one single helico-axial pump, depending on suction conditions.

In order to finalize the equipment selection for a particular application, a pump optimization would eventually be carried out to determine the optimum geometry for each compression stage, the number of compression stages and the pump performance curves.

HYDRAULIC DESIGN AND PERFORMANCE

Hydraulic Design

A helico-axial pump is a rotodynamic turbomachine that is a hybrid between a centrifugal pump and an axial flow compressor. It is based on the latest second generation helico-axial hydraulics (Poseidon license) developed by the Poseidon group (Institut Français du Pétrole, Total & Statoil) [17, 18]. These hydraulics are designed to prevent the gas-liquid separation which would typically occur in conventional pumps designed for single-phase liquid operation.

The helico-axial multiphase pump is an in-line multistage barrel pump (Figure 3). Each stage or compression cell comprises a rotating helico-axial impeller and a stationary diffuser. The main advantage of the Poseidon hydraulics are

- Ability to handle any GVF ranging from zero (100 percent liquid) to 1.0 (100 percent gas) on a continuous basis.
- Mechanical simplicity and reliability (rotodynamic principle).
- Compactness.
- Very tolerant of solid particles (open type axial impeller).

Each compression stage (Figure 4) consists of an impeller, mounted on a single rotating shaft, followed by a fixed diffuser.

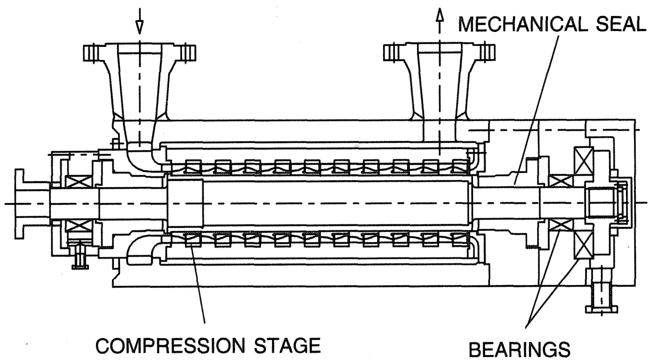


Figure 3. Pump Section.

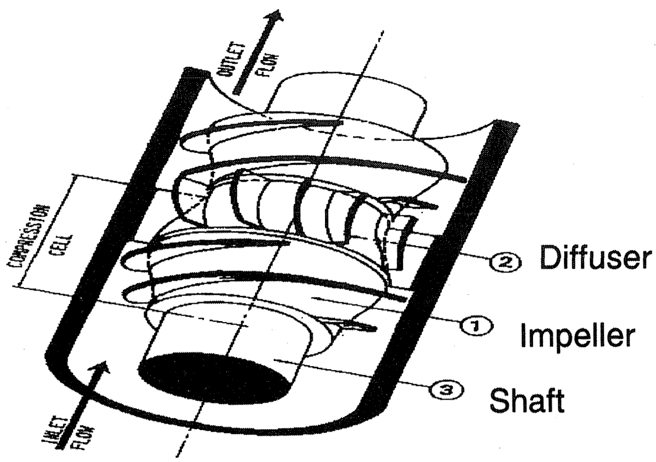


Figure 4. Compression Stage.

The impeller blades have a typical helical shape. The profile of the blades is specifically designed to prevent the separation of the gas-liquid mixture during the compression process. The hydraulic passages accommodate solid particles in suspension, and special care has been taken in the design to prevent their accumulation in the pump casing.

Due to the compression process, the gas volumetric flowrate decreases as it goes through the pump. The pump is therefore generally equipped with different series of hydraulics, i.e., different series of geometries, whereby one series would have identical compression stages. The changing geometry (internal diameters and blade profiles) from one series to the next provides adjustment for the decreasing volumetric flowrate.

An important feature of this design is the hydraulic flexibility which it affords and the wide range of duties that can be met by a single pump. This means that the multistage pump can be easily retrofitted to take account of changing reservoir characteristics during the production life of the field; in particular, the quantity of oil may be maintained as the GVF increases.

Two Phase Compression Computer Code

The computer program used to calculate the two-phase compression of helico-axial rotodynamic pumps is based on established correlations from IFP's test data bank. The program allows calculation of the two-phase compression, stage by stage, based on the theory of a compressible homogeneous fluid, corrected for two-phase performance. The model yields the results of multiphase pressure gain of each stage, summed for each different series, and finally summed for the entire pump. The total hydraulic power and the flowrate conditions at the pump discharge are also calculated. Multiphase pump performance curves (Figure 1) can be produced with the program. For given effluent properties, the following parameters can be plotted:

- For one given suction condition (pressure, GLR)–liquid flowrate and pressure rise for different shaft speeds.
- For one given suction pressure and speed, liquid flowrate and pressure rise for different GLRs at pump suction conditions.

The mechanical losses of the shaft seals and the bearings are calculated separately and then added to the calculated hydraulic power in order to define the power required at the pump coupling. The required motor power and pump speed will generally vary as a function of the evolving duty throughout the reservoir life.

MECHANICAL DESIGN

Design Data

The range is designed for onshore and offshore (platform) applications. To date, no multiphase pump design codes or standards have been defined. It is, therefore, suggested that the API 610 7th edition be followed as far as is practical. The pressure containing parts are calculated according to ASME Section VIII Pressure Vessels Code, Division 1.

The helico-axial multiphase pump range is designed for

- Design pressure: typically class 1500 psi (approx. 100 bar) to suit pipe rating.
- Design temperature: typically 200°F (approx 100°C) at discharge nozzle (including temperature rise due to gas compression).
- Speed range: 3000 to 6800 rpm.
- Power range: 500 to 3000 kW (at pump coupling).
- Severe fluctuations in torque associated with liquid or gas slugs.

Pump Design

Based on Sulzer's experience of injection barrel pumps and centrifugal compressors, a modular design for the multiphase pump range [16], consisting of eight pump frame sizes, has been developed. The chosen pump design is a multistage barrel in-line pump with axially split inner casing (Figure 3). The inline layout is required to facilitate the transport of the solid particles contained in the pumped fluid. The pump is attached to the skid at the barrel centerline to allow free thermal expansion. In the case of onshore installations, the pump would be mounted horizontally. In the case of limited available floor area (platform, offshore) or for subsea application, a vertical arrangement can also be used.

