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On Leakage Flows in a Liquid Hydrogen Multistage Pump for Aircraft Engine Applications

A comprehensive operational characterization of a representative, liquid hydrogen (LH₂) aircraft engine pump, a key enabler for future hydrogen aviation, is presented in this work. The implications of leakage flows are investigated in a two-stage, high-pressure pump for a wide range of flow rates and rotational speeds, through three-dimensional (3D) (unsteady) Reynolds-averaged Navier–Stokes simulations. The study compares two configurations: a baseline model comprising the primary flow path components—inducers, impellers, and volutes, and a realizable pump hardware that includes hub, shroud, and power unit cavities. Performance metrics, including head changes and efficiencies, are extracted both at a component and system level. Leakage flow rates of 27.6% and up to 92.9% of the overall pump flow rate are recorded at design and lowest flow points, respectively. The head loss in the mid to low flow rates does not exceed 4.5%, but the efficiency diminishes by up to 13.5% at off-design operation. The component analysis indicates significant penalties in impeller efficiency. At high flow rates, the presence of leakage flows improves the overall pump performance by 43% and 27% in head rise and efficiency, due to reduced losses in volutes and connecting ducts. The detailed characterization of pump behavior described in this work is of importance in development of safe, reliable, and predictable design of aircraft LH₂ pumps. These aircraft pumps are different from LH₂ pumps utilized in rocketry and for cooling in nuclear industry due to the requirement to operate with wider turn-down ratios and often, at low specific speeds. Therefore, this study addresses design considerations in this enabling technology that ensures the delivery of preconditioned fuel according to the aircraft operating conditions. [DOI: 10.1115/1.4066827]

Keywords: pump, aircraft, hydrogen, leakage flow, head, efficiency

1 Introduction

The use of sustainable and environmentally friendly fuels in commercial civil turbofan engines is at the epicenter of current research efforts, due to their potential to decarbonize the aerospace industry and achieve aviation climate neutrality in the march toward net zero goals [1,2]. The implementation of zero-carbon fuels, particularly hydrogen, in commercial aircraft and rotorcraft applications (see Ref. [3]), is again at the forefront of future green technologies in aviation industry, after early explorations in the 1970s. Hydrogen-powered aircraft can be used for maintaining existing airline routes, covering different ranges, and can act in tandem with associated aircraft electrification technologies, such as fuel cells. A comprehensive discussion on hydrogen-powered aircraft including its technical and economic challenges along

with key observations regarding the availability of low-cost green hydrogen production at scale is addressed in a recent publication [4].

Due to the low density of liquid hydrogen (LH₂), cryogenic storage is crucial, as it maximizes its energy density and facilitates its effective harnessing, a key requirement to achieve high overall aircraft and turbofan engine efficiencies [5–8]. However, cryogenic storage of LH₂ poses challenges in the design of fuel tanks and in the fuel delivery system. Several studies and strategies have been identified to optimize fuel tanks, aiming to minimize boil-off and increase gravimetric efficiency. Moreover, the challenges associated with the preconditioning of the fuel from cryogenic conditions to high pressure, near ambient temperature conditions, have not been investigated in as much detail, except for some system level architectural studies [9,10]. Preconditioning of the fuel is critical to enable well-mixed, low, or nearly no NO_x, stable combustion in the engine. The on-board availability of cryogenic hydrogen can be beneficial from a thermal management point of view, since it can act as a thermal sink. This provides opportunities for rejecting heat from the engine and other auxiliary systems to achieve the required

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temperature rise of hydrogen, prior to its injection into the combustor. Few studies have focused on the design of heat exchangers to enable this temperature rise and to explore its synergies with engine cycle design optimization [11,12]. However, the aspect of pressure rise in hydrogen pumps for aircraft applications has not been investigated in sufficient detail. In engine cycle design exploration and system level architectural studies, a general pump behavior characteristic is often assumed to enable simulations.

In LH₂ fuel system architectures, typically two different pumps are used: a booster pump at the tank outlet to move the liquid and overcome the pressure drop and a high-pressure pump to deliver the fuel at the required combustor pressure. Of these two, the high-pressure pump is critical to ensure appropriately conditioned LH₂ delivery for a wide variety of operating points in the aircraft mission profile. For high-pressure pumps, typically centrifugal designs are preferred due to their ability to achieve higher pressure rise in smaller number of stages, resulting in lower packaging volumes and weights as compared to alternative pumping solutions.

