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PERFORMANCE EVALUATION OF TWIN-SCREW PUMPS OPERATING IN SERIAL ARRANGEMENT

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Abstract.
Multiphase pumping from wellheads to processing facilities has been a challenge during last decades, especially in remote areas such as deep waters. Twin-screw boosting technology has shown a versatile and reliable performance for a wide range of gas fractions inlet streams. However, the use of positive displacement rotating pump in field conditions is restricted to constant displacement pumps (i.e., no reduction of fluids volume), due to the variability of the gas fraction in the produced streams, which can eventually be zero. On the other side, when operating with high inlet gas volume fractions and high compression relations, the constant displacement condition leads to a very inefficient operation, mainly due to uneven distribution of the power delivered to the fluids stream along the screw, producing higher wear, uneven dilatation and eventual rotor/stator interference. Therefore, beyond the energy efficiency issues, this condition can significantly shorten the pump lifespan, making unfeasible their application in deep-water environment due to the high cost of maintenance. An alternative for dealing with high differential pressures and highly variable gas volume fractions is to divide the compression in two or more pumps, which displacements are controlled through the rotation speed, according to the gas fraction at inlet of the pumping system. In this work, we present an experimental investigation of the performance of a boosting system based on the serial arrangement of twin-screw pumps. Results show that the system performance is significantly increased in terms of both, volumetric and energetic efficiencies, when operating in serial arrangement. Some recommendations for the system design are presented based on the experience gained along with the investigation.

Keywords: *Boosting, Multiphase pump, Twin-screw pump, Serial arrangement*

1. INTRODUCTION

The pumping of liquid-gas mixtures, without prior separation of phases, is extremely attractive in any industrial process. In the case of offshore oil production, pumping the produced fluids, from sub-sea wellheads through long distances without prior phase separation, have enormous economic and technical advantages since, in addition to the elimination of separators, the pumping is carried out by a single machine, instead of a pump and a compressor, thereby reducing both CAPEX and OPEX. In particular, deep water oil fields applications for pumping from the seabed, where these aspects are certainly even more relevant, make multiphase pumping a strong technically and economically attractive option. The future horizon of production, in deep and ultra-deep waters, is pumping along the seabed without prior treatment of the fluids produced, reducing the number of platforms needed in a given field or even direct pumping for onshore treatment (the concepts "*subsea to shore*" or "*subsea to somewhere*"). In this context, pumping systems must be capable of robustly handling diverse operating conditions, mainly in terms of gas volume fractions and large pressure increments, Δp.

In applications with high gas fractions (typically above 70%), rotodynamic pumps, such as centrifugal or helicoaxial, become incapable of operating properly due to the well-known *gas-locking* phenomenon, mainly in cases of low absolute suction pressure. The gas locking tendency depends on phase densities relationship at suction. However, this relationship is function of suction pressure, which is desirable to keep as low as possible, to increment the production rates, mainly in seabed pumping. Therefore, for transfer/boosting applications, the positive displacement pumps presents as an important (sometimes the unique) alternative.

Among the positive displacement pumps, twin-screw pumps can operate at high speeds and, therefore, relatively high flow rates, in comparison with other pumps of this type, as Progressing Cavity Pumps, for example. The operation of multiphase twin-screw pumps with compression ratios ($p_{discharge}/p_{suction}$) of the order of 10 or more, with gas fractions of up to 90 %, is feasible, although the energy efficiency can be very low, in virtue of the internal fluid recirculation.

In constant displacement pumps (i.e., when there is no volume reduction of the pumped mixture throughout the process) at high pressure differentials and high GVFs, the pressure increase along the rotor becomes non-linear, leading to an uneven distribution of the power delivered to the stream along the screw. The work transfer to the fluid is concentrated in the last stages of the pump and a there is high internal recirculation of liquid, producing higher wear and uneven dilatation. These phenomena can significantly increase the pump wear in addition to promoting high local expansion and high flexion of the screw, which can also lead to and eventual rotor/stator interference (contact between rotor and stator) which, at high speeds, can lead to machine inoperability. In summary, beyond the energetic efficiency issues, the machine's lifespan can be seriously compromised in the operation with high Δp an high GVFs, which can make its use unfeasible in environments where maintenance costs are extremely high, such as in deep waters. On the other hand, variable displacement pumps would be unable to operate in situations with 100 % liquid (0% of gas), or when the compressibility of the mixture is very low. Thus, for robust field operation, it is necessary that the pumping system be able to operate with a condition of 100 % of liquid up to high gas fractions of (95+ %) in the suction stream.

One solution for this problem, which can significantly reduce the internal recirculation and improve the distribution of power transfer along the rotor, is to divide the compression of the multiphase mixture into steps, using pumps in serial arrangement. This serial arrangement divides the compression of the gas along two (or more) pumps, resulting in a more efficient operation for both pumps.

Several experimental studies of the performance analysis of twin-screw pumps operating in multiphase condition can be encountered in literature, but few of them were published in peer-reviewed journals. Vetter and Wincek (1993) presents one of the first studies of the performance of a twin-screw pump in multiphase operation. Vetter *et al.* (2000) presents an experimental study, with focus on abrasive wear. Egashira *et al.* (1998) developed an experimental study focused on backflow analysis. Rabiger *et al.* (2006) and Rabiger *et al.* (2008) developed some experiments with the focus on the development of a mechanistic model. More recent experimental studies were presented in Patil and Morrison (2017), Liu *et al.* (2017) and Liu *et al.* (2019), developed at Texas A&M University turbo-machinery group. To the knowledge of the authors, the only studies which address the multiphase pumps performance in serial operation were present in Vauth *et al.* (2004) and Rausch *et al.* (2005). However, these studies address some dynamic aspects of the process and no results of the system performance is preseted.

In this paper we present a systematic experimental study the performance of twin-screw pumps operating in two-phase conditions in a serial arrangement. The experimental set-up used provides detailed data of the operation, as pressure and temperature in all stages of the system, gas and liquid flow rates and rotation and torque of the pumps. First, the performance of a single pump is evaluated for different rotations, differential pressure and inlet gas volume fractions. Then, the operation of the pumping system, consisting of two identical twin-screw pumps in serial arrangement is assessed. The performance is evaluated through the volumetric and energetic efficiencies which can be independently assessed for each pump, operating in serial arrangement.

2. METHODOLOGY

2.1 Description of the experimental flow loop

A schematic view of the experimental flow loop is shown in Figure 1. The circuit consists of a compressed air line and a liquid flow line. The working fluid for the liquid phase is water. However, for this specific experiment, synthetic ISO VG 60 oil was added in order to preserve the internal pump components and some parts of the circuit, which are not stainless steel, from corrosion. The liquid stream operates in a closed loop, while the gas (air) works in an open loop.

