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Experimental investigations on a two-phase jet pump used in desalination systems

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Abstract

With an ever-increasing population and rapid growth of industrialization, there is a great demand for fresh water, especially for drinking. In view of this, different desalination technologies have evolved with a thrust for utilization of renewable energy sources such as solar energy, ocean thermal energy, geothermal energy, and waste heat. Flash desalination is a technology in which fresh water is produced from seawater by evaporation and subsequent condensation. One such desalination technique involves different processes like pressurization of seawater using a pump, creation and maintenance of vacuum using jet pump in a flash chamber, evaporation of seawater at reduced pressure using warm water from the sea surface and condensation of water vapour using cold water from the deep sea. A two-phase variable geometry jet pump was designed, developed and tested for the flash desalination system to create a vacuum inside the flash chamber. The area ratio of the jet pump used for the investigation was 0.25. Experimental results are reported from tests on the jet pump with a view to assessing the optimum distance between orifice exit and mixing tube entrance. Overall performance of the jet pump was assessed by plotting its characteristics. Performance of the jet pump was analyzed by plotting iso-efficiency curves on characteristics curves drawn for various primary water flow rates. The optimum orifice to mixing tube distance of the jet pump was found and corresponding maximum efficiency was reported. Maximum vacuum created by the jet pump at an optimum condition, i.e., at maximum jet pump efficiency, with respect to time was determined.

Keywords: Jet pump; Orifice spacing; Two phase; Flash desalination

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1. Introduction

Water is a basic natural resource and plays a crucial role in the overall development of the economy of any country. The scarcity of drinkable water exists in most of the coastal areas due to seepage of seawater into the water aquifer. Also, due to an ever-increasing population and rapid industrialization, the natural supply of fresh water is unable to cope with the increasing demand. So, the conversion of sea or brackish water into potable water through desalination techniques has been strongly brought into focus. Converting brackish or seawater into potable water is an energy-intensive process. It is only viable if the source of energy is practically free and available in abundance like solar energy, geothermal source, ocean thermal gradient, waste heat, etc.

Flash desalination is one such technology, in which fresh water is produced from sea or brackish water by evaporation and subsequent condensation. This desalination technique involves different processes like pressurization of seawater using a pump, creation and maintenance of a vacuum using a jet pump, evaporation of seawater at reduced pressure using warm water from the sea surface and condensation of water vapour using cold water from the deep sea. The main advantages of this desalination system are low maintenance cost and utilization of lower grade energy.

A jet pump was designed, developed and analyzed for this flash desalination system to create and maintain a vacuum inside the flash chamber. A jet pump is a device capable of pumping fluids to a higher pressure employing an orifice, mixing chamber and diffuser combination. It has the following merits: an unsurpassed simplicity on account of the absence of moving parts, low initial and maintenance costs, absence of sealing problems and suitability for high compression ratios. Further, a liquid jet air pump has generally a better efficiency than a steam ejector and can attain much higher compression ratios. The good sealing possibilities and the absence of moving parts suggest applications in the pumping of corrosive, toxic, radioactive or dust laden gases. The design of a liquid jet air pump to provide maximum efficiency requires either a correct mixing tube length or orifice to mixing tube spacing to help the jet break up characteristics.

A jet pump with orifice was selected since it is the best in terms of producing liquid jets with superior air induction effectiveness, ease of manufacture and pump operating stability at all flow ratios. The area ratio of the jet pump was taken as 0.25 for the present work. The experimental investigation led to determination of the effect of distance between the driving orifice exit to mixing tube entrance (S), the suction rate, etc., and also to understand the mixing process and optimization of the jet pump performance. For testing the above, the supply tube with orifice was moved axially inside the jet pump from S = 28 to 38 mm in four intervals and other parameters such as mixing tube length, diffuser length, diffuser angle, etc., were held constant. Performance of the jet pump at four orifice distances was studied. Iso-efficiency curves were obtained on performance curves. With the help of these characteristic curves, it is possible to locate the point of best performance of the jet pump.