There is a wide body of literature around the design, analysis, and operation of LH₂ pumps for applications in liquid propellant rockets, as well as in the nuclear industry for cooling purposes. However, the operating regime of these pumps is restricted to a small range of mass flow and power settings and are typically operated in nearly design conditions during their entire utilization profile. On the other hand, pumps for aircraft engine applications are required to operate for considerable durations of time in a significantly wider range of mass flows and power settings, from maximum engine fuel demand at high power takeoff, to low power settings during idling, descent, and landing. Moreover, aircraft engine LH₂ pumps often tend to be low-specific speed designs, powered by electric motors due to their smaller absolute LH₂ flow rates. To help increase the specific speed of the pumps, the pressurization is achieved between multiple units. The flow-field description and operational characterization of such representative low-specific speed LH₂ pumps for different flow rates and rotational speeds has not been explored before. Typically, in the characterization of pump operation, a baseline geometry of the inducers, impeller wheels, and volutes are used without any leakage paths. However, it is known that regardless of the fluid, leakages in impeller shrouds, hubs, and power unit assemblies substantially affect the pump performance characteristics [13,14]. These are the characteristics that the actual pump or compressor hardware will eventually exhibit; thus, it is important to consider the implications of leakage flows for design, modeling, and performance/operability assessment purposes. Consequently, the pump leakage behavior, flow-field structures, and its corresponding effect on the operational characteristics constitute a critical, yet insufficiently explored field, which is inextricably linked to the successful design of future hydrogen fuel systems.

To address the research gap, this study discusses a comprehensive computational flow-field investigation into the unexplored operational characteristics of aircraft representative LH₂, high-pressure fuel pumps using two pump configurations: a baseline design with only the main turbomachinery elements and a hardware-realizable design, with the full inclusion of all possible leakage paths in the hub, shroud, and the electric motor assembly. The work represents a first of its kind for LH₂ pumps which describes a multilevel characterization into the effects of different leakages and their impact on the performance of a realistic aircraft LH₂ pump design. The observations made in this work are of importance to the design and development of this key enabling technology as well as its safe, reliable, and predictable operation which is critical to future hydrogen aviation. Moreover, the general form of the operational characteristics with and without leakages discussed in this work is essential for lower order modeling methods exploring fuel system architecture and engine cycle innovations, which has hitherto relied on general assumptions for LH₂ pumping behavior.

2 Hydrogen Pump Geometry and Modeling Approach

The pump under consideration is representative of a two-stage, high-pressure hydrogen pump, intended for application in future,

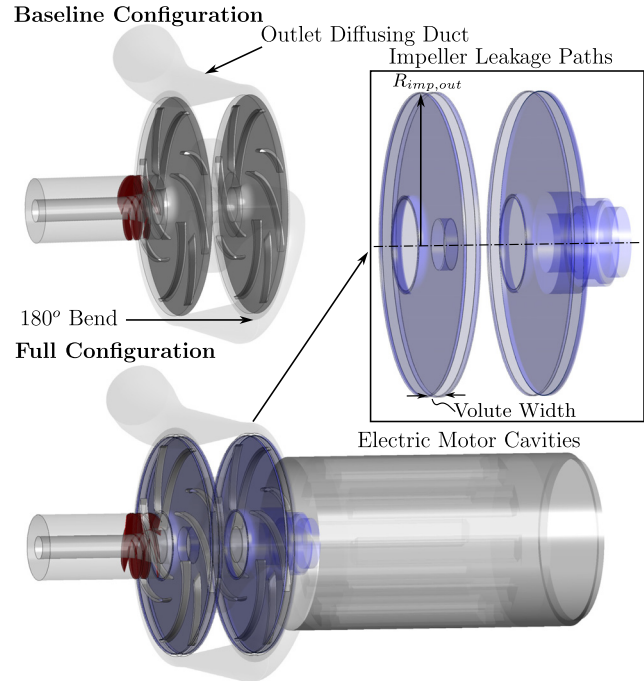


Fig. 1 Overview of the baseline (top) and full configuration (bottom) pump CFD models

hydrogen-powered aircraft engines. Both stages comprise identical radial impellers and volutes, with an inducer preceding the first stage. The inclusion of the inducer enables operation at low inlet pressures, reliable operation when hydrogen vapor forms at the inlet of the pump and provides additional head rise. Intermediate ducting components guiding the flow through the different stages are included, while the pump shaft is considered to be coupled with an electric motor. To assess the impact of leakage flows on pump performance, two configurations are compared using computational fluid dynamics (CFD) methods; the baseline, “clean” configuration and the full model. The first comprises the turbomachinery and ducting components, but omits hub and shroud leakage flows as well as all cavities within the motor, where potentially hydrogen leakages are expected. The two configurations are shown in Fig. 1.

