Efficiency of liquid-jet high-pressure booster compressors

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Abstract. There are almost no experimental data on the head-capacity curves for liquid-jet compressors with the inlet gas pressure of liquid-jet apparatus more than 1 MPa. Meanwhile, this range is important for many engineering applications in which relatively low compressor ratio is required for the pumping of gas under high pressure. This is mostly the case when gas circulation is to be provided in a closed or almost closed circuit. A head-capacity curve of a liquid-jet apparatus has been estimated experimentally for the air pumping at up to 2.5 MPa by a water jet. To obtain this curve, a new original technique has been submitted and verified which is based on an inverse unsteady problem of gas pumping and allows derivation of the whole curve instead of one operating point, which is the case for conventional methods. The experiments have demonstrated that the relative head of the liquid-jet compressor grows with the apparatus inlet air pressure in the middle part of the curve.

1. Introduction

It is interesting to note that the first setups that we would call water-air ejectors today were invented in 1590 by the Italian scholar Giambattista della Porta. These setups were called trompes. They were used in metallurgy together with Catalonian hearths till the 19th century. The design of liquid-jet apparatuses of the first generation was almost the same as the conventional design of single-phase liquid-jet pumps. Their efficiency was comparatively low, since in short chambers and divergent channels typical of the one-phase apparatuses the two-phase operating process could be hardly arranged. Then the need for complete disintegration of liquid jets into droplets was established [1, 2].

The review by [3] distinguishes three generations of liquid-gas jet apparatuses. The third-generation apparatuses are characterized by the original design of adjustable liquid-jet pumps with agitators of active jet disintegration and formation of the mixing zone.

The main application of liquid-gas jet apparatuses was and still is the pumping of different gases and vapor with subsequent discharge into the atmosphere (typically). The extensive data are available in literature on the characteristics, calculation techniques, application spheres, design and arrangement of processes in vapor-jet apparatuses and vacuum-generating apparatuses [4, 5]. But only scarce information can be found on application of jet apparatuses to high-pressure gas pumping. It was only in 2000 that the US patent [6] was filed, in which the gas in liquid-jet compressor was pressurized up
to the high pressure of at least 7 atm by the liquid under high pressure of at least 16 atm. Some other rare references to pressurized liquid-jet compressors include e.g. [2, 7].

The increase in the apparatus efficiency follows the path of the component mixing enhancement in the apparatus ducts involving the enhanced liquid disintegration, reduction of liquid sedimentation on the duct walls, augmentation of the divergent channel efficiency. The calculation procedures based on the integral laws of mass, momentum and energy conservation yield satisfactory results for uniform media but when there is more than one phase of working media, the uncertainty in estimation of jet apparatus parameters becomes unacceptable. For the case of liquid-gas apparatuses, the calculation approaches are based on correlations derived from the empirical data generalization. Individual attempts at numerical simulation of processes occurring in jet apparatuses have been made recently [8, 9]. In the operating modes of high-pressure liquid-jet compressors that are of practical interest the volume fractions of liquid and gas of the two-phase mixture are close to each other. Such a phase correlation is the most complicated for analytical and numerical calculations of two-phase flows, because many simplifying assumptions are not valid under comparable fractions of liquid and gas. Due to diversity and complexity of theoretical description of even the simplest processes in the liquid-jet compressor ducts, the experimental research is the primary source of information in this sphere.

There is almost no experimental data for the liquid-jet compressors operating under the gas pressure at the apparatus inlet higher than the atmospheric pressure, especially when the pressure is higher than 1 MPa. Meanwhile, this range is important for many engineering applications in which relatively low compressor pressure ratio (5-15% higher than the inlet pressure) is required for the pumping of gas under high pressure. This is mostly the case in multiple power engineering, mechanical engineering and chemical engineering applications, where gas circulation is to be provided in a closed or almost closed circuit.

The paper elaborates on the experimental estimation of the head-capacity curve of a jet apparatus pumping the air under the inlet pressure of 2.5 MPa by a water jet. To obtain a head-capacity curve of the liquid-jet compressor, a new original technique has been submitted and verified which is based on an inverse unsteady problem of gas pumping and allows derivation of the whole curve instead of one operating point which is the case for conventional methods.

2. Experimental setup and procedure
The experiments were performed on the setup (figure 1) operating on pressurized air. Figure 1 shows the jet apparatus geometry as well. In the apparatus, water was supplied through the central nozzle with the outlet diameter $d=10$ mm. The air was supplied to the peripheral part of the cylindrical

![Figure 1. Experimental setup for liquid-jet compressor operation research](image-url)
chamber with the diameter of 50 mm, which was attached to the cylindrical mixing chamber (diameter $D=20$ mm, length $L=200$ mm) through a conical insert. There was a divergent channel (with the length $l=75$ mm) mounted at the outlet. Thus, the liquid-jet apparatus with the relative diameter $D/d=2$ and relative length of the mixing chamber $L/D=10$ was considered.