2. Previous investigations

The application of the liquid jet gas (LJG) pump for evacuation of steam condensers was discussed by Hoefer [1]. Rammingen [2] first reported the rapid mixing phenomenon and sudden rise in mixing tube pressure within a few times diameters, distance which has been noted by most investigators in subsequent years. An one dimensional analysis of the LJG pump mixing process in a cylindrical mixing tube (no diffuser) was reported by Folsom [3]. Takashima [4] developed a similar expression for mixing tube as well as a diffuser, and also reported experimental data. Somewhat better performance was achieved with a 4-hole nozzle compared with a single jet nozzle. Witte's [5,6] contributions are noteworthy, particularly his study of flow processes in the mixing tube and the "mixing shock" as he termed it. Experimentally he demonstrated high volumetric entrainment ratios and accompanying high isothermal compression efficiencies, up to 40%, by means of multi-hole nozzles and a relatively long mixing tube.

Witte [7] was the first to plan LJG pump tests using a dimensionless Euler number, *U*. Most experimental work in this field has been marred by incomplete data and widely varying test procedures. The minimum data required to assess performance are the two flow rates (primary and secondary), three pressures (upstream, downstream pressures of nozzle and diffuser pressure), and pump geometry, particularly nozzle diameter, mixing tube diameter and mixing tube length. Betzler [8] improved the analysis, particularly for the diffuser. He demonstrated good agreement between theory and experiment, provided the mixing zone remained in the mixing tube. Design refinements – particularly mixing tube lengths up to 23 diameters resulted in isothermal efficiencies as high as 19%.

A number of similar application references were included in the BHRA literature survey of jet pumps [9], but in many cases the information provided has been inadequate to describe pump operation. Bonnington [9] showed that measured efficiencies were decreasing with increasing jet velocities. He made an important contribution in reporting tests with a transparent mixing tube: best performance was obtained when the mixing zone was positioned in the cylindrical mixing tube section.

Cunningham [10] and Cunningham and Dopkin [11] presented the performance of various flow obstruction devices and reported that the one with orifice gives maximum efficiency for maximum flow rate. Schmitt [12] presented one-dimensional relations for the LJG pump including friction loss coefficients, and compared experimental results with theory. Isothermal compression efficiencies of about 10% were measured, using liquid–liquid pump configurations with short (four diameter) mixing tube lengths. Unaware of Takashima's results, he also tested pumps with transparent-mixing tubes. With a fixed water jet velocity and suction port air pressure, he found that the mixing zone could be positioned at will, by controlling the back-pressure. At the upper limit, the pump "flooded", i.e., water flowed back through the air inlet. With the discharge valve fully open, the jet passed through the mixing tube and mixing occurred in the conical diffuser. Optimum performance occurred when mixing was located just at the end of the mixing tube exit, i.e., at the diffuser entrance. Experimental efficiency points followed the theory fairly well but only if mixing occurred in the mixing tube; efficiency declined rapidly at back-pressure values low enough to permit jet penetration into the diffuser. A long (24-diameter) mixing tube produced efficiencies as high as 13%.

Cunningham [13] developed a one-dimensional model for liquid jet pump handling two-phase gas in liquid bubbly mixtures, and examined the characteristics of this LJGL pump. The LJGL model also encompassed the secondary flow extremes of a liquid and a gas, i.e., the LJGL model bridged and linked two established jet pumps. Sherif et al. [14] described in their paper the models of the flow of two-phase primary fluids inducing a secondary liquid injected into the jet pump mixing chamber. The model was capable of incorporating the effects of the temperature and pressure dependency in the analysis. The approach adopted an isentropic constant pressure mixing in the mixing chamber and at times employed interactive techniques to determine the flow conditions in different parts of jet pump.

To summarize the experimental results, both Witte [6] and Bonnington [9] have shown that isothermal efficiencies of 40% or better are possible with combinations of jet characteristics (nozzle design), mixing tube length and operating conditions. Experimental work otherwise, however, has generally involved short-mixing tube pumps, i.e., similar to liquid–liquid pumps, and reported efficiencies have been less than 10% in most cases.

Witte [6] optimized a fixed-mixing tube pump by successively refining nozzle design, finding that a 19-hole nozzle was best. Experimentally this program concentrated on a study of jet break-up characteristics and mixing tube length optimization, with the single jet nozzle as the basic configuration.

Experimental performance of a jet pump assisted vacuum desalination plant was given by Senthil Kumar et al. [15] for a typical operating condition. A method of designing of jet pump and selection of horizontal axis multi-stage centrifugal pump with variable frequency drive to provide high head motive fluid to jet pump was given by Senthil Kumar et al. [16]. The same authors [17] reported an analysis of a jet pump assisted vacuum desalination system. By applying the mass and energy balances across the various components such as the evaporator, condenser, etc., the governing equations were obtained. These equations were solved using simulation. This study was carried out by varying operational parameters like evaporator temperature, condenser temperature, evaporator flow rate, condenser flow rate and chamber pressure created by the jet pump. Fresh water (yield) obtained from the system increases as condenser temperature decreases and the evaporator temperature increases. Further, the yield increases as the chamber pressure decreases.