Pipes and components in all lines before the test section are specified in ASME Class 150, i.e., admit a working pressure up to about 18 bar, considering that temperature remains below 100 ^{o}C. The components located between the discharge point of the test section and the control valves after the separator, are specified in ASME class 300, i.e., admitting a working pressure up to about 50 bar. However the nominal working pressure of the gas-liquid separator vessel is 30 bar. Therefore the working pressure of discharge line of the flow loop is limited to this value.

Pilot lines of compressed air are included to operate the pneumatic control valves and also for the pressurization of the mechanical sealings of the twin-screw pumps (these last not shown in the Fig. 1)

The air is compressed through a screw compressor with nominal flow-rate of $20m^3/h$ at 12 bar. Then it is treated for removing humidity and eventual solid particles and its pressure is reduced to the set suction pressure of the pumping system, through a pressure regulation valve. The liquid circulation is promoted by two Progressing Cavity Pumps, PCP1 and PCP2, with maximum nominal flow rates of 30 and 3 m^3/h, respectively. On of the main advantages of this type of pumps is that the flow is almost proportional to the rotation, for relatively low differential pressures across the pumps, which is the case of the experiments developed, as the PCP outlet pressure is equal to the suction pressure of the tested pumping system. The liquid flow rate is controlled by the PCP1 rotation, through a frequency inverter (CFP). The PCP2 is used in cases of very low liquid flow rates, once the operation of the electric motor turns unstable at very low rotations. However, in the experiments developed in this work, the PCP2 was not used, once, for the maximum values of GVF attained in experiments, the liquid flow rates were still high enough to be controlled with PCP1.

The liquid flow rate is measured through a Coriolis flow-meter model CONTECH TCM 28K, with maximum flow rate $28m^3/h$ and maximum uncertainty of 1 %. The gas flow rate is measured with a thermal flow meter, model TCM 28K, with maximum flow rate $105Sm^3/h$ and maximum uncertainty of 1%. Then the air and liquid streams are mixed and directed to the test section. Check valves are used in gas and liquid lines (VR1 and VR2), before mixer, in order to prevent the flow return, when any of the lines pressure is higher than the other. After passing the test section, the gas-liquid stream is directed to the gas-liquid separator, at the pressure conditions of the pump discharge. In typical pump tests, the discharge pressure is controlled through a throttle valve. However, in multiphase condition this operation can turn unstable, mainly in high GVF conditions. Therefore, in order to keep the operational stability of the test loop, the pressure reduction is done after the phase separation. The discharge pressure, which is considered equal to the separator pressure, is controlled by a pneumatic valve which aperture is set by a PID controller ($PID1$), having as input the error in the set separator pressure (p_5). The liquid level in the separator is controlled by a second pneumatic valve, actuated by a second PID controller ($PID2$) which has as input the difference between the set and measured liquid levels, which is measured with a sensor based on the differential pressure between two taps located at known heights (sensor LL). After separation, the air stream passes through an additional coalescent separator, for further separation of small entrained droplets, which is manually drained periodically, and then filtered and vented through a silencer. The liquid stream passes through a cooler, which is refrigerated with air at ambient temperature, in order to maintain constant its temperature along he experiments, and directed to the liquid reservoir.

Pressure and temperature are measured at the indicated points (P_i, T_i) in the loop scheme (Fig. 1. Temperature sensors are WARME model WTD-4402, whit a resolution of 0.1 ^oC and uncertainty of 0.25% full scale. Pressure sensors are WARME model WTPI-2500, whit a resolution of $p_{max}/2000$, where $p_{max}/2000$ is the maximum pressure measurable with the sensor, and uncertainty of 0.25% full scale. The respective maximum pressures for each sensor are: 10 bar for sensors P1 to P3 and P7, 15 bar for sensors P4 and P6, and 30 bar for sensor P5.

The output signals of pressure and temperature sensors are in voltage (1 to 5 V) and flow-rate sensor outputs their signals in current (4 -20 mA). All sensor signals are collected and transferred through screened cables to the acquisition system which consists of a National Instruments PXIe-1062 system with two PXI-6135 boards and one PXI 6115 board. The NI PXi system is connected to a computer which provides the interface for loop supervision and control. A LabView®based software was developed which allows a visual interface, data plotting and writing, and input the pumps rotation and setting point of PID controll parameters, as discharge (separator) pressure and intermediate (p_4) pressure between the pumps, for the serial operation.

2.2 Experimental procedure

The flow loop described in the previous section can be adapted for different applications involving gas-liquid flows. The experimental investigation of the step multiphase compression process, was developed using a test bench designed in our research group and constructed by NETZSCH do Brasil, consisting of two twin-screw pumps in serial arrangement. The test bench is shown schematically in Fig 1.

The test bench consists of two NETZSCH pump prototypes, connected in serial arrangement, with an intermediate vessel which act as a volume buffer. The bench includes return lines for both pumps, with security valves, set to open at a given maximum admissible pressure. In addition, a set of valves allow the bypassing of the first pump, in order to allow the performance evaluation of a single pump. These tests for standalone operation were performed on the downstream pump and, for the serial operation, it was assumed that pumps are identical, although the upstream pump was not independently tested. At the discharge of the downstream pump a check valve is included (VR3), to prevent flow return form the high pressure line.

The pump prototypes used in these experiments are originally designed for single phase (liquid) operation and are based on the model NETZSCH NOTOS 4NS with a special mechanical seal adapted for multiphase operation. These pumps are capable to operate with some gas fraction at inlet, but are not specifically designed for multiphase operation. NETZSCH do Brasil company, which is a partner in this research project, is currently developing a multiphase pump and a first model is being released in the market. However, due to availability at the time of experiments, the original pumps were used in the assembly used for the experiments. Therefore, it is important to make clear that these pumps do not represent the current state of the technology available for multiphase twin-screw pumps, in terms of differential pressure and GVF capacity. However, the main objective of this investigation is to develop a proof of concept of the step multiphase compression. Therefore, the results will be presented in terms of comparison of the performance of the serial arrangement system, versus the same condition in the same pump in standalone operation.

The pumps' rotation is controlled through frequency inverters (CF1 and CF2). From the supervisory system implemented in LabView, the pumps' rotation can be manually and independently controlled or controlled by the suction pressure of the pumps. In the experiments developed in this study, the suction pressure of the upstream pump was maintained constant and equal to 0.5 bar (manometric pressure), and the upstream pump rotation was manually set.

