INTRODUCTION

The ATLAS superconducting toroidal magnets, the two End Cap Toroids and Barrel Toroid, are cooled by means of forced flow subcooled liquid helium (LHe) [1], pushed through aluminum tubes attached to the magnet coil casings. Two pumps are installed, each with 1200 g/s nominal mass flow. Preliminary results were published on the qualification tests and a detailed description of the test facility, built on purpose was given [2].

THE 1200 G/S CENTRIFUGAL PUMPS

The 1200 g/s test facility is described in detail elsewhere [2]. A LHe reservoir is connected to a 1.2 kW @ 4.4 K refrigerator, which keeps the level constant. The LHe enters by gravity the inlet of the 1200 g/s pump: the pumped LHe returns to the reservoir after having passed a Coriolis and a Venturi mass flowmeter. The pressure head produced by the pump is measured with a precise differential pressure transducer across the pump itself. The pump is mounted vertically with the motor, outside the cryostat, cooled by natural convection by the air. The rotational speed of the pump is measured by means of a three-teeth proximity sensor installed on the shaft of the pump, since the values are different from the frequency of the Variable Frequency Drive (VFD) connected to the motor.

THE PROTOTYPE PUMP

A prototype pump was developed in cooperation with the supplier and investigated for qualification.

![Fig. 1. Head coefficient as a function of flow coefficient for the prototype pump.](image)

The pump is of the full emission type. Two different impellers, both with 16 curved blades, were tested: the first with a diameter of 112.55 mm and straight blades on the back of the impeller; the second with a diameter of 105.1 mm and curved back blades. Attached to the impeller is a removable inducer. The second impeller is developed to prevent the moisture condensation that appears on the top flange of the running pump. In fact the inconvenience is not arising from the impeller geometry [3] but was solved by inserting three Teflon spacers between the impeller and the top flange of each pump.

The pump electrical motor is a 2 pole 220 V 3-phase. The rotational speeds of the impellers are in the range 71 Hz (4260 rpm) to 80 Hz (4800 rpm). The characteristic curves have been obtained, at a given speed, by throttling the control valve installed downstream prior to the inlet of the LHe reservoir, while the pressure is measured directly across the inlet and the outlet of the pump. For a given impeller diameter the pressure head is higher for a higher rotational speed. Conversely, for a given rotational speed, the pressure head is higher for larger diameter (affinity laws). The impeller with the diameter of 112.55 mm meets the requirements of 400 mbar (40 kPa) at 1200 g/s at 76 Hz. The same requirements are met by the impeller with a smaller diameter at 80 Hz. Data can be compared as in Figure 1, by using the non-dimensional head coefficient, KH and the flow coefficient, KQ, as follows:

\[ KH = \frac{gH}{\nu^2D^2}; KQ = \frac{Q}{\nu D^3} \]  

where \( g = 9.81 \text{ m/s}^2 \), head \( H \) (m), frequency \( \nu \) (Hz), diameter \( D \) (m), volumetric flow \( Q \) (m\(^3\)/s).

![Fig. 2. Slip factor versus mass flow for the 1200 g/s prototype pump.](image)

The reduced non-dimensional parameters are particularly useful to predict the performance of a pump in various conditions, e.g. in order to predict the pressure head by changing the impeller diameter and/or the pump speed.
Some authors use the head coefficient $\Psi$ and the flow coefficient $\Phi$, where $KH=\pi^2\Psi$ and $4*KQ=\pi^2\Phi$.

The efficiency $\eta$ for different mass flow has also been computed. The amount of vapour, returning from the reservoir of the test station towards the refrigerator is measured by a Venturi mass flowmeter, calibrated in situ with a precise resistor immersed in the LHe bath of the reservoir. The static heat load of the test facility was subtracted. The latter was measured when a mass flow in the pump circuit was induced by a calibrated resistor, immersed in the LHe, inducing a thermosyphon flow of the LHe. The efficiency $\eta$ is calculated as follows:

$$\eta = \frac{\dot{m}}{W} \int \frac{dP}{\rho} = \frac{\dot{m}}{W} \frac{\Delta P}{\rho} \quad (2),$$

where $\dot{m}$ is the LHe mass flow [kg/s], $\Delta P=P_2-P_1$ is the pressure head across the pump [Pa], $\rho$ is the density of LHe [kg/m$^3$], $W$ [Watt] is the net heat load at the test station.

![Figure 3](image1.png)

Fig. 3. The characteristic curves of the three final 1200 g/s pumps are summarized.

The efficiency is the ratio between the hydraulic power and the cooling capacity required by the refrigerator for a given mass flow and pressure head. At nominal mass flow the total efficiency is 60 %, increasing with increasing mass flow. The frequency values used for the non-dimensional data are the ones measured on the shaft. The shaft frequency is smaller than the Variable Frequency Drive value, and it depends on the load on the shaft, i.e. on the LHe mass flow and head. The difference of the two frequencies, known as a “slip factor” is shown in Figure 1. As mentioned in the description of the test station, the liquid helium enters by gravity the suction port of the pump so the pump can undergo cavitation if the hydrostatic pressure is lower than the Net Positive Suction Head (NPSH). Two methods have been employed to determine the NPSH value. At first, the feeding line of the reservoir from the refrigerator is closed, leaving the vapour return connected. In this condition, with the pump running, the LHe level decreases in such a way that, when the hydrostatic pressure reaches the NPSH value, the pressure head and the mass flow collapse. With the uncertainty in the two measurements of the LHe level, obtained by a differential pressure and a superconducting wire, the value of NPSH is estimated between 14 mbar (1.4 kPa) and 16 mbar (1.6 kPa).

A second method to estimate the NPSH is applied. The pump is running at 72 Hz and 1200 g/s with a constant LHe level in the reservoir. The throttling valve is slowly opened until the pressure head and the mass flow collapse. This point allows calculating the cavitation coefficient sigma as follows:

$$\sigma = \frac{P_1-P_c}{\frac{1}{2} \rho v^2} \quad (3),$$

where $P_1$ is the upstream static pressure, $P_c$ is the critical pressure at which cavitation occurs, and $v$ is the mean upstream fluid velocity. From the experiment $\sigma$ is 5.0 which in turn corresponds to a NPSH of 20 mbar at 1200 g/s. As a result it can be stated that the NPSH at 1200 g/s is below (or equal to) 20 mbar (2.0 kPa).

THREE FINAL 1200 G/S CENTRIFUGAL PUMPS

After the qualification tests the final configuration was defined and three more pumps were purchased and individually tested at the station before their installation in the ATLAS cavern. For each pump the characteristic curves are measured and the efficiency calculated as a function of the mass flow. The characteristic curves are shown in Figure 3, and their efficiencies in Figure 4. It can be clearly seen that pump #1 and pump #2 have similar performances. Pump #3 has shown a slightly lower efficiency between 700 and 1000 g/s.

![Figure 4](image2.png)

Fig. 4. The calculated efficiency of the three final 1200 g/s pumps.