3. Apparatus and procedure

3.1. Two-phase jet pump

A schematic representation of the two-phase jet pump is shown in Fig. 1. The jet pump for this investigation consists of the following elements: orifice, suction chamber, orifice spacing shims, secondary inlet, mixing tube and diffuser. The primary fluid, i.e., water is pressurized by an independent source and leaves the orifice as a core of high velocity. This high velocity jet entrains secondary fluid in the direction of the driving fluid. The acceleration of the particles of the surrounding (suction) fluid is achieved by impact of particles from the driving orifice fluid in the mixing tube. Finally conversion of kinetic energy to a pressure rise of mixed fluid takes place in the diffuser. The jet pump is designed using equations of continuity and momentum. Table 1 shows the design details of jet pump.

3.2. Test facility

The test facility created for this purpose was an open loop, continuous circulation system. Fig. 2 shows the schematic diagram of the test rig. The jet pump test section was fabricated from acrylic plastic to have visual observations. The jet

Table 1
Jet pump dimensions

Part name	Dimension
Orifice diameter	5 mm
Suction chamber diameter	50 mm
Mixing tube diameter	10 mm
Length of the mixing tube	262 mm
Distance between orifice exit and mixing tube entrance (*S*)	28 mm
Length of the diffuser	135 mm
Diameter of the diffuser at outlet	21 mm
Diffuser semi cone angle	$2°30'$
Length of jet pump	800 mm

pump was pressure tested first with water and then with nitrogen for withstanding the hydrostatic pressure of 4 bar. It was also tested for sealing against vacuum.

Water from an open tank is pressurized by a multi-stage horizontal axis centrifugal pump having a variable frequency drive and the high head water is supplied to jet pump orifice as the motive fluid. Orifice produces high velocity jet and creates vacuum in the suction chamber; hence, entrainment of secondary air from chamber takes place. The water and air mix thoroughly in the mixing tube. The diffuser converts energy of this mixture partially from kinetic to pressure. Then the mixture returns to water tank through the pipe line.

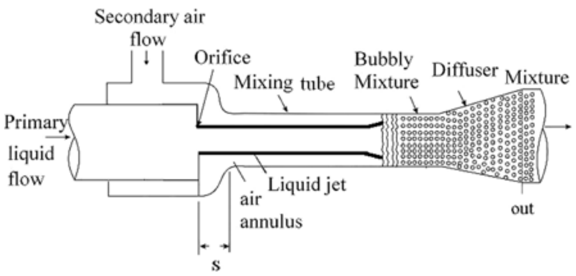

Fig. 1. Schematic diagram of jet pump.

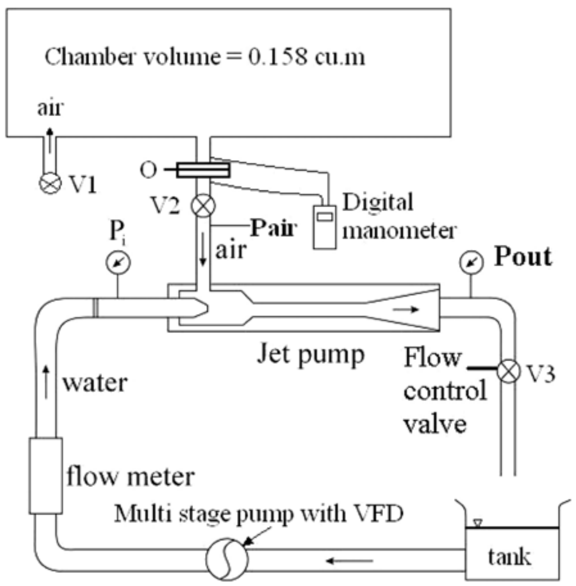

Fig. 2. Schematic diagram of water jet pump test rig.