2.1 Pump Design. The inducer consists of three flat plate blades with rounded leading edge (LE) and trailing edge (TE). The impellers are shrouded and consist of four full and four splitter blades with rounded LE and flat, cut-off TE. The impeller and volute designs are identical and are connected downstream with identical diffusing ducts. The two stages are connected via a 180 deg bend, guiding the flow to the inlet of the second stage as it exits the diffusing duct of the first stage. Leakage paths are formed in the gap between the shroud and the casing as well as in the gap between the hub and the casing. The hub leakage path of the first stage impeller is obstructed by the presence of mechanical components, thus is not linked to the second stage flow path, while the hub leakage path of the second stage is connected with the internal cavity network of the motor. The cavities inside the rotor are modeled in detail and are representative of the cavities between the rotating components, the casing and the shaft. Key design parameters of the turbomachinery components and the volutes normalized with the impeller outlet radius are shown in Tables 1 and 2.

2.2 Performance Metrics. The performance of the different components and the entire pump are estimated using pitchwise-averaged flow variables at planes positioned upstream and downstream of each component. To account for changes in density due to compressibility phenomena, the average density along the

Table 1 Nondimensional turbomachinery design parameters

Design parameter	Inducer	Impeller
Number of blades	3	4 + 4
Inlet blade height/ $R_{imp,out}$	0.2	0.14
Outlet blade height/ $R_{imp,out}$	0.14	0.066
$R_{in}/R_{imp,out}$	0.87	0.24
Tip clearance/ $R_{imp,out}$	0.0033	(Shrouded)

Table 2 Nondimensional volute design parameters

Design parameter	Volute
Inlet radius/ $R_{imp,out}$	1.05
Outlet radius/ $R_{imp,out}$	1.11
Width/ $R_{imp,out}$	0.07

component under consideration is preferred. The head rise and hydraulic efficiency are computed according to Eqs. (1) and (2), and the shaft power is obtained by integrating the product of the torque and angular velocity along the turbomachinery components

$$\Delta H = \frac{\Delta P_0}{g\rho_{avg}} \quad (1)$$

$$\eta_p = \frac{Q\Delta H g\rho_{avg}}{P_{sh}} \quad (2)$$

$$P_{sh} = \int_{rotors} F_{\theta}\omega r = \int_{rotors} \tau\omega \quad (3)$$

In the above equations, the volumetric flow rate is computed at the component inlet using the average density, i.e., $Q = \dot{m}_{in}/\rho_{avg}$ with the average density being the arithmetic average of the inlet and outlet density of the component under consideration, i.e., $\rho_{avg} = 0.5(\rho_{in} + \rho_{out})$.

2.3 Grid and Boundary Conditions. The grid topology comprises a combination of structured (hexahedra) and unstructured (tetrahedra) grids. The grids of all turbomachinery components are structured (ANSYS TURBOGRID), while the grids of the volutes, additional ducting components, leakage paths, and motor cavities are unstructured (ANSYS MESHER). The inducer and impeller grids

include a total of approximately 3.39×10^6 and 4.74×10^6 number of elements, respectively, the volutes along with the adjacent ducting components 5.83×10^6 , and the hub, shroud, and motor cavities incorporate a total count of about 60.1×10^6 elements. Inflation layers are used in all turbomachinery components but not in ducted components to preserve computational requirements within reasonable levels. This is not expected to substantially affect the performance quantities of interest as the inertia forces are primarily driving the flow-field due to the excessively low viscosity of hydrogen. An overview of the mesh on the meridional plane along with the mesh of individual components is shown in Fig. 2. The total element count is approximately 18.7×10^6 elements for the clean and approximately 78.8×10^6 elements for the full configuration.