The mechanical torque of both pumps is measured through a HBM T22-500 transducer, which can measure torques

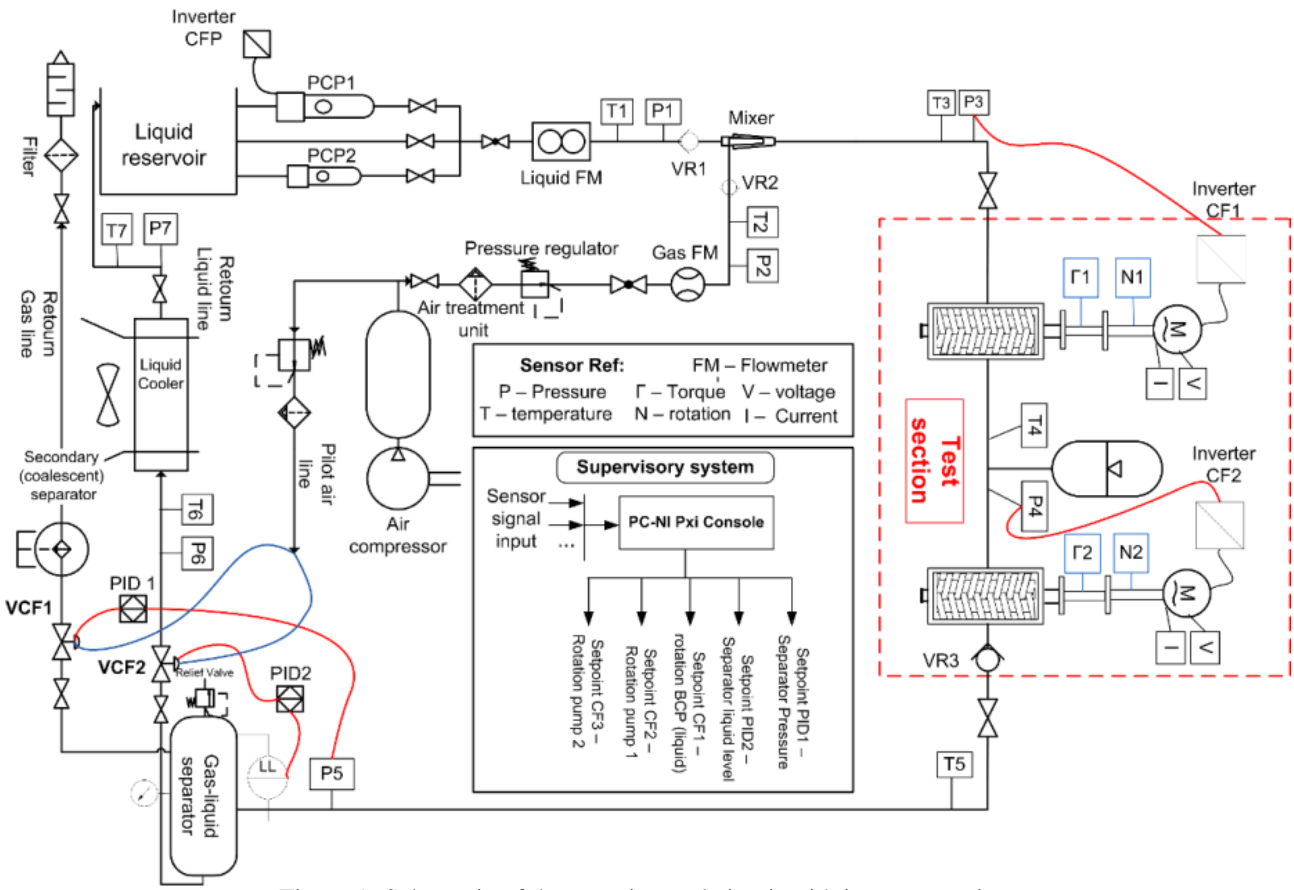


Figure 1: Schematic of the experimental circuit with instrumentation.

from 0 to 500 Nm with a 0.5 % full scale uncertainty. The pumps' rotation was measured, independently of the frequency inverters, through an encoder coupled to the motor shaft. However, the difference between the set (from frequency inverters) and measured rotations were below 1% in all test conditions.

The working fluid used in the experiments were compressed air and water-oil mixture as liquid phase. As commented in previous section a small amount of oil is added to the liquid phase (water) to protect the pumps loop components for corrosion. The liquid phase consist of a water-oil mixture with approximately 4% of oil, which represents the maximum solubility of this oil in water. The addition of oil does not promote significant changes on liquid phase properties, which remain very next to water properties. The liquid density, measured with the Coriolis flow-meter, was about $980kg/m^3$, and the viscosity remained between 1.2 to 1.8 centipoise at typical experimental conditions, where temperature varied between 20 and 45^oC.

Each experiment consists of the data acquisition of all measured parameters, including pressure and temperatures at all measurement taps, and gas and liquid flow rates, for a given Δp, which is equal to $p_5 - p_3$, suction pressure (p_3) and pump rotation. For each of these conditions the gas volume fraction at inlet is varied from zero (pure liquid) to the maximum gas volume fraction (GVF) admissible in each condition. As will be discussed in the results section, at a given GVF at the suction, gas begins to flow more intensively through peripheral gaps, and there is a loss of hydraulic sealing [1]. This condition is observed in experiments, as a steep reduction in the pumped flow rate, for small increase in the GVF at the suction. This maximum GVF depends on pump geometry and, mainly, on rotation speed, as the centrifugal effect trend to displace the liquid to the peripheral gaps.

The suction gas volume fraction (GVF) is calculated assuming homogeneous flow, as,

$$GVF_{suc} = \frac{\dot{V}_{G@SUC}}{\dot{V}_L + \dot{V}_{G@SUC}} \tag{1}$$

where $V_{G@SUC}$ is calculated from the gas flow rate measured with the flow meter $V_{G@FM}$, which outputs at standard pressure and temperature conditions, corrected by temperature and pressure at suction, p_3 and T_3, considering ideal gas.