3.3. Instrumentation

The primary flow rate was measured with a wheel-type flow meter and the secondary air flow with an orifice meter. Three pressures (P_i, P_{air} and P_{out}) were measured and the pump efficiency was determined. A precision bourdon-type pressure gauge was used for measuring input pressure (P_i) of the jet pump. Secondary air pressure (P_{air}) at jet pump suction and pressure at exit of the jet pump (P_{out}) were measured by precision digital compound gauges. A vacuum created in the flash chamber with respect to time was measured by a pressure transducer. Pressure gauge, pressure transducer, flow meter and orifice meter were calibrated adopting standard procedures. The instruments/sensors used for measurement of parameters, their range and accuracies are shown in Table 2.

Table 2
Range of measurements and accuracies

Measurement parameters	Instruments	Range	Accuracy
Chamber pressure	Pressure transducer	0–3 bar (abs)	± 0.5% F.S.
Secondary air flow	Digital manometer	0–35 kPa (abs)	± 0.1% F.S.
Vacuum pressure of jet pump	Digital compound gauge	0–4 bar (abs)	± 0.1% F.S.
Outlet pressure of jet pump	Digital compound gauge	0–4 bar (abs)	± 0.1% F.S.
Primary water flow rate	Wheel type flow meter	0–10,000 lph	± 1 lph
Primary water pressure	Bourdon pressure gauge	0–25 bar	± 2% of F.S.

3.4. Experimentation

Performance was obtained over a wide range of operating conditions for the given jet pump. Test runs were conducted by closing the flow control valve of the jet pump in discrete steps. Hence, flow ratio was varied by keeping primary the water flow rate constant. Valves V1 and V2 were always opened for all operating conditions for optimizing the jet pump. This procedure was followed at four orifice spacings between 28 mm and 38 mm. Several values of flow ratio which spanned the best-efficiency point were obtained and the corresponding peak efficiency was found. Iso-efficiency curves were obtained on performance curves to determine the point of best performance of the jet pump. The jet pump was optimized by following the above procedure. The chamber vacuum created by the jet pump at optimized conditions was determined with respect to time for four orifice spacings. To create the chamber vacuum, valve V1 was closed and V2 was left open.

There are four fundamental jet pump parameters which affect jet pump performance. These are expressed in dimensionless form.

- Orifice to mixing tube ratio, $R = A_o/A_t$
- Secondary to primary flow ratio, $M = Q_{\text{air}}/Q_{\text{w}}$
- Pressure characteristic, N
- Efficiency = the equivalent of the net output power divided by net input power.

Other parameters such as mixing tube diameter, diffuser angle, diffuser length and area ratio were held constant. Orifice spacing and primary flow rates were varied for this study the during experiments.

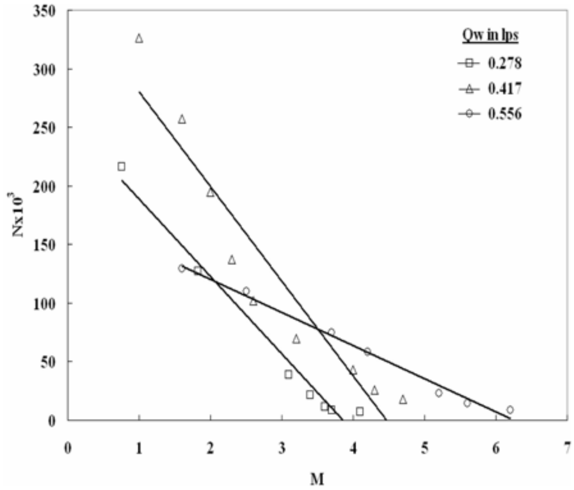

Fig. 3. Plot of M–N curve of jet pump.

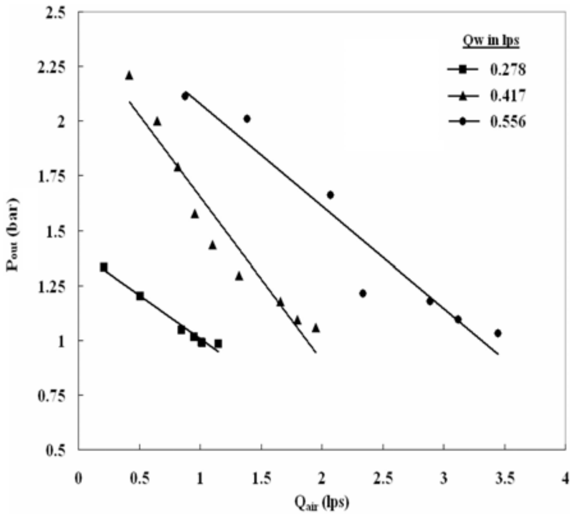

Fig. 4. Dimensional plot of performance of jet pump.