At the pump inlet, denoted with number “0,” uniform stagnation pressure and temperature are applied, while at the outlet (plane “5”), mass flow is imposed. The working fluid is liquid hydrogen. The compressibility factor along the pump is expected to range by approximately 0.28 between the inlet and the outlet; thus, real gas modeling is required. The standard, real gas models embedded in CFX were deemed inadequate for the operating range under consideration and proved to be relatively unstable in certain operating conditions. This was addressed by introducing real gas property (RGP) tables containing both liquid and gaseous hydrogen properties from the National Institute of Standards and Technology database. The $k - \omega$ turbulence model is used, and the energy equation is also resolved. Frozen rotor interfaces are set between rotating and stationary domains, since mixing planes introduced significant numerical instabilities. All walls are adiabatic with no-slip condition. All rotors are driven by the same shaft, thereby having similar angular velocities. Steady-state, single phase, Reynolds-averaged Navier–Stokes simulations are carried out using with higher order schemes.

2.4 Numerical Uncertainty. To evaluate the metrics of interest, it is essential to quantify the uncertainty associated with the mesh as well as the unsteadiness of the steady-state flow solutions on which the characterization of the design and off-design pump performance is based upon. The pump performance metrics are time averaged (in steady simulations they are iteration-averaged) to extract the average performance metrics. The average fluctuations are then determined by averaging the difference between the mean and the instantaneous/recorded value of the property, over a range of iterations, typically 10% of the iterations before the simulation was stopped. Denoting with Λ an arbitrary quantity and replacing time with iterations for steady-state simulations, the mean of the quantity

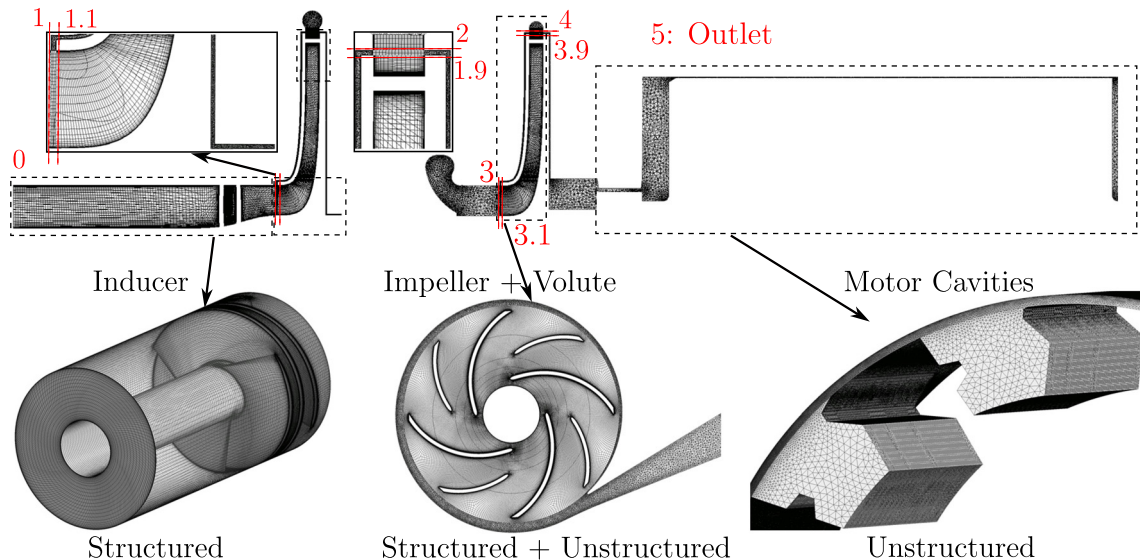


Fig. 2 Overview of the grid on the meridional plane (top) and of individual components (bottom). A combination of structured and unstructured grids is used.

Table 3 Mass flow, head rise, hub, and shroud leakage path fluctuations at design speed (bold) and different flow rates as percentage of the averaged values

Q_{norm}	$\dot{m}'_{\text{out}} \%$	$\Delta H' \%$	$\dot{m}_{\text{h,out}}^{\text{LP}} \%$	$\dot{m}_{\text{s,out}}^{\text{LP}} \%$
0.18	1.72	6.05	1059.78	4.36
0.22	1.05	5.14	282.23	4.47
0.27	0.58	4.57	353.74	2.87
0.36	0.45	2.87	112.58	1.25
0.45	0.21	5.49	114.81	1.73
0.48	0.28	1.98	80.22	1.00
0.54	0.23	2.62	95.63	1.24
0.64	0.12	1.50	244.75	1.53
0.77	0.21	4.38	79.25	1.02
0.90	0.17	4.81	87.83	1.57
1.00	0.13	7.45	97.21	1.31

and the average fluctuations are computed by the following equations:

$$\bar{\Lambda} = \frac{\int_{t_1}^{t_2} \Lambda dt}{\Delta t} \quad (4)$$

$$\overline{\Lambda'} = \frac{\int_{t_1}^{t_2} (\bar{\Lambda} - \Lambda) dt}{\Delta t} \quad (5)$$