Due to the potentially large torque variations, the impellers are mounted with keyways on the shaft. The axial thrust is taken by a shaft shoulder. The diffusers are in two parts and located in an axially split inner case. The axially split design allows an easy inspection of the rotating parts. Further, the rotor does not have to be dismantled during the pump assembly. Therefore, a low residual rotor unbalance can be achieved. The impeller tip clearance is in accordance with API 610 7th edition requirements.

For ease of maintenance, the hydraulic parts (complete rotor with impellers, diffuser, inner case, and suction cover) are designed as a pull-out block. The suction and discharge nozzles are equipped with ANSI flange, class 600 lb, RTJ seal as standard, but other flange ratings are possible if required. Due to the stiff design of the baseplate and barrel, external nozzle forces and moments equal to four times the values given in API 610 can be met.

The suction is arranged radially with a short suction branch to prevent slug formation between the buffer tank and the first impeller. Due to potential explosive gas decompression, elastomer O-rings in "Viton A" or "GF" are used for the static seals. The shaft sealing is achieved with mechanical seals. In order to avoid gas or solid particles contaminating the seal chamber, a pressurized external fluid is injected (API 610 plan 53 or similar, synthetic oil as seal barrier fluid). Between the seal fluid and the suction/discharge nozzle, a pressure difference of 5.0 to 10 bar is continuously maintained - both in operation and on standby, by means of

- Pressure multipliers.
- Differential pressure regulators with seal fluid accumulators.

A seal fluid reservoir is located on the skid. Forced seal fluid circulation is used for mechanical seal cooling. A double mechanical seal is fitted a the drive end and nondrive end. This prevents ingress of the effluent into the seal fluid, which is, therefore, also isolated from the low pressure bearing lubrication oil.

The mechanical seals have identical sliding face diameters to limit the impact of suction pressure variations on the axial thrust. The axial thrust is always oriented towards the drive end; no axial thrust reversal will occur for the whole operating range. The axial thrust is taken up by a single hydrodynamic tilting pad thrust bearing. The lubrication is secured by oil injection, in order to minimize friction losses. The thrust bearing capacity is selected in order to fulfill the safety factor required in API 610 at the rated duty.

The bearing configuration is similar to the type used in compressors or injection pumps with two hydrodynamic tilting pad journal bearings normally being used. The tilting pad journal bearing offers good damping and remains stable even if unloaded. Normally, the journal bearings are oil lubricated. Alternatively, if produced or treated water is available with sufficient pressure, water lubricated ceramic bearings could also be used.

The pump rotodynamic behavior is a key issue for smooth mechanical operation. Several damped lateral vibration/critical speed analyses and damped unbalance response analyses, including coupling unbalance sensitivity, have been carried out to optimize the rotordynamic behavior of the pump.

The criteria in API 617 5th edition (Centrifugal Compressors for General Refinery Service) have been used to judge the lateral rotordynamic behavior of the pump. This code choice was dictated by the pump requirement to operate at high speed in 100 percent gas. These multiphase pumps are designed to run above the first, and below the second critical speed, with the required separation margins, guaranteeing a safe operating speed range free of any critical speeds.

Due to the high speed driver and the torque changes of the multiphase pump, a quill shaft (a thin shaft flexible coupling) is installed between the pump and the driver as in compressor trains. This quill shaft smoothes out the torque variations due to liquid slugs and makes it possible to tune the natural torsional critical speeds out of the operating range.

Special considerations are required for the selection of materials for multiphase pump parts for

- Corrosion: Multiphase fluids very often contain hydrogen sulfide with formation water and chloride, often at high temperature.
- Erosion: Multiphase fluids very often contain solid particles.

Therefore, in order to cater for aggressive effluents of this nature, all wetted parts would usually be in duplex stainless steel or in some cases super duplex stainless steel; (pitting resistance equivalent number larger than or equal to 40.) A chromium oxide coating can also be applied on wear surfaces where clearances are lowest.

Instrumentation for Pump Monitoring

As with injection pump units, multiphase pumps are typically equipped with the following conventional instrumentation suitable for hazardous areas and outdoor operation:

- Pressure and temperature at suction/discharge branches
- Bearing bracket velocity sensors (tappings for shaft vibrations are provided)
- Speed measurement
- Bearing temperature
- Temperature/pressure monitoring of mechanical seal circuits

The seal, lubrication, and driver are also monitored. For unmanned operation, automatic monitoring of the multiphase pump unit via a PLC is available.

Package Design

Like the injection pump unit, the multiphase pump is mounted on a horizontal skid designed for outdoor operation (Figure 5). A three-point skid is used for offshore applications. The following items of equipment are located on the skid:

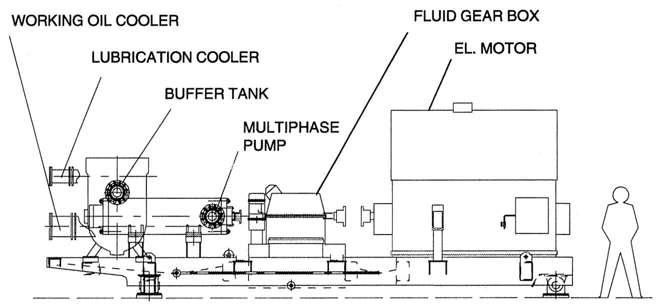


Figure 5. Horizontal Arrangement.

- Multiphase pump
- Buffer tank fixed near the pump suction nozzle (if required)
- Seal fluid system
- Lubrication system with oil coolers
- Variable speed driver
- Pumpset-mounted instrumentation

Various drivers can be used for the multiphase pump unit:

- Fixed speed electric motor and geared fluid coupling
- Variable low speed electric motor and gearbox
- Variable high speed electric motor (direct drive)
- Gas turbine
- Gas engine with gearbox and optional fluid coupling
- Diesel engine with gearbox and optional fluid coupling

EXPERIENCE: PUMP P302 FOR PECORADE FIELD

General

P302 is a normal 40,000 bbl/d (total volumetric flowrate at suction conditions) helico-axial multiphase pump based on the Poseidon hydraulics. It has water lubricated bearings and is driven by a 600 kW variable high speed electric motor. This pump is designed to suit the design point and working domain, (shown in Table 1). These specifications are based on the requirements for the Pécorade field conditions. The P302 pump features significant advances in terms of helico-axial MPP. The pump was designed for relatively low suction pressure (60 psig) and high compression ratios (up to 6).

The pump was delivered to Institut Français du Pétrole (IFP) at the end of 1993. It was then installed on the IFP multiphase loop at

Solaize (France) for bench testing. Pump tests were carried out under steady state and transient conditions. The pump was tested both under single phase conditions (liquid or gas) and with a multiphase mixture (fuel oil and nitrogen) at various GVFs and suction pressures. Theoretical predictions were compared with measured data. Experimental measurements agreed well with the theoretical prediction and the achieved pressure rises exceeded the design specifications.