4. Results and discussion

4.1. Overall performance

Jet pump overall performance is a plot of pressure ratio (N) as a function of flow ratio (M). A typical performance achieved in this investigation is shown in Fig. 3 for an area ratio of 0.25 and orifice spacing (S) of 38 mm. The points are grouped into three curves based on the supply flow rate even though they were expected to coincide as one single curve. Thus, in order to understand the real behaviour and performance of a pump under varying flow rates, the dimensional characteristics of the jet pump were plotted.

Fig. 4 discerns the dimensional characteristic plot of the jet pump. It shows the variation of the

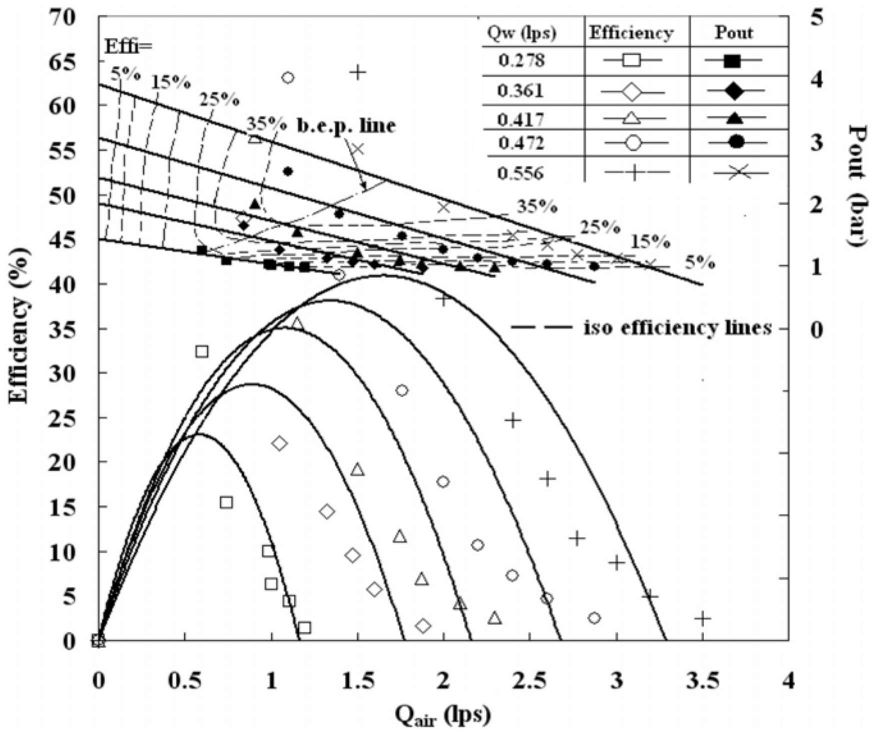

Fig. 5. Jet pump performance at an orifice spacing of 33 mm.

secondary air flow rate on jet pump outlet pressure for water flow rates 0.278, 0.417 and 0.556 lps at an orifice spacing of 38 mm. From Fig. 4, it was found that the three-dimensionalized curves form a pattern as the water flow rate is varied.

Fig. 5 shows the characteristics of the jet pump operated with five primary water flow rates to jet pump for the orifice spacing of 33 mm. For each test, the rate of flow Q_w was varied using a variable frequency drive of a multi-stage pump and the corresponding values of Q_{air}, P_{out}, were measured and overall efficiency was calculated. The same operation was repeated for different discharges by changing the speed of the multi-stage pump. Then P_{out} vs. Q_{air}, efficiency vs. Q_{air} curves for different discharges were plotted (Fig. 5). These curves are useful for arriving at the performance of the jet pump at different supply conditions at an orifice spacing of 33 mm.