Table 3 summarizes the mean fluctuations, as percentage of the corresponding averaged metrics. The fluctuations of the mass flow at the outlet, overall head rise, and leakage mass flows are shown at different operating points at design speed. The mass flow fluctuations are small, not exceeding 0.56%. The average amplitude of the head rise oscillations is larger, reaching its maximum at the highest and lowest mass flow off design points (DPs). The excessive oscillations observed in the hub leakage paths are in absolute numbers negligible, as the paths fill with hydrogen and the incoming or exiting mass flow is almost zero. The shroud leakage mass flow is fluctuating by less than 1.73% across most operating points and is increasing to less than 4.5% at the lowest mass flow conditions. The design point values are shown in bold letters.

A grid convergence study has been carried out at design conditions for each component individually, to estimate the numerical uncertainty associated with the mesh. The head changes

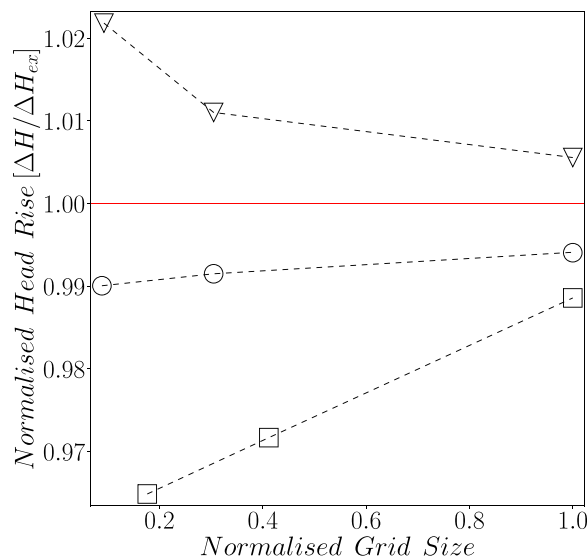


Fig. 3 Overview of isolated grid convergence studies for impellers and volutes carried out at design conditions. The metrics are normalized with the corresponding extrapolated values.

across the inducer, the impeller, and the volute with the intermediate connecting duct are normalized with the corresponding extrapolated values, using Richardson extrapolation, and are plotted in Fig. 3. Since the second impeller and volutes are identical and only the downstream diffusing duct differs (meshed following identical approach), a separate grid sensitivity analysis has not been carried out for the second stage volute and diffusing duct. In the case of the first stage volute/ducts, the displayed property is essentially the head loss. The errors associated with the inducer and impeller domains, discretized with the medium grids, are less than 1.01%, while the head loss error of the volute and downstream duct is within a reasonable 2.82%. The minor accuracy benefits of the more refined grids were not deemed sufficient to justify the additional computational cost; thus, the medium grids were selected for all components.

3 Pump Performance Analysis

The implications of the leakage flows and their origins are evaluated by comparing the baseline and full configuration following a threefold approach: comparison of the full pump performance, assessment of individual component metrics, and examination of the prevalent flow-field features. The analysis is initially focused on the design point; subsequently, low and high mass flow off-design conditions are investigated, and finally, the full characteristics at different operating conditions and rotational speeds are presented, aiming to generalize the observations made in the analysis of the isolated operating conditions.

3.1 Design Point Operation. The design point corresponds to the normalized volumetric flow rate of 0.64 (see Table 3). The implications of the leakage flows on the pump performance are quantified both on a component as well as on a system basis, by determining the efficiency and the head variations extracted from the baseline and full configuration. Planes (0–5) are taken at the pump inlet (0), inducer outlet (1), first stage impeller outlet (2), second stage impeller inlet (3), second stage impeller outlet (4), and pump outlet (5). For the full configuration analysis, additional planes are defined exactly at the inlet and outlet of the impellers, coinciding with the streamwise position of the boundaries of the shroud leakage path rotating walls (see Fig. 2). These enable the quantification of properties transferred by the leakage flow and will be referred to hereafter as planes 1.1, 1.9, 3.1, and 3.9.