For each experiment, a series of steps are necessary. The procedure for each experiment can be summarized in the following steps:

[1]this point is sometimes called "prime loss" in multiphase positive displacement pumps literature and should not be confused with the "gas locking" in rotodynamic pumps, which causes a similar performance loss, but is a completely different phenomenon

- The flow loop is pressurized with air up to about 8 bar (manometric pressure) in order to not compromise the pumps sealings (located at pumps inlets). Liquid line remains closed along the duration of this step.
- Then, the air line is closed and the downstream pump is turned-on, operating in single-phase (liquid) condition and the separator pressure is increased up to the selected discharge pressure, by the liquid injection. The discharge pressure is controlled by adjusting the liquid level in the separator. Once the separator pressure is controlled by the air discharge from valve VCF1, this automatic control cannot be operated in operation with pure liquid .
- The downstream pump rotation is gradually increased to the set value, simultaneously increasing the PCP rotation (trough CFP), in order to equalize the pumped flow rates from PCP1 and downstream twin-screw pumps, and the operation data for the condition of GVF=0 is acquired, for the given operating conditions.
- For the case of serial operation, after the steady operation of downstream pump is attained (previous step) with the upstream pump bypassed, the suction valve of upstream pump is opened, letting the liquid flow through the pump and the bypass simultaneously. Then the upstream pump is turned-on and its rotation adjusted to the same value of the second pump. Then the bypass valve is slowly closed (manually) directing the whole flow through the pump. During the operation with liquid, due to the incompressibility, any mass imbalance of pumped streams (between PCP and test bench inlet or, between upstream and downstream pumps) lead to a very quick pressure increase. Therefore, the cases with GVF=0 are those which bring more risk to the flow loop and pump integrity. Then, the actions in the flow loop must be taken very carefully.
- After the acquisition of data for the operation with GVF=0, the air line is opened and the suction pressure (p_3) is adjusted by the pressure regulator. Once air begins to be injected in the separator (in the two-phase stream) the separator pressure can be controlled by the PID1 controlled, through valve VCF1.
- With the inlet pressure (p_3) maintained constant by the pressure regulator (located downstream the compressor) the liquid flow rate is gradually decreased by decreasing the PCP rotation speed. In this way the inlet GVF is increased. Data from all sensors is acquired for each value of GVF which is increased at about 10% for each acquisition. This step is the same for the standalone and serial operation. However, while the downstream pump rotation (N_2) is kept constant along the whole experiment for the standalone operation, in the case of serial operation, the rotation of upstream pump (N_1) is kept constant and the rotation of the downstream pump is controlled by the intermediate pressure, which is set to the optimal value for step multiphase compression.

2.3 Definition of performance parameters

The parameters used to evaluate the performance of the twin-screw pumps in the experiments, as well as to compare the performance of a standalone operation versus serial arrangement, were volumetric efficiency and energetic efficiency.

The volumetric efficiency is defined as the ratio between the actual flow admitted in the suction and the theoretical flow-rate of the pump, which corresponds to the displaced volume rate, for a given rotation,

$$\eta_{vol} = \frac{\dot{V}_{meas}}{\dot{V}_{theor}} \tag{2}$$

where $\dot{V}_{meas}$ is the sum of the liquid and gas flow-rates admitted at the pump suction, $\dot{V}_L$ and $\dot{V}_{G@SUC}$, respectively

$$\dot{V}_{meas} = \dot{V}_L + \dot{V}_{G@SUC} \tag{3}$$

where $\dot{V}_L$ is measured directly from the Coriolis liquid flow-meter and $\dot{V}_{G@SUC}$ is obtained from the thermal gas flow-meter, corrected by the temperature and pressure measured at the pump suction.

The theoretical flow-rate, in turn, depends only on the pump geometry and rotation speed (se e.g. Rabiger *et al.* (2006))),

$$\dot{V}_{theor} = k\dot{N} \tag{4}$$

where k is the theoretical volume displaced by one rotation, equal to 0.000435 m^3/rev for the prototypes tested, and $\dot{N}$ is the rotation speed.

It is noteworthy that the volumetric efficiency only takes into account the effects of fluid return from the first cavity to the pump suction. The recirculation of liquid inside the pump, which compresses the gas, between the discharge and the other cavities is not accounted for in this parameter. This is why, while the volumetric efficiency trend to increase with the increase of GVF at suction (before the initiation of "prime loss"), once the pressure distribution becomes non-linear and the differential pressure between the first cavity and suction is lower, the energetic efficiency trends to decrease, as the internal liquid recirculation and re-pumping is larger that the case of single phase liquid flow.

In the case of serial operation, the gas flow rates in upstream and downstream pumps are different due to the gas compression. Therefore, volumetric efficiency is defined in terms of the displaced volumetric flow at the upstream pump, i.e.,

$$\eta_{vol-serial} = \frac{\dot{V}_l + \dot{V}_{G@SUC1}}{k_1 \dot{N}_1} \tag{5}$$

Pump energy efficiency is defined as the ratio between the power delivered for the fluid that is transported from the suction to the pump discharge and the power consumed by the pump. The difference between these occurs due to the viscous effects but also due to the energy losses associated with the pumping work of the fluid that returns through the gaps.

The power consumed by the pump is defined as the product of the measured torque, Tor, and the pump rotation speed, $\dot{N}$,

$$\dot{W}_{mec} = Tor\dot{N} \tag{6}$$

To assess the power delivered to the fluid and the corresponding "ideal process", some options arise. In the literature on multiphase pumps, isothermal or isentropic compressions are considered as limiting forms for the ideal compression process for the gas phase, while the liquid phase is considered isochoric. Unlike the case of pumping incompressible liquid or the compression of a gas, there is no 'ideal" (iso-entropic) equivalent phenomenon in the case of compression of two-phase mixtures. This is because, even considering a frictionless process in an adiabatic machine, without recirculation, there is a heat transfer process between the phases inside the pump which is inherently irreversible. It is for this fact that, low to moderate GVFs, the phenomenon is sometimes considered isothermal, since the thermal inertia of the liquid phase is much larger than that of the gas phase. However, as the gas compression process and the interfacial heat transfer take place simultaneously, there is no "ideal" equivalent reference machine.

Thus, in the authors' opinion, the best option is to calculate the ideal work delivered to the mixture as the sum of the ideal work of gas compression (isentropic) and the ideal work delivered to the liquid (isochoric and isentropic), **as if these processes were performed on separately ideal pump and compressor**.

The ideal work on the the liquid phase, assumed incompressible, can be calcualted as,

$$\dot{W}_l = \int_{p_{suc}}^{p_{desc}} \dot{V}_l dp = \dot{V}_l (p_{desc} - p_{suc}) \tag{7}$$

and, for gas phase, assuming an adiabatic and reversible (iso-entropic) compression process,

$$\dot{W}_{g,adiab} = \int_{p_{suc}}^{p_{desc}} \dot{V}_g dp = \frac{n}{n-1} p_{suc}\, dot V_{g,suc} \left[\left(\frac{p_{desc}}{p_{suc}} \right)^{\frac{n-1}{n}} - 1 \right] \tag{8}$$

where n is given by the ratio between C_p and C_v.