Then iso-efficiency curves were plotted on the P_{out} vs. Q_{air} curves. These curves show the efficiencies of the jet pump for all conditions of running and hence these are also known as universal characteristic curves of the jet pump. In order to plot the iso- efficiency curves, horizontal lines representing constant efficiencies were drawn on the efficiency vs. Q_{air} curves. The points at which these lines cut the efficiency curves at various discharges were transferred to the corresponding P_{out} vs. Q_{air} curves. The points corresponding to the same efficiency were then joined by smooth curves which represent the iso-efficiency lines. It is, however, clear from Fig. 5 that the innermost iso-efficiency curve represents the highest efficiency of the jet pump and the outer curves represent the lower efficiencies. Best efficiency points (bep) were found for each flow rate and the bep line was plotted by joining the points of best efficiencies. Here the maximum

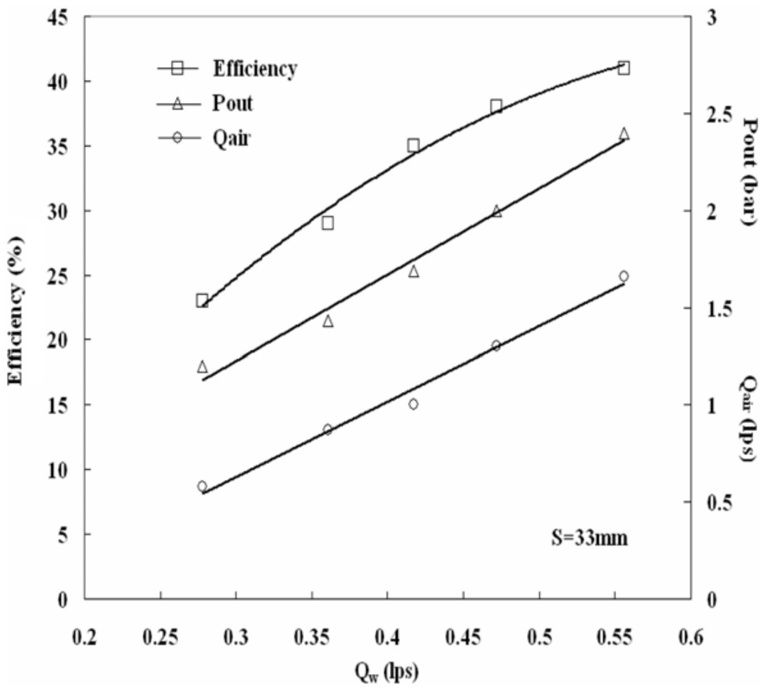

Fig. 6. Effect of change of Q_w on performance of jet pump at bep.

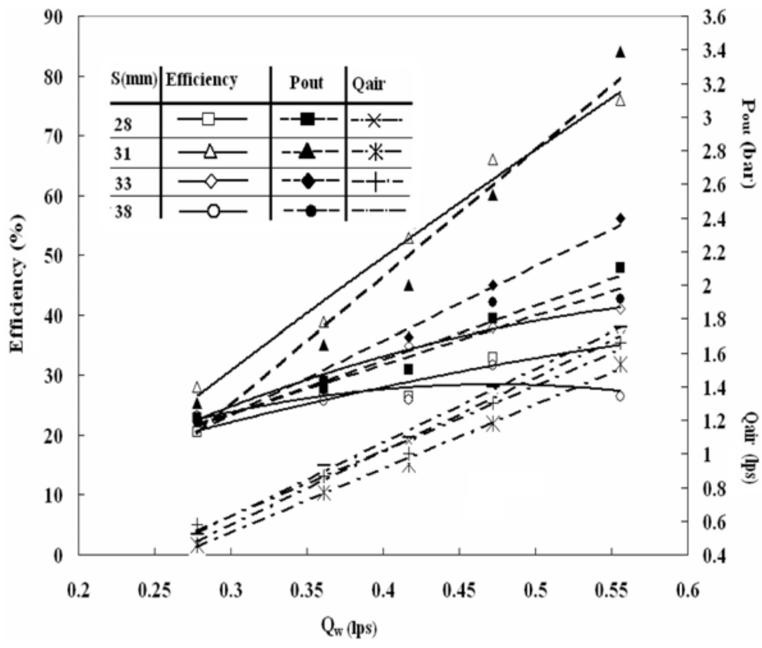

Fig. 7. Effect of Q_w on performance at bep for various orifice spacings.

efficiency of the jet pump realized was 41% for a primary water flow rate of 0.556 lps.