The density and the mass flow measured at the aforementioned planes and design conditions for the full configuration are shown in Fig. 4 and are normalized with the corresponding inlet values. The

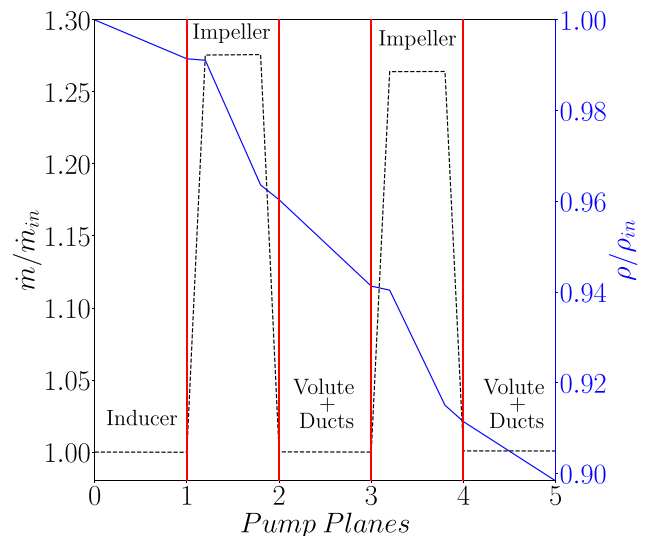


Fig. 4 Normalized density and mass flow along the full pump model measured at different planes including planes upstream and downstream the shroud leakage paths

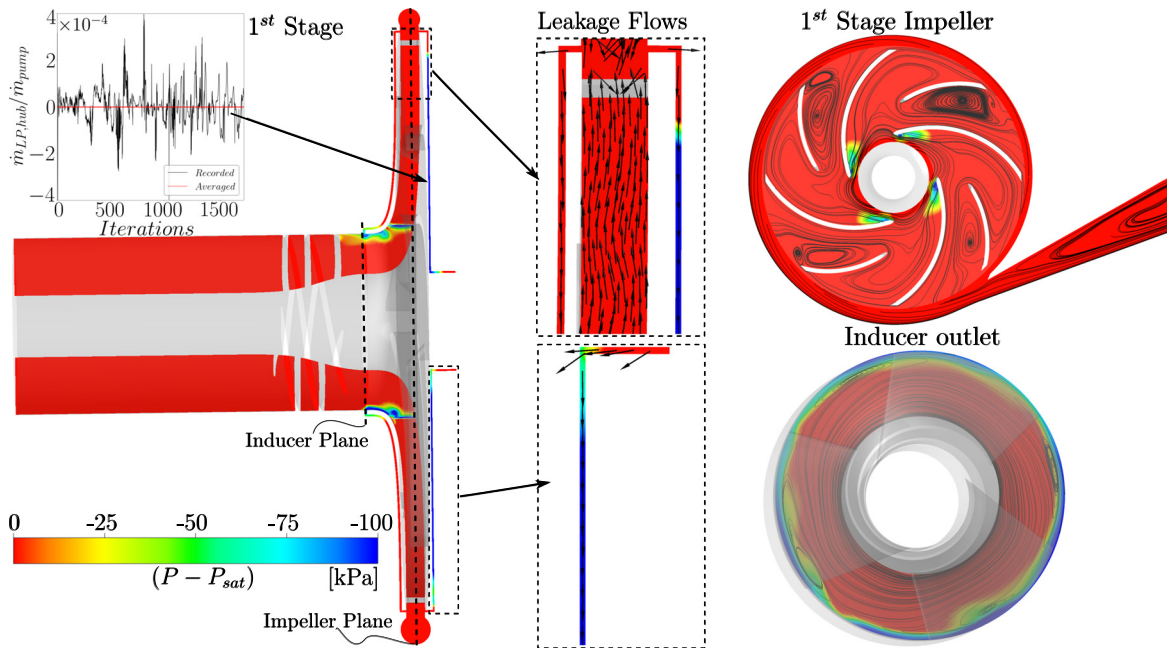


Fig. 5 Difference between local and saturation pressure along first stage impeller indicating regions with vapor and liquid properties

measured mass flow rate at the inlet of the impellers indicates that a significant amount of pressurized hydrogen exits the impellers, subsequently enters the shroud leakage cavity, traveling backwards all the way to the inlet of the impellers where it mixes with the incoming main flow (see leakage contours in Fig. 5). The mass flow through the first and second stage shroud leakage paths reaches 27.6% and 26.5% of the total pump mass flow, forcing the impellers to operate at a substantially higher mass flow. This entails design implications, particularly if the pumped substance is compressible, e.g., gaseous hydrogen, since choking could occur at a lower overall flow rate. While this is not a concern in the current configuration, the higher flow rate in the impeller shifts the operating point toward the high flow region, where the pump efficiency is lower. A redesign of the impeller, to account for the increased mass flow through the impellers, is expected to alleviate the efficiency penalties by adjusting the optimum operation of the impeller for higher flow rates.