Then, the global pump efficiency is calculated as,

$$\eta_{adiab} = \frac{\dot{V}_l (p_{desc} - p_{suc}) + \frac{n}{n-1} p_{suc} \dot{V}_{g,suc} \left[\left(\frac{p_{desc}}{p_{suc}} \right)^{\frac{n-1}{n}} - 1 \right]}{Tor\dot{N}} \tag{9}$$

For the case of step multiphase compression, through the serial arrangement of pumps, the adiabatic efficiency was determined by the sum of the powers delivered for the compression of gas and liquid in each pump as well as the sum of the powers consumed in each pump, therefore

$$\eta_{adiab} = \frac{\dot{V}_l (p_{desc,2} - p_{suc,1}) + \frac{n}{n-1} \left\{ p_{suc1} \dot{V}_{G@SUC} \left[\left(\frac{p_{desc1}}{p_{suc1}} \right)^{\frac{n-1}{n}} - 1 \right] + p_{suc2} \dot{V}_{G@SUC2} \left[\left(\frac{p_{desc2}}{p_{suc2}} \right)^{\frac{n-1}{n}} - 1 \right] \right\}}{Tor_1 \dot{N}_1 + Tor_2 \dot{N}_2} \tag{10}$$

In this case, the gas flow, $\dot{V}_{G@SUC2}$, corresponds to the flow measured by the thermal gas flow-meter, corrected by the temperature and pressure values measured between the pumps (p_4, T_4).

The sub-indices 1 and 2 represent the parameters referring to the upstream and downstream pumps, respectively.

3. RESULTS

In this section the performance of the pumping system based on the serial arrangement of twin-screw pumps will be analyzed. First, the performance of a single pump is evaluated, once this is the first test for the operation of this pump prototype in two-phase conditions and no operational data for this condition is available. In addition, the pump standalone operation in two-phase condition must be characterized for a large range of rotation velocities once, in the serial operation the downstream pump will operate in different rotations, depending on the inlet GVF, even for a constant rotation of the upstream pump.

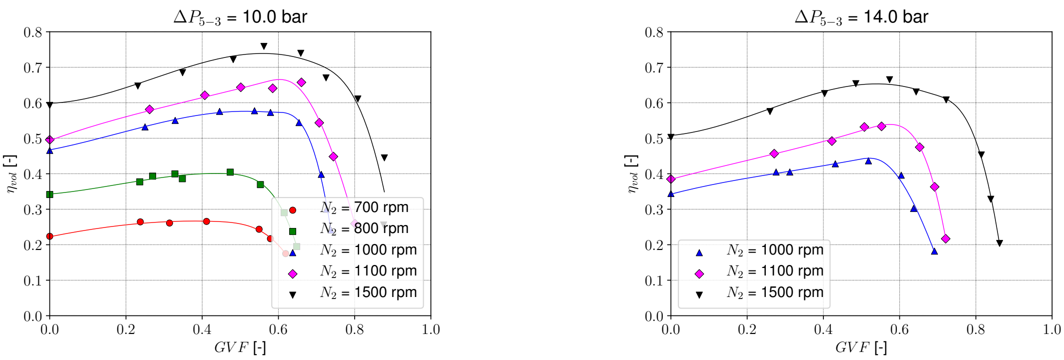
3.1 Performance analysis of a single pump

The experimental conditions considered for the performance characterization of the twin-screw pump in standalone operation are shown in Tab.1. The parameter ΔP_{5-3} = P_5 - P_3 represents the differential pressure across the pump and P_3 the suction pressure. Due to operational limitations of the flow loop and the tested pumps themselves, not all operational conditions could be evaluated for the whole operational envelope. For instance, as the differential pressure is increased, the volumetric efficiency is reduced and the pump operation at lower rotations turn unstable (flow rate oscillates around zero). Therefore, the test could only be performed for higher rotations as differential pressure is increased. On the other side, for high rotations and low differential pressures, the pumped flow rates extrapolated the range of flow rate sensors, in particular, the Coriolis liquid flow-meter.

Table 1: Experimental conditions for the characterization of the performance of the twin-screw pump in two-phase conditions

Experiments	ΔP_{5-3} = P_5 - P_3	P_3	N_2
1	7.0 bar	1.5 bar	700 rpm
2			800 rpm
3			1000 rpm
4			1100 rpm
5	10.0 bar	1.5 bar	700 rpm
6			800 rpm
7			1000 rpm
8			1100 rpm
9			1500 rpm
10	12.0 bar	1.5 bar	800 rpm
11			1000 rpm
12			1100 rpm
13			1500 rpm
14	14.0 bar	1.5 bar	1000 rpm
15			1100 rpm
16			1500 rpm

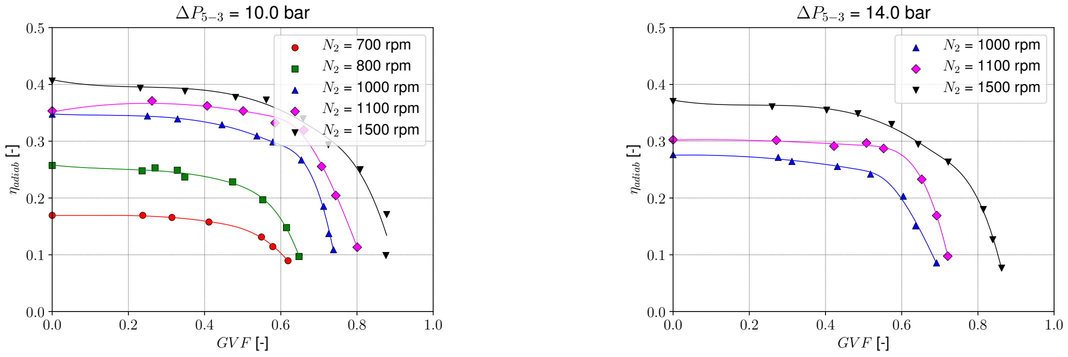
The volumetric efficiency for different rotation, differential pressure and suction GVFs for the twin-screw pumps is shown in Fig. 2. As expected from the theoretical analysis, the volumetric efficiency trend to increase with GVF, due to the axial pressure distribution and then trend to decrease the axial pressure gradient at the suction (Vetter and Wincek (1993), Rabiger *et al.* (2008), PATIL (2000)). However, for a given value of GVF, which depends on differential pressure and rotation, the prime loss process is initiated and a steep decrease is observed in the volumetric efficiency as GVF is increased. It is noteworthy the influence of the rotation speed on the volumetric efficiency and also on the value of the GVF for the initiation of prime loss.



(a) (b)

Figure 2: Volumetric efficiency η_{vol} of the twin-screw pump as varying with *GVF* for different values of differential pressure ΔP_{5-3} and rotation speed N_2.