Similarly, experiments were conducted for other orifice spacings (28, 31, 38 mm). During these experiments at all values of *S*, there was flow reversal found when the jet pump was operated at higher primary flow rates higher than 0.556 lps. Hence the experiments have been

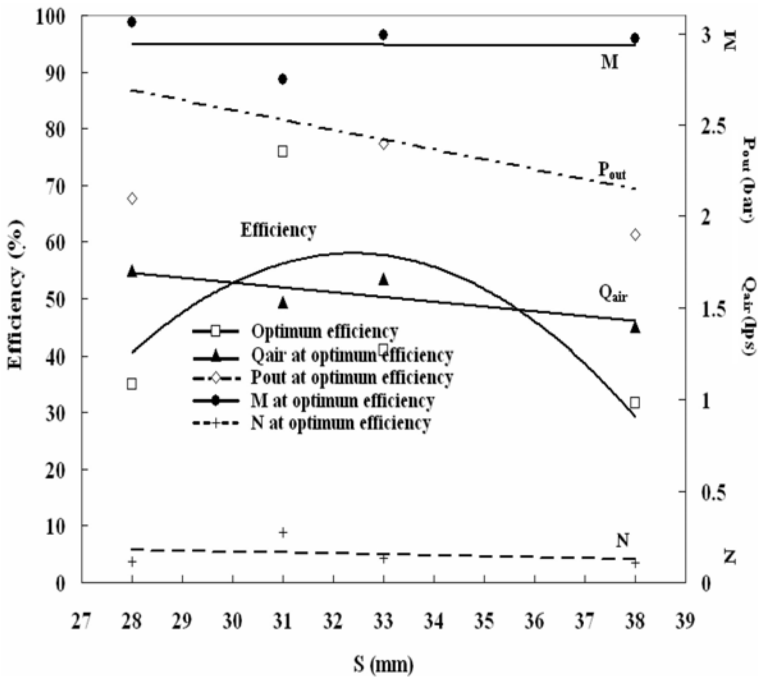

Fig. 8. Effect of orifice spacing on values at bep at maximum Q_w.

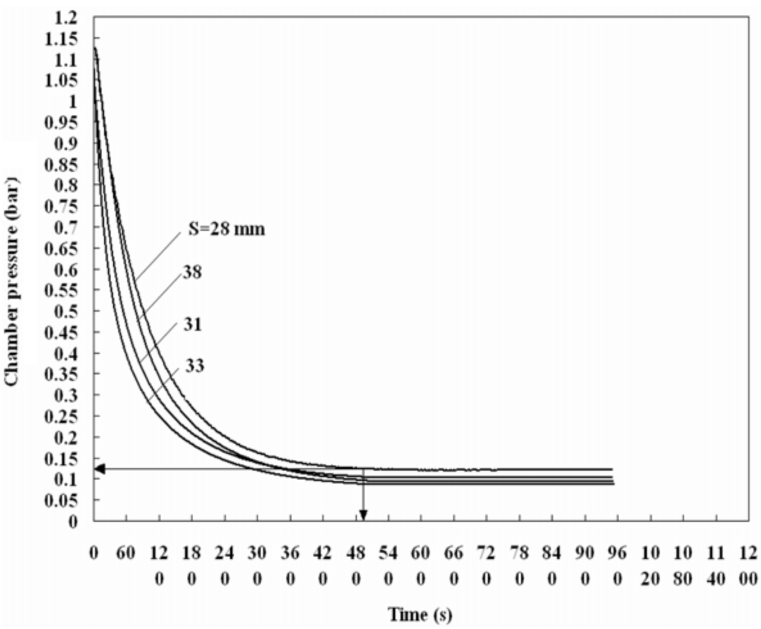

Fig. 9. Variation of chamber pressure with time for four orifice spacings at optimum conditions.

limited to a maximum of a primary flow rate of 0.556 lps.

Fig. 6 shows the effect of change of Q_w on performance of a jet pump at bep. for orifice spacing of 33 mm. At bep, secondary air flow rate, jet pump outlet pressure and jet pump efficiency increase with the increase in Q_w. The maximum efficiency realized was approximately 41% at a higher value of Q_w.