Subsequently, the pump was commissioned in June 1994 on the Pécorade field (France, onshore) operated by Elf Aquitaine Production and has been running since then in a fully operational field environment (over 6300 hr as of the beginning of November 1995).

Pumpset Package

The P302 multiphase pump with its electric drive and lubrication unit is mounted horizontally on a common baseplate (Figure 6). Overall dimensions are 6.3 m long and 2.3 m wide. Total weight is 9.2 metric tons, baseplate included. The control panel and electrical utilities are installed at some distance away in a nonhazardous area.

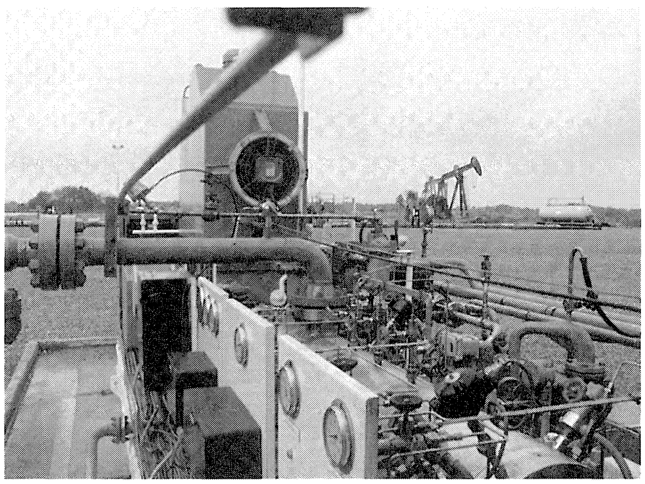


Figure 6. P302 Multiphase Pump Installed at Pécorade (France).

The pump rotor, as well as the diffusers, bearings, shaft seals, and end cover are designed as a cartridge assembly that is introduced into the casing to reduce maintenance downtime. Materials were selected for hot sour fluids (high H_2S content: 9.0 mole percent) which may contain solid particles. All wetted components are made of duplex stainless steel. A chromium oxide coating has been applied on the wear parts.

Bearings and the mechanical seal are designed to be lubricated by formation water supplied from an external source. Sea water could also be used, e.g., for platform applications. The lubrication system is pressurized above the pump internal pressure to prevent any ingress of effluent. An accumulator stores the pressurized water to lubricate the bearings during pump rundown in the event of a power failure.

The pump is driven by a 600 kW variable high speed electric motor. It is an asynchronous two pole 660 V motor. The speed can vary from zero to 6800 rpm. It is adjusted by changing the supply frequency between zero and 113 Hz. The electric motor is air-pressurized for use in hazardous areas, with protection according to Class IP 55 EExp II T3. Its bearings are forced lubricated cylindrical radial bearings. Frequency adjustments are obtained by a thyristor controlled inverter. The 750 kVA inverter is inserted between the constant voltage (660 V) and frequency supplied by the network and the asynchronous motor. The frequency inverter, transformer and control panel are not designed to operate in an explosive atmosphere and are located in a remote safe control room.

Field Experience

The Pécorade field is producing about 90,000 bbl/d (600 m^3/d) of fluid (oil and water) and 2,475,000 scf/d (70,000 Sm^3/d) of gas with 12 mole percent of H_AS. The effluent temperature is about 140°F (60°C). Oil and gas are transported separately to the Lacq plant through tow 6.0 in pipes over a distance of 35 km. A manifold built for the purpose at Pécorade allows operation of the pump upstream and downstream of the test separator. The piping is designed for both hydraulic performance testing and field operation.

First, hydraulic performance tests were conducted. The test separator was then connected upstream of the pump. The configuration allowed a controlled and steady state multiphase flow to the pump. This phase was completed at the end of November 1995. It showed excellent hydraulic performance exceeding the specification (Table 1). For given suction conditions (pressure, GVF, flowrate), the calculated and measured pump speeds were compared for the same pressure rise achieved during the tests. Measured speeds are generally between 5.0 and 10 percent lower than the calculated ones for GVF up to about 0.95. For a given rotation speed, the measured multiphase pressure rises are above the calculated ones. In summary, the P302 pump features excellent pressure rise capabilities in multiphase flow, even at low suction pressure and at high GVFs.

Table 1. P302 Design and Operating Specifications.

	Units	DESIGN POINT	WORKING DOMAIN	OPERATING DOMAIN
Suction Pressure	psig	109	72 to 218	43 to 246
Discharge Pressure	psig	362	100 to 493	261 to 653
GVF at suction	%	86	66 to 91	50 to 100
Total flow at suction	bbl/d	26,000 to 40,000	15,000 to 55,000	6,000 to 53,000
Speed	rpm	4,600 to 5,500	3,000 to 6,600	1,800 to 6,600
Hydraulic power	kW	230 to 300	100 to 500	100 to 500

In July 1995, the pump had accumulated about 4000 hr, with an average availability of approximately 90 percent during the operating periods. The pump has been dismantled and inspected twice and showed no signs of wear, erosion, or corrosion. Problems were limited to a partial removal of a protective coating (without damage to the underlying duplex stainless steel) and a slight displacement of an impeller on the rotor shaft during long operation in surge / severe slugging conditions. Currently, endurance testing is in progress. The pump is connected to the well manifold. This is the configuration under actual operating conditions, since the pump has to cope with natural flows from the wells.

The multiphase effluent from the low pressure wells is boosted by the pump so that it can be delivered to the first stage (high pressure) separator pressure of 275 psig (19 barg). In doing so, the produced gas can be recovered instead of being flared. Valuable experience has been gained, and the pump appeared easy to operate by the normal field operating staff without extensive knowledge of MPP.

CONCLUSIONS FOR SECTION III

MPP is essentially a means of adding energy to the unprocessed effluent which enables the liquid/gas mixture to be transported over long distances in a single pipeline without the need for prior separation. This has been made possible by recent major advances in understanding and predicting multiphase flow phenomena in pipelines as well as in developing a fully standardized multiphase pump range incorporating Poseidon helico-axial hydraulics. Interest in multiphase production that leads to simpler and smaller installations is primarily influenced by the appearance of more cost-effective production systems such as this one, which enables the exploitation of small, economically marginal or declining fields both onshore and offshore, often in hostile or remote environments.