Figure 3 presents the energetic efficiency considering an adiabatic process, as calculated by Eq. 9. This efficiency takes into account all losses, once it relates the shaft power delivered to the pump with the actual energy gained by the fluids, as if they were compressed in an ideal process separately. Therefore it accounts for mechanical losses (viscous and



(a) (b)

Figure 3: Energetic efficiency, considering adiabatic process, η_{adiab} of the twin-screw pump as varying with *GVF* for different values of differential pressure ΔP_{5-3} and rotation speed N_2.

friction in mechanical components, as gears and bearings), recirculation (re-pumping) losses and also mechanical energy degradation due to heat transfer between phases (this can be seen as if a gas is compressed iso-entropically, reaching a given temperature, and then mixed with the liquid, which is at the suction temperature). However, one of the main loss component at moderate to high GVFs and high differential pressures is the internal liquid recirculation and re-pumping. In fact, as the pump displacement is constant, i e., the mixture volume is not reduced along the process, it is the returning liquid that promotes the gas compression (Rausch *et al.* (2005), Rabiger *et al.* (2008)).

3.2 Performance of the step multiphase compression

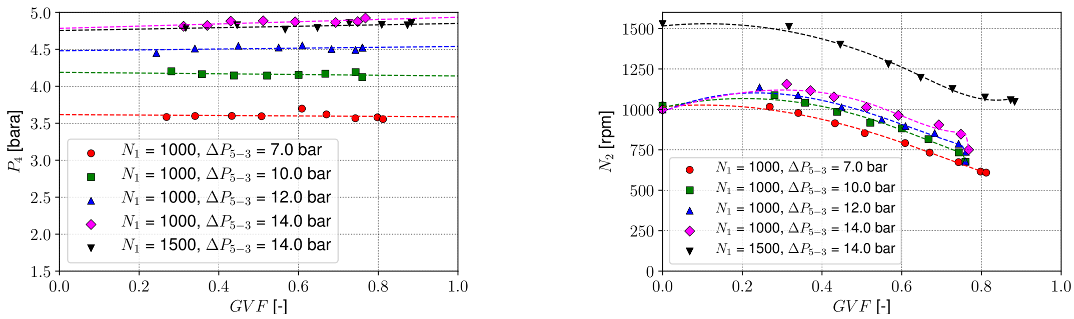
In the serial operation experiments, the rotation of upstream pump is kept constant while the rotation of the second pump is controlled by the intermediate pressure p_4, which is set to the optimum value that minimizes the shaft work consumed in both pumps. From a thermodynamic analysis of the compression process of a gas-liquid mixture, considering the fact that the shaft work consumed by each pump is independent of the suction GVF, (Palioff *et al.* (2015)), this pressure is given by $p_4 = \sqrt{p_5 \times p_3}$. This pressure does not depend on inlet GVF or any other parameters than the suction and discharge pressures, since a small amount of gas flow with the liquid (considered incompressible). However, for operation with pure liquid (GVF=0) the intermediate pressure must be set to the average of suction and discharge pressures, $p_4 = 0.5 \times (p_5 + p_3)$. Therefore, in the case of operation with pure liquid, the pressure increment is equally divided between pumps.

Table 2: Experimental conditions for the characterization of the performance of the twin-screw pumps in serial arrangement

Experiments	$\Delta P_{5-3} = P_3 - P_2$	$\Delta P_{pump1} = P_4 - P_3$	$\Delta P_{pump2} = P_5 - P_4$	N_1
1	7.0 bar	2.1 bar	7.9 bar	1000 rpm
2	10.0 bar	2.7 bar	8.8 bar	1000 rpm
3	12.0 bar	3.0 bar	10.5 bar	1000 rpm
4	14.0 bar	3.8 bar	11.7 bar	1000 rpm
5	14.0 bar	3.8 bar	11.7 bar	1500 rpm

The operating conditions considered in the experiments for serial multiphase compression are listed in Tab. 2. The resulting Δp for each pump are also shown. However, these values are only considered for GVF> 0. In the serial operation, only two rotation speeds of the upstream pump (related to the "admitted" volumetric flow rate in the system) were considered. However, it must be taken into account that the rotation speed of the downstream pump varies with the suction GVF. Therefore, in order to adequately characterize the operation of each pump in serial arrangement, a larger range of rotation speeds was considered in the experiments, for the pump in standalone operation.

Figures 4 (a) and (b) present, respectively, values of the intermediate pressure, p_4 and the corresponding rotating velocities of the downstream pump N_2, obtained from the control system, through CF2 inverter. From theoretical point of view, the rotation speed N_2 should decrease as GVF is increased, once the mixture volume at upstream pump discharge (downstream pump suction) is reduced. However, as the volumetric efficiency of the first pump increases with the GVF, the volumetric flow rate at upstream pump discharge is first increased, while the increment due to the increase in η_{vol} overcomes the reduction due to the gas compression, and then decreased.



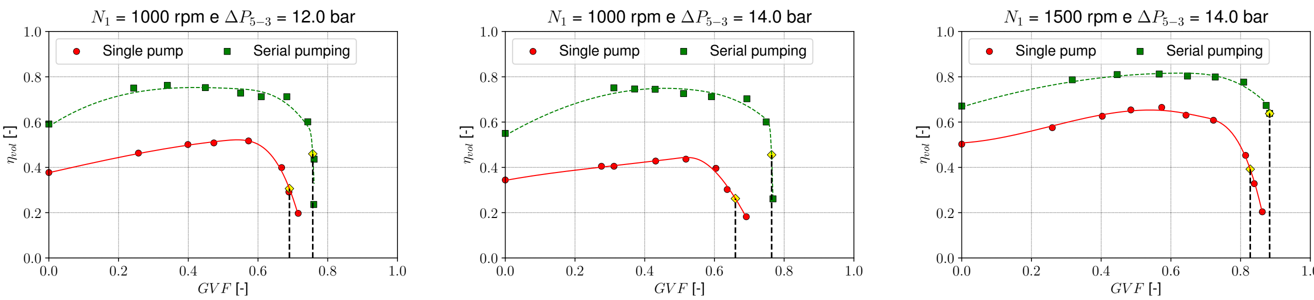
(a) (b)

Figure 4: Intermediate pressure p_4 (a) and rotation N_2 of downstream pump (b), as function of suction GVF for different ΔP_{5-3} for the wot rotations of the upstream pump tested in the experiments (N_1 =1000 and 1500 rpm)

Figures 5 and 6 present the volumetric and energetic efficiencies of the pumping system in serial arrangement. Clearly, both efficiencies are significantly increased with the serial arrangement.