Similarly, the effect of change of Q_w on

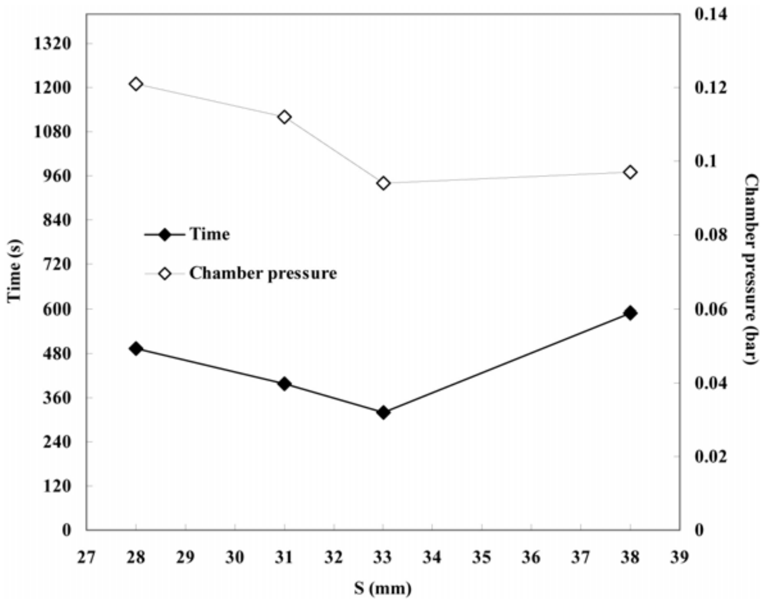

Fig. 10. Effect of orifice spacing S on time and chamber pressure for four orifice spacings at optimum conditions.

performance of the jet pump at bep for other orifice spacings (28, 31, 38 mm) were evaluated and the result is shown in Fig. 7. Generally, efficiency, P_{out} and Q_{air} increase with an increase in Q_w. From Fig. 7, it is found that, at an orifice spacing of 31 mm, maximum efficiency and P_{out} are higher than that for other spacings for almost all Q_ws. At an orifice spacing of 38 mm, the efficiency started increasing as Q_w increased, reaching a maximum and decreasing at higher Q_w. This may be due to flow disturbance at higher Q_w as observed in the jet pump.

Fig. 8 gives the effect of orifice spacing S on various parameters when the pump was operated at bep, spacing and of maximum Q_w. This figure was plotted based on the results from Fig. 7. The points of this figure were taken at optimum conditions of four orifice spacings within the operated range. From Fig. 8 it was found that the peak efficiency seems to occur at an orifice spacing of 33 mm. Q_{air} and P_{out} decrease with an increase in orifice spacings and M and N are independent of S for this operative condition. In addition, this figure also shows the value of Q_w when highest efficiency was obtained for that S.

Fig. 9 shows the variation of time and chamber pressure. The curves are obtained for four orifice spacings at the condition indicated in Fig 8. From Fig. 9 it was found that when time increases, the chamber pressure decreases. This experiment was done by closing V1 and opening V2. A minimum chamber pressure with respect to time and the time taken to reach 0.015 bar higher than the minimum pressure were found as shown in Fig. 9 for S = 28 mm.

Fig. 10 shows the minimum chamber pressure created by the jet pump and the time taken to reach 0.015 bar higher than the minimum pressure for four orifice spacings at conditions indicated in Fig. 8. Minimum chamber pressure and corresponding time were found from Fig. 9 and plotted against S. The jet pump gives the maximum efficiency at S = 33mm as shown in Fig. 8; the pumping capability (evacuating capability) of the pump is high (Fig. 9). At S = 33 mm, the minimum chamber pressure of 0.094 bar is obtained at 320 s. Thus, from the above results, it may be concluded that the best orifice spacing is S = 33 mm, since the maximum vacuum, i.e., minimum chamber pressure, is obtained in a reasonably lesser time.

5. Conclusions

A jet pump was designed, manufactured and operated to create a vacuum. Experimental results showed the performance of a jet pump for four orifice spacings from 28 mm to 38 mm, which was used for optimization of the jet pump based on performance and maximum vacuum obtained at optimum conditions.

Among the distances at which tests were done, an orifice spacing of 33 mm was found to give a relatively high efficiency. Pumping capabilities of the optimized jet pump at four orifice spacings were determined and shown. In addition, as per pumping capability, an orifice spacing of 33 mm was found to be efficient. This study to determine optimum orifice spacing facilitates a rational design of two-phase jet pumps for practical applications based on maximum efficiency and pumping capability.

6. Symbols

A	—	Cross sectional area, m^2
H	—	Head of fluid, m
M	—	Flow ratio, Q_{air}/Q_w
N	—	Pressure characteristic
O	—	Orificemeter
P	—	Pressure, bar absolute
Q	—	Volumetric flow rate, lps
R	—	Area ratio, A_o/A_t
S	—	Distance between orifice exit to mixing tube entrance spacing, mm
V1, V2, V3	—	Valves

Subscripts

air	—	Air
i	—	Inlet
out	—	Outlet
w	—	Water

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