The main components of the setup are three tanks (1-3 in figure 1) and the jet apparatus 4 designed for high pressure operation. The first tank 1 contains the operating medium (water) for the liquid-jet compressor. The second tank 2 contains the gas (air) whose pressure is to be increased. The third tank 3 is intended for water-air mixture collection. In other words, the second tank is the low-pressure tank, the third one is the high-pressure tank (in terms of air pressure). As for functionality, the first tank supplies the high-head water jet to the ejector, low-pressure air is injected from the second tank, and the mixed flow is supplied to the third tank. Valves were installed between the tanks. The tanks and ejector were equipped with pressure measurement instruments. The experimental setup was connected to the high-pressure cylinder manifold.

Each tank had the volume of approximately 0.1 m$^3$, which was accurately measured before the experiments. Prior to the experiments the first tank was filled with certain amount of water (about 0.01 m$^3$). Then the air from the high-pressure cylinder manifold was pumped into the second and third tanks up to the operating pressure of 1 MPa to 2.5 MPa. The air pressure in the first tank should exceed the pressure in the other two tanks by approximately 1 MPa. Such positive pressure of air provides the required water flow rate and high quality of its disintegration in the jet apparatus duct during the experiment. After the required pressure has stabilized, the valve between the cylinder manifold and the experimental setup is closed. The setup is ready for operation.

A new original technique for liquid-jet compressor characteristics estimation has been developed which is based on an inverse unsteady problem of air pumping and allows derivation of the whole curve from a single experiment unlike the conventional plotting of the curve using the “points” obtained in steady regimes under different boundary conditions. Operation of the liquid-jet compressor during a single experiment was unsteady: the water pressure at the ejector inlet, $P_1$, was decreasing and the counterpressure, $P_3$, at the ejector outlet was growing. This was accompanied by the corresponding decrease in the water flow rate. The gas line exhibited similar process, i.e. the decrease in pressure $P_2$ in the pressurized air receiver and the pressure growth in the receiver $P_3$.

Typical variation of pressure (primary experimental data) with time, $\tau$, in three tanks is shown in figure 2.

In the subsequent analysis the corrected specific gas flow rate is used. To estimate the latter, the flow rate of both the gas and liquid are required. Unfortunately, there are no flow meters capable of wide range metering, the more so in conditions of low flow resistance in a duct. The water and air flow rates were estimated indirectly from the dynamics of pressure in the tanks.

![Figure 2. Typical time variation of pressure in the experiments](image-url)
The water flow rate technique was essentially the following. Water was forced from the tank by pressurized air whose mass above the water was kept constant throughout the experiment until the water was completely forced out. The air was cooled due to expansion, and its temperature became lower than the tank wall temperature, with which the former was in thermodynamic equilibrium before the expansion started. Thus, the proper conditions for heat supply to the air emerged. However, in conditions of fast water displacement the heat supplied due to heat transfer appears to be three orders less than expansion work performed by gas, so the expansion process can be considered adiabatic with high accuracy. Pressure buildup that accompanies the air heating after the valves are closed also attests to almost adiabatic expansion of air. Under adiabatic air expansion the pressure-volume correlation is well-known: \( PV^k = \text{const} \), where \( k \) is the heat capacity ratio.

This relation allows estimation of the gas volume change from the pressure variation at a given initial gas volume

\[
V_1(\tau) = V_{10} \left( \frac{P_{10}}{P_1(\tau)} \right)^{1/k}
\]

where \( V_{10} \), \( P_{10} \) are the gas volume in the first tank and its pressure at the initial time, respectively.

The flow rate of displaced liquid is obviously determined by the rate of gas volume variation

\[
Q_W(\tau) = \frac{dV_1(\tau)}{d\tau}.
\]

Air flow rate was estimated in a similar way from the air expansion (pressure drop) in tank 2. Due to negligible effect of heat transfer if compared with expansion work, the air expansion was considered adiabatic. In this case, knowing the volume of tank 2 with the attached pipelines that are under the same pressure, the pressure change relative to the initial pressure before the air supply valve is opened yields the actual air volume, and the rate of change of this volume yields the actual volume flow rate of air under the actual pressure and temperature of air supplied to the liquid-jet apparatus. This flow rate is usually considered when building head-capacity curves. The following relations were used:

\[
V_2(\tau) = V_{20} \left( \frac{P_{20}}{P_2(\tau)} \right)^{1/k}, \quad Q_G(\tau) = \frac{dV_2(\tau)}{d\tau}.
\]

where \( V_{20} \), \( P_{20} \) are the gas volume and pressure in the second tank at the initial time, respectively; \( Q_G \) is the gas (air) flow rate.