ACKNOWLEDGMENTS

The authors are indebted to IFP and Elf Aquitaine Production for the P302 test results and field experience.

SECTION IV. ROTARY-SCREW MULTIPHASE PUMPS

Allan J. Prang is Manager of Engineering and Quality Assurance for Ingersoll-Dresser Pump Canada Inc. During his 29 years with the company, he has been involved with the research and development of various centrifugal-and rotary-pump designs for special applications. Recent product developments include large centrifugal slurry pumps, and rotary pumps for multiphase systems. He received his education in mechanical engineering at the Hamilton Institute of Technology and the University of Waterloo.

INTRODUCTION

Rotary two screw pumps are ideally suited for handling multiphase products in hydrocarbon production. Normally, two screw pumps are used for pumping liquids which can range in viscosity from water like consistency to highly viscous polymers with viscosities of millions of centipoise. However, screw pumps are positive displacement machines that have the ability to pump any product which can be introduced into the suction passages of the screws. This ability allows screw pumps to handle multiphase products that can range from 100 percent liquid to 100 percent gas and any combination between these extremes. With proper material selections for the wearing components, a significant amount of solids such as sand or contaminants can also be passed safely, provided particle sizes and concentrations are not excessive.

SCREW PUMP DESIGN

Pumps have been used for many years to handle liquids and solids but the gas phase normally creates problems for standard designs. Traditionally, centrifugal pumps have been used for pumping low viscosity products, but these pumps tend to lose efficiency and often complete pumping ability even with relatively low concentrations of entrained gas. Modifications to centrifugal pumps have improved their gas handling abilities, but they are generally limited to void fractions less than 50 percent.

In contrast, positive displacement pumps have been used for many years to pump difficult products and can generally handle any type of fluid that can be introduced into the inlet. The two-screw type of rotary pump (Figure 7), is particularly suited to handling difficult liquids. These pumps are designed with intermeshing screws on parallel shafts operating inside close fitting bores. Generally, left and right hand screws are mounted on each shaft to pump liquid from both ends of the pump into the middle. This arrangement has the advantages of doubling the pump capacity and balancing the axial hydraulic thrust created by the discharge pressure generated. This design also ensures that the shaft seals are subjected to only the suction pressure instead of the full discharge pressure of the pump. The screws are kept in mesh by precision timing gears on each shaft, thus preventing screw contact. The use of externally lubricated bearings and timing gears allows nonlubricating products to be pumped.

With this type of pump, the screw pitches can be machined to meet specific system requirements, thus permitting direct connection of standard speed drives with reduced power consumption. The screw profiles and clearances can also be custom designed to suit specific application requirements. The integral screw and shaft design provides greater shaft strength and less shaft deflection for reduced wear and longer life.

This type of pump is also very adaptable to different casing or body configurations required for this special multiphase application. In high viscosity applications, hopper type construction is used to feed the product directly to the screws with minimum losses. In other applications, jacketed construction can be provided to facilitate temperature control of the pump and process liquid. Special split casings have also been provided on large screw pumps to facilitate installation and maintenance.

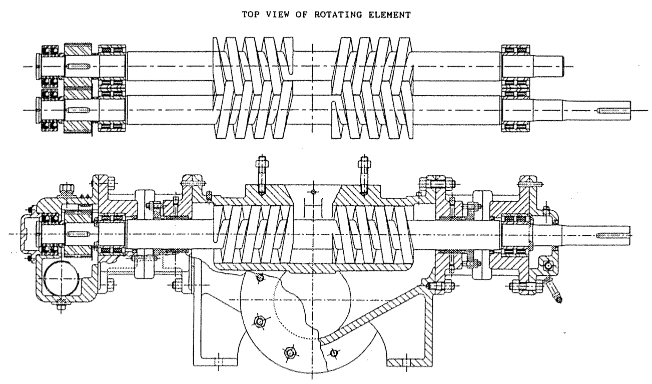


Figure 7. External Screw Pump.

PUMPING PRINCIPLE

As indicated, the two-screw type of rotary pump is a positive displacement type of pump and, therefore, can normally move any type of fluid product that can be introduced into the screw passages. Usually the fluids are incompressible liquids that flow through the screw passages from the suction area to the discharge area as the screws are turned in the figure eight shaped casing bores. This action produces trapped areas of fluid or locks. The number of screw turns on each of the meshing screws determines the number of locks available to trap the fluid. The finite clearances in the pump allow some slip of the product back into the suction area thus reducing the actual capacity from the theoretical displacement of the pump. The amount of slip is dependent on a number of factors including clearance, the number of locks, viscosity of the product and differential pressure.

The flow of the product through the casing bores is axial, thus providing a low shear type of pumping action and relatively low velocities. The variety of screw pitches that can be machined for a given screw size make this type of pump very flexible and allows custom design of the pump to suit exact capacity requirements. Many parameters such as pitch, screw length, clearances, screw profiles, shaft diameters, casing style, sealing methods and materials are optimized for each application to ensure the best pump fit providing maximum efficiency and reliability.

Normally with incompressible liquids, the pressure is developed gradually as the liquid moves through the screws from the suction areas to the discharge as shown in Figure 8. However, with compressible fluids such as the multiphase products, this pressure increase occurs mainly in the last lock adjacent to the discharge pressure area. If the screw pitch is small, there are many locks to trap the fluid and the cavities in the screws are relatively small. This design produces low slip and pressure pulsations, even with the compressible multiphase products.

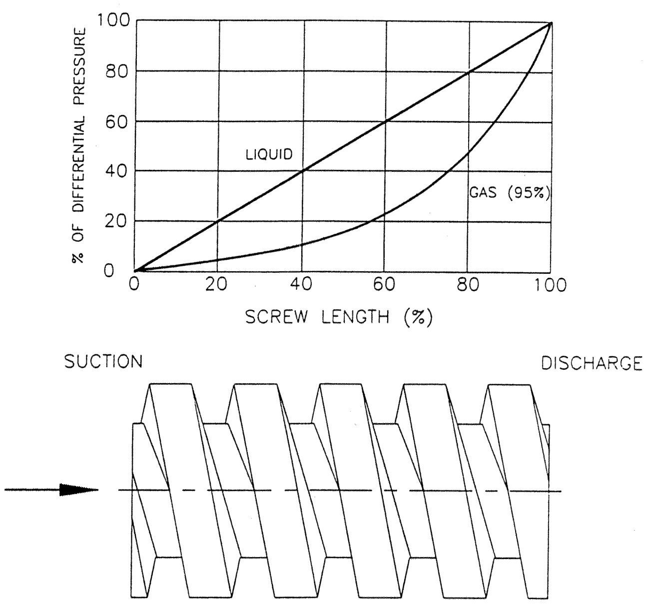


Figure 8. Pressure Profile in Screws.