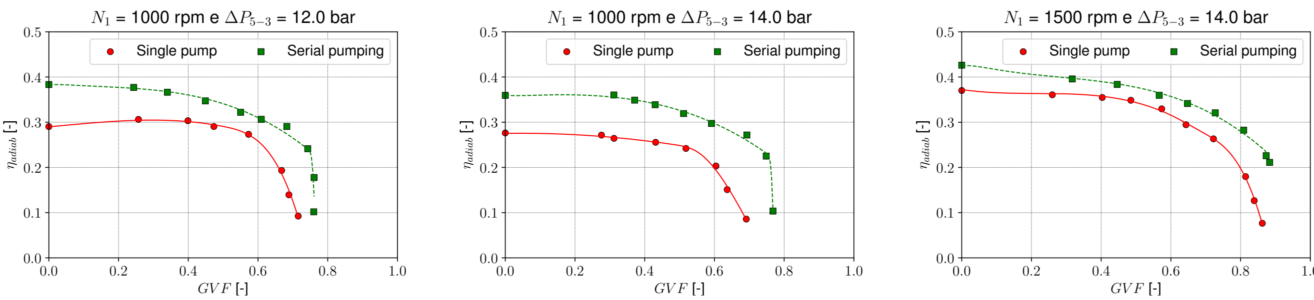
Another important parameter of these multiphase pumps is the maximum GVF that they can attain at suction without prime loss. However, when approaching the prime loss GVF, the pump operation turns unstable and, therefore, it is difficult to determine this point for all experiments. Instead, we define a parameter, $GVF^{0.6}$ which represents the GVF value at which the volumetric efficiency attains 60% of the maximum efficiency. Although this parameter does not represent exactly the GVF for prime loss, it clearly indicates the GVF at which the steep decrease in volumetric efficiency is initiated. In Fig. 5 these points are indicated with yellow rhombus.

In Figure 7 (a) the $GVF^{0.6}$ parameter is shown, as a function of the differential pressure, for the pumps functioning in standalone operation and serial arrangement and, for the same rotation speed, its value remains high for larger values of differential pressure. In Figs. 7 (b) and (c) the results for volumetric and energetic efficiency are summarized, plotted against the differential pressure. Clearly, these efficiencies are higher in all cases but the effect of step compression is more evident for higher differential pressures. Beyond the more efficient operation, one of the objectives of this type of studies would be to evaluate the possibility of increasing the available differential pressure considering the current state of the technology in multiphase pumps. In this study, for comparison purposes, the tests were developed considering the same differential pressure for standalone and serial operation conditions. However, even for relatively low rotations of the first pump (N_1 = 1000 rpm) the volumetric efficiency is much less reduced as differential pressure increases. Therefore, it would be expected that the system was be capable of operating at large values of Δp with high volumetric efficiencies.



(a) (b) (c)

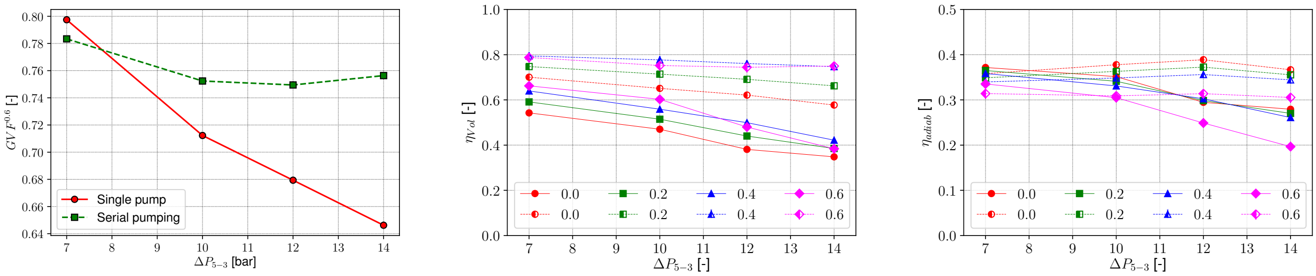
Figure 5: Volumetric efficiency as function of suction GVF for different ΔP_{5-3} for the two rotations of the upstream pump tested in the experiments (N_1 =1000 and 1500 rpm). Yellow seuare indicate the values of $GVF^{0.6}$ parameter for each operational condition



(a) (b) (c)

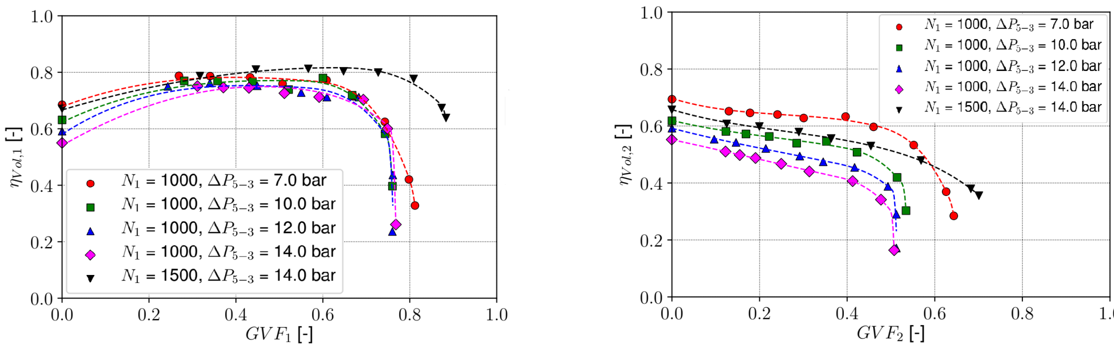
Figure 6: Energetic efficiency as function of suction GVF for different ΔP_{5-3} for the two rotations of the upstream pump tested in the experiments (N_1 =1000 and 1500 rpm)

Figures 8 and 9 present, respectively, the volumetric and energetic efficiencies of the individual pumps, when operating in serial arrangement. The efficiency values are plotted against the suction GVFs of each pump, i.e., as if each pump would be operating with that suction condition. In addition, it must be recalled that the Δp which each pump is subjected, varies with the total Δp (as can be seen in Table 2). However, while the Δp for the upstream pump remain at relatively low values for all operating condition, the value for the second pump is larger and more variable. Therefore, while the volumetric and energetic efficiencies of the first pump remain close for all operational conditions, the second pump operates at relatively low efficiencies. In addition, as observed in the results for standalone operation, the volumetric and energetic efficiencies trend to steeply decrease with rotation speed. Therefore, as the downstream pump rotation trend to decrease as suction GVF is increased, it ends up operating at too low rotation velocities, resulting in the low efficiencies observed and also affecting the whole system efficiency. A recommendation for this issue would be the use of a lower displacement pump as the downstream pump when the operation is predominantly with moderate to high GVFs. However, this could lead to high rotations of the downstream pump for the case of the operation with liquid or low GVFs.