When processing the experimental data, we are interested in the process starting from the opening of valves that supply components and up to the moment when the pressure \( P_2 \) attains its minimal value (figure 2). When the pressure \( P_3 \) ceases to drop, it means that the ejected air flow rate is zero. After this, the pressure \( P_3 \) keeps growing due to air compression resulting from the reduction in its volume when water is supplied to the tank 3; air can even flow from the tank 3 back to the tank 2 (negative flow rate) if the pressure drop exceeds the one corresponding to the zero flow rate.

3. Results and discussion

Water and air flow rates estimated from the oscillograms (figure 2) are shown in figure 3. The decrease in water flow rate in time agrees well with the dynamics of water supply line pressure drop. Besides, the water flow rate integrated over the process time with the accuracy not exceeding 4% agreed well with the amount of water weighed before the filling.

Figures 2 and 3 show that the duration of informative part of unsteady process is about 2 seconds. This time is short enough to neglect heat transfer between the working medium and the construction elements. But this time is long enough if compared with the residence time of two-phase mixture in the jet apparatus duct. Indeed, when the water injection rate is more than 30 m/s and two-phase
mixture velocity is about 20 m/s, the travel time through the 0.3-meter long duct is approximately 0.01 s. For this reason, under the velocity implemented in experiments, the internal processes in the jet apparatus ducts can be considered being under quasi-steady conditions.

![Figure 3](image-url)  
**Figure 3.** Dynamics of water, $Q_W$, and air, $Q_G$, flow rates in experiment

The head-capacity curve was estimated from the obtained data in the most universal normalized form $P(Q)$ (figure 4). Here $P = \Delta P_G / \Delta P_W$, $Q = Q_G / Q_W$, where $\Delta P_G = P_3 - P_2$ is the air pressure buildup (difference between the section upstream of the compressor and downstream of the one), $\Delta P_W = P_1 - P_2$ is the excess in water discharge pressure over the air suction pressure.

One of the experimentally obtained head-capacity curves is shown in figure 4 together with the curve calculated according to the technique [10]. It is clear that while the curves agree qualitatively, they are significantly different quantitatively, particularly in the middle (in terms of the flow rate) part of the curve, which is the range that is most often used in engineering applications. It should be noted that calculation technique [10] is based on the experimental data obtained under the jet apparatus inlet pressure close to the atmospheric pressure. Another possible affecting factor could be the divergent outlet from the ejector present in our case while the technique was developed for the apparatus without the divergent outlet. The divergent section obviously improves the apparatus performance but its individual effect has not been considered in the present paper.

Thus, higher relative head in the liquid-jet compressor was obtained in almost the whole range of relative gas flow rates considered in the experiments with high-pressure air testifying to the improved performance of the compressor due to the increased gas pressure.

![Figure 4](image-url)  
**Figure 4.** Head-capacity curve of the liquid-jet booster compressor: 1 — empirical data; 2 — calculated according to [10]
4. Conclusions
A new original technique has been submitted and verified which is based on an inverse unsteady problem of gas pumping and allows derivation of the whole curve at once instead of one operating point, which is the case for conventional methods.

The experiments have demonstrated that the relative head of the liquid-jet compressor grows with the apparatus inlet air pressure in the middle (most interesting in engineering practice) part of the curve which probably results from the enhanced water disintegration due to its contact with more dense high-pressure air.

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References
[1] Witte J H 1969 J Fluid Mech 36 639-55
[2] Cunningham R G and Dopkin R J 1974 J Fluids Eng 216-26
[3] Spiridonov E K 2005 Vestnik YuUrGU. Mashinostroenie series 1 94-104 (in Russian)
[4] Chunnanond K and Aphornratana S 2004 Appl Therm Eng 24 311-22
[5] Butterworth M D and Sheer TJ 2007 Appl Therm Eng 27 2145-52
[6] Leverett G F 2000 U.S. Patent No. 6019820 (Washington, DC: U.S. Patent and Trademark Office)
[7] Cramers P H M R and Beenackers A A C M 2001 Chem Eng J 82 131-41
[8] Beithou N and Aybar H S 2000 Int J Multiphase Flow 26 1609-19
[9] Subramanian G, Natarajan S K, Adhimoulame K and Natarajan A 2014 Int J Therm Sci 84 134-42
[10] Lyamaev B F 1988 Liquid-Jet Pumps and Facilities (Leningrad: Mashinostroenie) (in Russian)