MULTIPHASE PUMPING

When pumping multiphase products with high gas void fractions (GVF), the pump must be designed with a small pitch to provide the maximum number of locks. The key to pumping multiphase products is to ensure there is always some liquid available to seal the screw clearances and reduce the slip. Even a small amount of liquid is sufficient to provide this seal and allow the screw pump to operate with gas void fractions approaching 100 percent. Again, the performance is a function of the differential pressure, the clearances and the number of locks.

However, as the gas void fraction increases, the slip decreases until the capacity is equal to the pump displacement. This performance can be explained by examining the pressure drop of the multiphase product across a finite clearances. This theoretical analysis confirms that a small amount of liquid in the clearances will effectively seal these clearances and reduce the slip to near zero.

MULTIPHASE TESTING OF TWO-SCREW PUMPS

Although multiphase products have been pumped for many years with two screw type pumps, the performance on products with very high void fractions required special testing. A special test arrangement was developed to simulate operation on multiphase products.

For the initial tests, a standard LSH size screw pump was set up for operation on water and air (Figure 9). The suction piping was modified to allow the introduction of air into the pipe and also injection of small amounts of water. A standard oil field gas separator was mounted in the discharge piping to separate the air and water. This separator allowed tests of the "slugging" ability of the pump by rapidly switching from air to water operation.

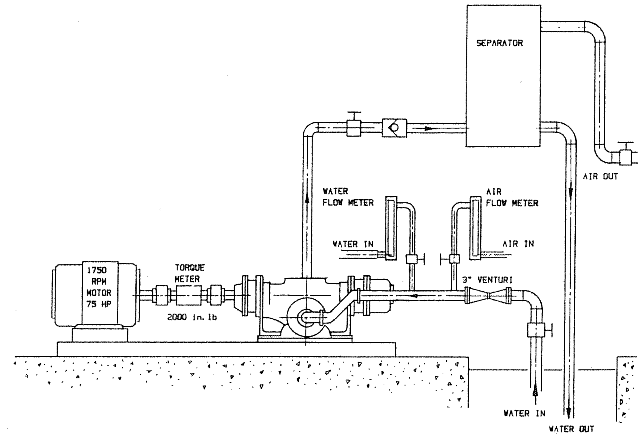


Figure 9. LSH Screw Pump Multiphase Test Arrangement.

The LSH screw pump is a standard design screw pump normally used for pumping lubricating or nonlubricating products. The screw diameter is approximately 5.31 in (135 mm) and in this case had a pitch of 1.0 in (25.4 mm). This pump is equipped with integral screws and shafts and mechanical seals to provide the maximum screw strength and minimum shaft deflection under load. The small screw pitch and the 9.5 in (241 mm) screw length provided 9.5 turns on each screw flight.

The pump was initially tested on water to establish a base performance level for comparison with the multiphase performance. In this test, the pump was operated in the normal mode with water being drawn from the suction pit and discharged through a pressure control valve back to the pit. The capacity was measured at various discharge pressures to obtain the characteristic performance on water. A torque meter was used to measure the power required at the various pressure settings and obtain efficiency values.

A series of tests was then conducted with increasing amounts of air being drawn into the inlet pipe. The air flowrate was measured with a variable area flow meter adjusted to the pump inlet pressure. At low void fractions, the water was drawn from the pit and measured by a venturi, as in the water only tests. At higher void fractions, the suction valve was closed and the water was injected into the suction pipe, with the flowrate measured by another variable area flow meter. This allowed testing at many different gas void fractions, as shown on Figure 10. The capacities shown are the totals of the water and air measurements adjusted to the inlet pressure at the pump.

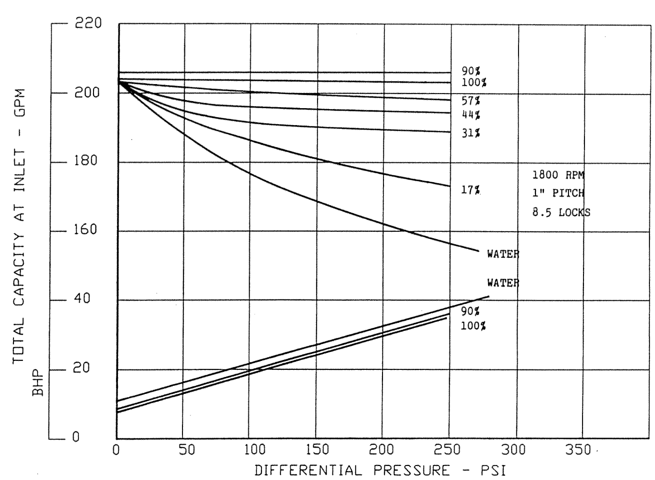


Figure 10. LSH Screw Pump Performance–Air and Water. Effect of inlet gas volume fraction (percent) for screws with 8.5 locks.

As the curve shows, the volumetric efficiency increased as the air content increased. The minor changes in capacity with air content above 90 percent are attributed to the accuracy of the inlet pressure measurement at the various conditions. At 100 percent air content, the water injected into the inlet was shut off completely. The pump operated smoothly at this condition but the casing temperature began to rise, since there was no water to remove the heat. However, even with no cooling or sealing water, the pump was able to operate for more than 10 minutes at 250 psi (17.2 bar). The pump was momentarily operated at 450 psi (31 bar), with no water, in order to demonstrate even higher pressure capabilities.

The improvement in volumetric efficiency with increased air content is explained by the way in which the pressure is generated with compressible fluids. As shown in Figure 8, the pressure increase occurs mainly in the last lock with compressible fluids as compared to the gradual pressure rise with incompressible fluids. The slip back past the last lock causes compression of the product in the screws, but if there are sufficient locks, this slip does not extend back to the suction. Therefore, the volume of fluid drawn into the suction area remains relatively constant regardless of the discharge pressure.

The horsepower curves show that there is little difference in the power required for water or the various concentrations of air. This means that the efficiency drops significantly as the air content is increased. Modifications could be made to improve this efficiency by using a variable volume instead of the constant volume design but this would then create problems when handling incompressible fluids unless other special design features are used.