Despite the pressurization of the fluid, the density drops by approximately 10.2%; this is related to the compressibility of hydrogen as well as the local flow-field and the RGP tables. The pump is operating close to saturation conditions, particularly at the inlet, thus being prone to phase changes. Even though phase changes are not explicitly modeled (single phase), the RGP tables contain both liquid and gas data and will adjust the fluid properties locally based on the state of the flow. This is captured in the contours of Fig. 5, where the difference between the local and saturation pressure (approximated by a fourth-order polynomial) is plotted across the first stage flow path. Values below zero indicate properties within the two-phase regime. These form in the high-acceleration regions; at the impeller LE and in the leakage paths. The flow through the shroud leakage accelerates since the area reduces as the flow is traveling backwards toward the impeller inlet. This causes the local pressure to drop below the saturation pressure, mixing with the incoming flow at the inducer outlet (see Fig. 5). At the hub leakage path, a similar phenomenon is observed; however, the obstruction of the flow causes repressurization toward the bearing, raising its pressure above saturation. At the LE of the impeller, the local acceleration of the flow is precipitating a similar phenomenon, as shown in the impeller contours in Fig. 5.

Prior to the performance quantification of the pump and its subcomponents, an overview of the main flow features underpinning the operation of each component is provided in Fig. 6. Similar to the

first stage hub leakage flow, the motor cavities fill with the incoming liquid hydrogen from the second stage impeller hub leakage path. This results in an equilibrium of the pressure forces between the impeller fluid and the motor cavity fluid and, in turn, in a negligible interaction between the two flow paths. This is evident by the minuscule mass flow exchange at the hub leakage path inlet which is of similar magnitude to the one shown in Fig. 5 (see normalized mass flow plot). Large separation regions develop inside the impeller passages, which vary in size according to the state of the flow-field and the circumferential position of the blades relative to the volute. Large separation regions also form in the first and second stage diffusing ducts downstream the impeller volutes, including the 180 deg bend. The presence of separated/recirculating regions induces head loss, reducing the efficiency of the components and thereby of the pump.

To better understand the mechanisms driving the large separation regions in the diffusing ducts, the first stage impeller flow-field is examined. Figure 7 shows relative velocity vectors in the first stage impeller and volute along with the flow angle distributions at different planes in the vicinity of the volute tongue.

The absolute flow angle, α_θ , is computed with respect to the tangential component, with increasing positive values indicating

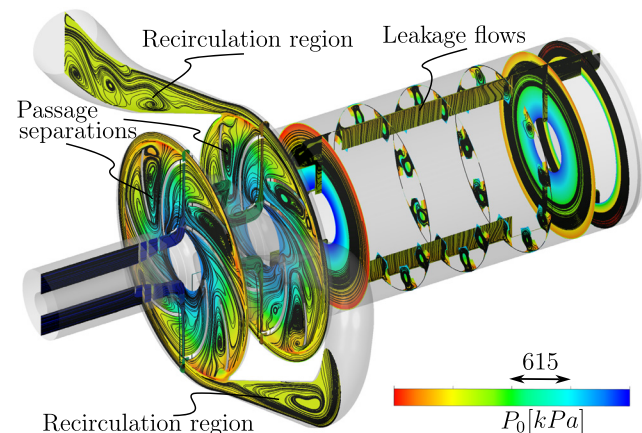


Fig. 6 Overview of the main flow features of the full configuration pump model at design conditions