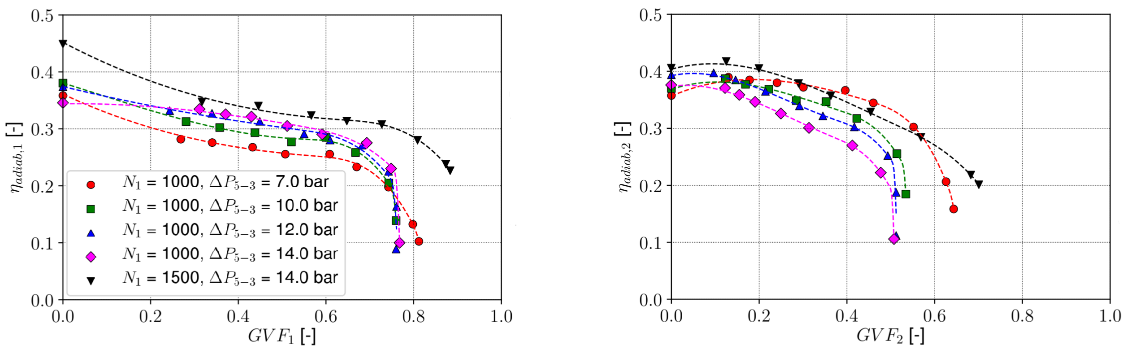
(a) (b) (c)

Figure 7: $GVF^{0.6}$ parameter (a) and Volumetric (b) and energetic (c) efficiencies as function of the differential pressure $p_5 - p_3$ compared for standalone (filled symbols) and serial operation (partially filled symbols) for a rotating velocity of upstream pump N_1 = 1000 rpm



(a) Pump 1 (b) Pump 2

Figure 8: Volumetric efficiency η_{vol} of the individual pumps, operating in serial arrangement as function of the inlet $GVFs$ of each pump. The GVF_2 is estimated form the measured volumetric flow rates at suction of firs pump, corrected by the intermediate pressure and temperature p_4 and T_4



(a) Pump 1 (b) Pump 2

Figure 9: Energetic efficiency η_{ener} of the individual pumps, operating in serial arrangement as function of the inlet $GVFs$ of each pump. The GVF_2 is estimated form the measured volumetric flow rates at suction of firs pump, corrected by the intermediate pressure and temperature p_4 and T_4

4. CONCLUSIONS

This work presented a systematic analysis of the multiphase compression processes through the serial arrangement of twin-crew pumps. Despite the fact that the pumps tested do no represent the current state of the technology available for multiphase pumping, the use of this type of pumps in serial arrangement appears very promising, in terms of volumetric and energetic efficiencies. Analyzing the energetic efficiency of the pumps operating in this condition, it is expected that, beyond the energy consumption issues, problems related to the shortening of lifespan as uneven distribution of the power transferred to the fluids, and screw deflection, be reduced leading to large usable periods, which is critical in deep water applications. In addition, the serial arrangement can be used to increase the currently available differential pressure of the multiphase pumps.

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6. REFERENCES

Egashira, K., Shoda, S., Tochikawa, T., Furukawa, A. *et al.*, 1998. "Backflow in twin-screw-type multiphase pump". *SPE production & facilities*, Vol. 13, No. 01, pp. 64–69.

Liu, P., Morrison, G. and Patil, A., 2019. "Experimental and analytical investigation of a novel multistage twin-screw pump". *Journal of Energy Resources Technology*, Vol. 141, No. 12. ISSN 0195-0738. doi:10.1115/1.4044975. URL `https://doi.org/10.1115/1.4044975`. 122101.

Liu, P., Patil, A. and Morrison, G., 2017. "Multiphase flow performance prediction model for twin-screw pump". *Journal of Fluids Engineering*, Vol. 140, No. 3. ISSN 0098-2202. doi:10.1115/1.4038080. URL `https://doi.org/10.1115/1.4038080`. 031103.

Palioff, F., Paladino, E.E., Carvalho da Silva, A.F. and Brito, R.M., 2015. "Estudo numérico e experimental da termo-fluidodinâmica de bombas multifásicas de duplo parafuso - primeiro relatorio técnico". Technical report, SINMEC Lab - Universidade Federal de Santa Catarina.

PATIL, A.R., 2000. *PERFORMANCE EVALUATION AND CFD SIMULATION OF MULTIPHASE TWIN-SCREW PUMPS*. Ph.D. thesis, Texas A & M University.

Patil, A. and Morrison, G., 2017. "Performance of multiphase twin-screw pump during the period of wet-gas compression". *SPE Production & Operations*, Vol. 33, No. 01, pp. 68–72. ISSN 1930-1855. doi:10.2118/186099-PA. URL `https://doi.org/10.2118/186099-PA`.

Rabiger, K., Maksoud, T.M. and Ward, J., 2006. "Thermo- and fluid dynamic model of a multiphase screw pump , operating at very high gas volume fractions". In *Sonderdruck Schriftenreihe Georg-Simon-Ohm-Fachhochschule Nürnberg*. p. 18.

Rabiger, K., Maksoud, T.M., Ward, J. and Hausmann, G., 2008. "Theoretical and experimental analysis of a multiphase screw pump, handling gas-liquid mixtures with very high gas volume fractions". *Experimental Thermal and Fluid Science*, Vol. 32, No. 8, pp. 1694–1701. ISSN 08941777. doi:10.1016/j.expthermflusci.2008.06.009.

Rausch, T., Vauth, T., Mewes, D., Reichwage, M. *et al.*, 2005. "Experimental and theoretical investigations of multiphase twin screw pumps for new offshore applications". In *The Fifteenth International Offshore and Polar Engineering Conference*. International Society of Offshore and Polar Engineers.

Vauth, T., Rausch, T., Mewes, D. and Reichwage, M., 2004. "Test facility for multiphase pumps in serial and parallel operation". In *ASME 2004 International Mechanical Engineering Congress and Exposition*. American Society of Mechanical Engineers, pp. 449–454.

Vetter, G. and Wincek, M., 1993. "Performance prediction of twin screw pumps for two-phase gas/liquid flow". *ASME-PUBLICATIONS-FED*, Vol. 154, pp. 331–331.

Vetter, G., Wirth, W., Korner, H. and Pregler, S., 2000. "Multiphase pumping with twin-screw pumps-understand and model hydrodynamics and hydroabrasive wear". In *PROCEEDINGS OF THE INTERNATIONAL PUMP USERS SYMPOSIUM*. pp. 153–170.

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