The separator was then used to demonstrate the "slugging" capabilities of this pump. The pump was set at 250 psi (17.2 bar) when operating on 100 percent air and the suction valve was then opened quickly while the air inlet valve was closed. This switched the pump from 100 percent air operation to 100 percent water operation in a matter of seconds, thus simulating a slug of liquid entering the pump when operating at a high gas void fraction. This rapid switch from air to water caused a slight change in the sound of the pump but the operation was smooth with no significant changes in the power. The pump could be switched back and forth from water to air with no mechanical or hydraulic problems confirming the ability to handle slugs of liquid.

Several other tests were conducted to determine the effect of various parameters on the performance of the pump. Varying speed tests showed that the air handling capabilities were significantly reduced at lower operating speeds. The maximum gas void fraction possible is shown in Figure 11 at various speeds at 300 psi (20.7 bar) discharge pressure. The maximum conditions were determined by gradually reducing the water volume at the inlet and, therefore, increasing the gas void fraction as more air was drawn in to take the place of the water. When these maximum conditions were reached, a dramatic decompression occurred when the compressed air in the discharge piping rushed back through the pump and into the suction piping and into the water pit. At speeds of 1800 rpm and above, this decompression did not occur within the pressure capabilities of this standard screw pump.

The length of each screw was then shortened by machining, resulting in fewer turns of the screw flight and, therefore, fewer "locks" to trap and seal the high pressure fluid. As shown in Figure 12, this reduction in screw length reduced the ability to handle the very high gas void fractions at discharge pressures above 170 psi (11.7 bar). Although the performance was slightly lower than with the longer screws, the pump was still able to handle gas void fractions of 95 percent at pressures above 200 psi (13.8 bar).

The pump was then rebuilt with new screws to obtain the 8.5 screw locks and was operated at 3600 rpm to determine the effect of the higher speed. These tests were conducted similarly to the lower speed tests with the pump performance measured at various gas void fractions. These tests confirmed the ability of the pump to operate at the higher speed and demonstrated that high gas void fractions can be pumped at 3600 rpm.

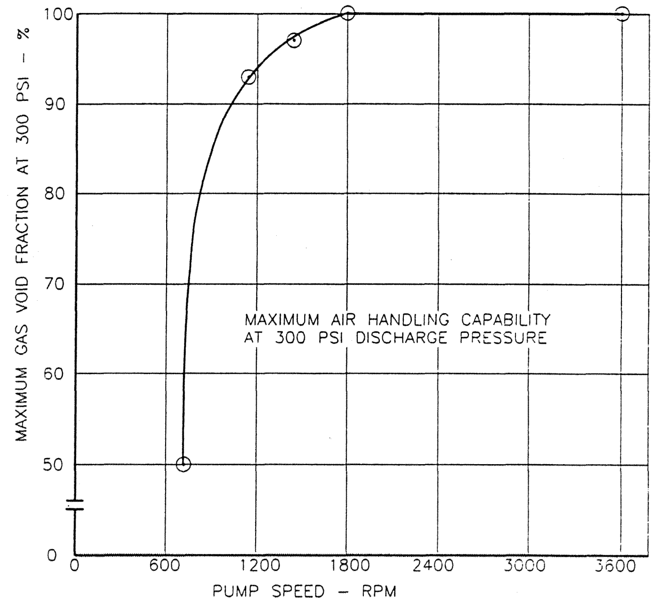


Figure 11. LSH Screw Pump Performance–Air and Water. Effect of pump operating speed on air handling capability.

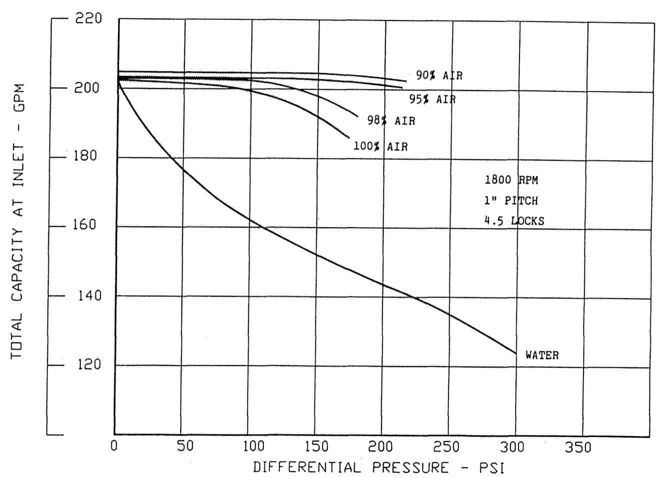


Figure 12. LSH Screw Pump Performance–Air and Water. Effect of inlet gas volume fraction (percent) for screws with 4.5 locks.

These tests confirmed the ability of this type of two-screw rotary pump to handle multiphase products with high gas void fractions. Special features will be incorporated into production multiphase pumps to optimize the pumping capabilities and reliability under field conditions.

An arrangement between IDP and Texaco, enabled additional limited testing to be conducted on this pump, utilizing the facilities and procedures outlined in Section V. Theses tests confirmed the above multiphase performance characteristics that had been obtained earlier in the manufacturers facilities.

SPECIAL MULTIPHASE BODY DESIGNS

For multiphase products with gas void fractions up to approximately 95 percent, there is normally sufficient liquid in the product to provide sealing in the screw clearances provided there are enough locks and the differential pressure is not excessive. The amount of liquid in the product will also remove any of the heat generated by the gas compression. However, when the gas void fraction approaches 100 percent or when significant slugs of gas are expected in the product, a special body must be used to ensure that there is sufficient liquid available for sealing and cooling. Several different designs are used to effectively ensure that the screws are always running with a slight amount of liquid present. This can be accomplished by setting the screws low in the body to trap liquid in the bores or by using a separating devise to separate the liquid from the product and recirculate it back into the pumping chamber. The separating concept has the advantage of being able to also provide a recirculating liquid flush for the mechanical seals as well as to facilitate cooling if required.

An example of this separating body concept is shown in Figure 13. This pump has a top suction with a suction chamber to feed the multiphase product to the ends of the screws. This ensures that the mechanical seals see only the suction pressure of the pump. The screws pump and compress the product to the center of the body providing axial hydraulic balance and exit from the side into a separating chamber. As the product moves through the separating chamber, the velocity decreases, allowing the liquid to separate from the gas and collect in a lower sump under the screws. The discharge pressure forces the separated liquid back into the pumping chamber through small recirculation lines back through the mechanical seals. This also provides lubrication for the mechanical seal faces.

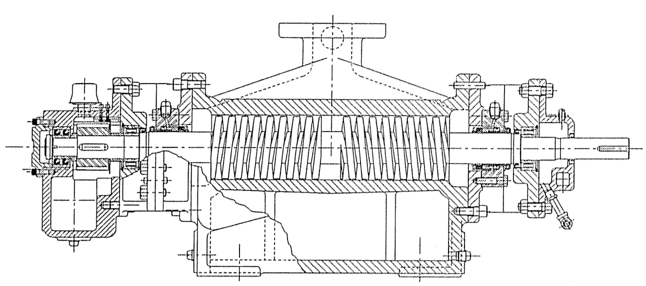


Figure 13. Multiphase Screw Pump.

The next pump to be tested is a true multiphase pump with the special separating body design shown in Figure 13. The first size developed is the MP1-125, which has the following performance characteristics and features:

Inlet Volume	6450 bpd	187 gpm
Differential Pressure	700 psi	50 bar
Product GVF depending on pressure	Up to 100 percent GVF for up to 2.0 hr	

Integral screws and rotors, Nominal pitch = 1.0 in, 7.5 locks

Normal "hard tipped" screws running in chrome plated bores

Gas separating body design

Pumps of this type will be placed in the field to confirm their ability to meet customer requirements.

SUB SEA APPLICATIONS

The newest potential area of application for multiphase pumps is sub sea. With special modifications, the rotary screw multiphase pumps can be coupled to submersible motors and mounted on the sea bed instead of on surface platforms. The idea of pumping multiphase products directly from a sub sea well head to shore facilities by means of submersible multiphase pumps has significant potential savings in separating equipment and platforms. A conceptual design is shown in Figure 14 of an MP1-300 size screw pump with special product lubricated outboard bearings and pressure compensated lure oil chambers.

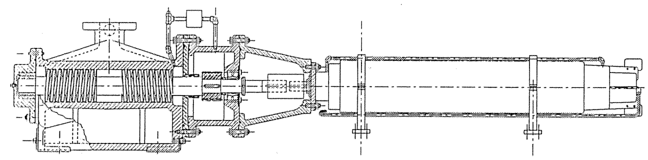


Figure 14. Sub-Sea Multiphase Screw Pump.

This pump design will be completed shortly and a prototype will be built and tested to prove the concept. The initial parameters of this MP1-300 prototype pump are indicated below:

Inlet Volume flowrate	60345 bpd	1750 gpm
Differential Pressure	500 psi	35 bar

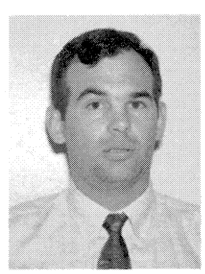
This pump will be built and placed in the field.

CONCLUSIONS FOR SECTION IV

The pumping of multiphase products requires special considerations in selection and design. Rotary screw pumps have proven to be capable of handling multiphase products with gas void fractions up to 100 percent along with handling slugs of 100 percent liquid. Two-screw pumps with timing gears appear to offer the best combination of properties and features required for these difficult applications.

While the market for multiphase pumps is growing, it is also changing to higher capacities and pressures. Larger pumps with higher pressure capabilities are being developed to meet these new market demands. Other modifications can also be made to these screw pumps to increase capacity and decrease power requirements resulting in increased efficiencies for these multiphase screw pumps. More development work is required to produce and prove reliable designs for sub sea multiphase rotary screw pumps.

SECTION V. MULTIPHASE PUMP TESTING EXPERIENCE IN LABORATORY AND FIELD



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INTRODUCTION

The deployment of multiphase pumps for hydrocarbon production is still considered an emerging technology to most operators. There base of historical experience relating to the application of multiphase pumps is small when compared to application of conventional liquid pumps. Testing is necessary to supplement the multiphase pump operator's limited experience and to reduce the risk of misapplication. This presentation describes the different levels of multiphase pump testing that have been performed by both manufacturers and major oil companies and also addresses the value and significance of the testing. From the viewpoint of the operator, the goals should be to determine the level of testing that presents the most value for their application and how the results from the testing can be used to improve their multiphase pumping applications.

LEVELS OF TESTING

For the purpose of this presentation, the multiphase pump testing is divided into four groups based on objectives, scope and cost. These groups consist of

- Post-assembly testing of the pump at the manufacturing facility.
- Precommissioning acceptance testing of the pump and driver system at a multiphase flow loop.
- Detailed pump performance testing.
- Long term endurance testing in the field.

All of these tests have been conducted by several different multiphase pump manufacturers along with the major oil companies that utilize these pumps. These tests were conducted with the objective of increasing the understanding of multiphase pump characteristics and also to minimize the risks associated with deployment.

MANUFACTURER TESTING

The various multiphase pump manufacturers have the capability to test their pumps at their own facilities. The objective of these tests is usually to verify the mechanical function and integrity of the pumps when operating at predetermined duty points. These tests can also be used to benchmark volumetric flowrates and the power consumption of the pump. This post-assembly testing of the pump can increase the end user's confidence in the pump by identifying infant mortality (burn-in) failures before the pump is deployed.

The testing can typically be performed quickly and at a relatively low cost. Problems that do surface during the testing can be quickly addressed due to the rapid access to the expertise of the manufacturer. Typically, the tests are conducted on single phase liquids, or possibly, air and water mixtures. The test matrix is usually limited in scope by time and cost. Additional limitations to the test matrix may be a result of either a power or a volumetric flowrate limitation of the test bed. The focus of the test is usually centered exclusively on the pump and not on the entire pumping system. The prime mover utilized for the test typically belongs to the test bed and is not the one scheduled for deployment with the pump. Safety and control logic are not normally tested as part of the pumping system.

PRECOMISSIONING LOOP TESTING

For applications where the end user needs a higher level of confidence in the initial success, of the multiphase pump deployment, the post-assembly testing can be replaced or supplemented by testing the entire pump and driver package in a multiphase flow loop. Testing the entire pump and driver package allows verification of the entire system prior to deployment. Both safety and control systems can be checked out without incurring the unforeseen cost of interrupted production in the field. By utilizing a dedicated multiphase flow loop, with live hydrocarbons as the test media, accurate flow capacity and power consumption information can be obtained for conditions that closely match the projected field. The test matrices for the precommissioning tests are typically limited to a handful of selected duty points to keep the cost of the test moderately low. The precommissioning test can also be used as a training opportunity to familiarize field operators with the characteristics of a multiphase pump system prior to the actual deployment. The use of a multiphase flow loop might entail some scheduling delays, depending on the current backlog of the multiphase facility.

DETAILED PERFORMANCE TESTING

Detailed performance testing of multiphase pumps is conducted for any combination of the following reasons:

- The multiphase pump is to be placed in an application that is outside the normally accepted operational envelope.
- The pump is a prototype or incorporates untested modifications.
- The operator wants to gain a better understanding of the pump characteristics over a broad range of conditions.
- The criticality of the pump reliability warrants positive verification of the pump performance.

The detailed performance test must evaluate the effect of several factors on the multiphase pump. The use of a multiphase flow loop is the only way to quickly and efficiently vary these factors to develop detailed performance curves under a broad range of conditions. Due to the multivariable nature of multiphase flow there is not a unique pump curve for a given shaft speed, rather there is a complete family of pump curves at each shaft speed that depend on the gas to liquid fraction of the suction conditions.

A list of some of the parameters that must be controlled during a detailed multiphase pump test are:

- The gas-to-liquid volume fraction at the pump suction (GVF).
- Pump shaft operating speed.
- Viscosity of the liquid phase.
- Pump behavior under transient suction conditions (slugging).
- The effect of GVF and differential pressure on discharge pressure.

Testing of the pump over a broad matrix of conditions allows an accurate mapping of the pump performance over a wide range of conditions. The use of live hydrocarbons in the multiphase flow loop to simulate actual flow conditions will ensure a high level of end user confidence for deployment of multiphase pumps in the field. Two of the drawbacks associated with detailed performance testing of multiphase pumps are the time and cost involved with performing the test. The procedures utilized for the detailed performance tests do not always mirror the operational procedures in the field; consequently, maintenance and operational considerations are not completely addressed by the testing.

LONG TERM FIELD ENDURANCE TESTING

Field endurance testing of multiphase pumps has been carried out by several of the major oil companies to monitor long term pump performance. These tests have also provided a basis for estimating the maintenance requirements and life cycle cost history for multiphase pump deployment. Additionally, the tests have enabled the operators to quantify the effect of multiphase pressure boosting on production. Testing pumps in the field provides a method of capturing operational experience and provides valuable design feedback to the manufacturer.

The critical requirement of a multiphase field trial is the effort and "buy in" of the field operators. The importance of the daily end users routinely and systematically recording pump data and their understanding of how this extra work will benefit them cannot be overstressed. One other consideration for field testing is the limited number of test conditions that the pump can be subjected to, due to local production constraints.

The following key parameters should be potentially monitored during a multiphase pump field trial:

- Volumetric flowrates of the individual phases
- Pump inlet pressure
- Pump differential pressure
- Pump shaft speed
- Pump vibration monitoring
- Slug frequency and duration at pump suction
- Temperature at pump suction and discharge

RECOMMENDATIONS FOR MULTIPHASE PUMP TESTING

For both the multiphase pump manufacturers and end users to maximize the value of multiphase pump application, they should

- Determine the level of testing that matches the application.
- Understand the value and significance of the test information.
- Utilize the test results to improve both current and future applications.

REFERENCES

1. Lovie, P., and Barker, K., "Multiphase Pumping–Where to now?" Offshore, pp. 105-110, 150, (May 1995).
2. Schiavello, B., "Two-Phase Flow Rotodynamic Pumps–Experiments and Design Criteria," Pumps Offshore, Course No. 7133, Norwegian Society of Chartered Engineers, Norway (1986).
3. Murakami, M., and Minemura, K., "Effects of Entrained Air on the Performance of Centrifugal and Axial Flow Pumps (1st Report, Performance and Flow Conditions)," Bulletin of the JSME, *17* (110), p. 1047 (1974).
4. Saccenti, P., "Theoretical and Experimental Investigation on the Performance of a Mixed-Flow Pump with Air/Water Mixture," B.S. Thesis (in Italian), University of Florence (1978).
5. Neumann, W., "Efficient Multiphase Pump Station for Onshore Application and Prospects for Offshore Application," *Proceedings of the Eighth International Pump Users Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 43-48 (1991).
6. Muench, R. A., "The Prediction of Pump Differential Pressure for Two Phase/Two Component Compressible Flow," ASME Paper 76-WA/HT-75 (1976).
7. McWilliams, D., and Duggins, R. K., "Speed of Sound in Bubbly Liquids," Proceedings of the Institution of Mechanical Engineers, 184, (3C) (1969-70).
8. Karplus, H. B., "The Velocity of Sound in a Liquid Containing Gas Bubbles," Armour Research Foundation Report 000-248 (IID-4500) (June 1958).
9. Cappellino, C. A., et al, "Design Considerations and Application Guidelines for Pumping Liquids with Entrained Gas Using Open Impeller Centrifugal Pumps," *Proceedings of the Ninth International Pump Users Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 51-60 (1992).
10. Schiavello, B., and De Michele, C., "20-PP6-23 Aerated Broth Pump," T.N. 92 and T.N. 98, Worthington S.p.A., Desio (Internal Report) (June 1982).
11. Cline, W. B., and Garford, D. W., "Artificial Lift Systems for Offshore Use," International Petroleum Times, p. 22 (Dec. 1, 1978).
12. Lea, J. F., "Effect of Gaseous Fluids on Submersible Pump Performance," Society of Petroleum Engineers AIME Paper 9218.
13. de Salis, J., et al, "Helico-Axial Multiphase Pump Technology for Enhanced Oilfield Production," BPMA 14th Technical Conference on "Using Pumps Better," Birmingham, United Kingdom (1995).
14. Deparis, F., Verdier, J., "Role of Departure Pressure, GOR, Water Depth in Multiphase Pumping," Offshore, pp. 65-72 (August 1994).
15. deSalis, J., Moins, G., Oudin, J.C., and Falcimaigne, J., "Helico-Axial Multiphase Pump-Product Development–Selection Tools and Applications," OSEA 94046, 10th Offshore South East Asia Conference, Singapore, p.16 (1994).
16. deSalis, J., de Marolles, Ch., and Schachenmann, A., "Multiphase Pumps for Mariginal Oil Fields," World Pumps, pp. 38-41 (July 1994).
17. Lafaille, A., "Poseidon: The Multiphase Boosting and Transportation System," OSEA 90128, 6th Offshore South East Asia Conference, Singapore, pp. 583-595 (December 1990).
18. Gie, P., Buvat, P., Bratu, C., and Durando, P., "Poseidon Multiphase Pump: Field Tests Results", OTC 7037, 24th Offshore Technology Conference, Houston, p. 20 (May 1992).