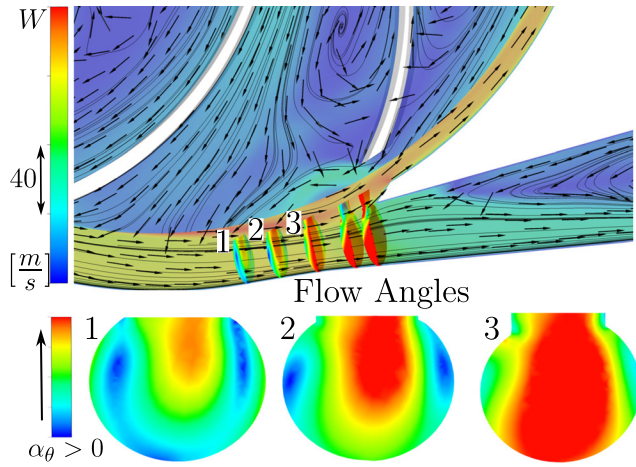


Fig. 7 Velocity vectors (relative in impeller, absolute in volute) and absolute flow velocity angle distributions at five planes near first stage volute tongue

vectors directed more toward the positive radial direction. Focusing on the TE of the impeller blade, it can be seen that the recirculation region is deflecting the flow, causing part of it to migrate toward the suction side near the TE, increasing its absolute flow angle. That portion of the flow clashes locally with the bulk flow of the adjacent passage near the volute tongue, increasing locally the absolute flow angle of the adjacent passage. This causes a radial shift of the flow in the proximity of the volute tongue, which is reflected on the increase in the absolute flow angle, shown on planes (1–3) in Fig. 7. The radial shift along with the diffusion in the duct gives rise to flow separation, resulting in total pressure (and head) loss. The evolution of the losses along with the three-dimensional (3D) streamlines in the ducts downstream the first and second stage volutes are shown in Fig. 8. The highest loss region is in the diffusing duct where the flow forms a large recirculation region, gradually becoming more uniform as it flows through the 180 deg bend (Fig. 8(a)). The flow downstream the second volute (Fig. 8(b)) exhibits identical behavior driven by similar flow mechanisms, further increasing total pressure losses.

To quantify the head changes along the pump, including the aforementioned losses and the impact of leakage flows, the normalized averaged head obtained by the baseline and the full configuration is plotted in Fig. 9, at the planes shown in Fig. 2.

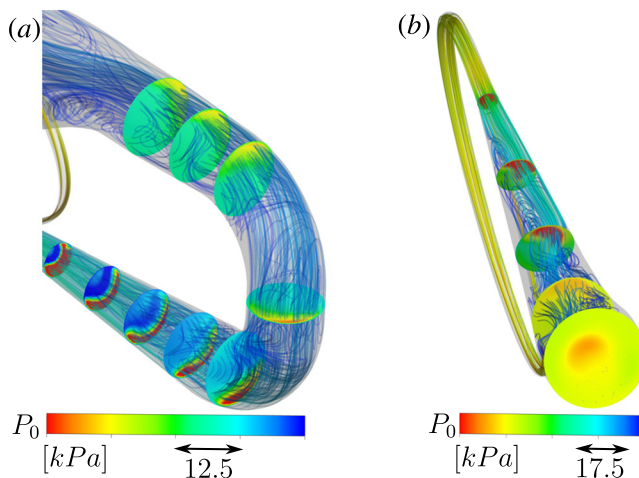


Fig. 8 Overview of 3D streamlines and total pressure losses at different planes along the ducts downstream the (a) first and (b) second stage volutes

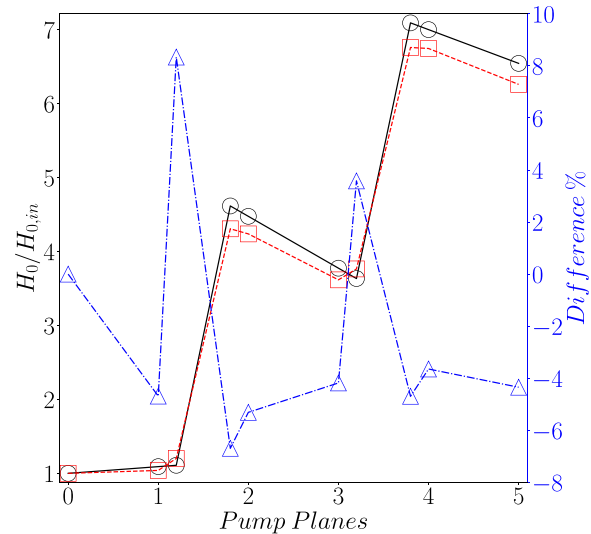


Fig. 9 Comparison of normalized head at different planes along the pump with and without leakage flows

At the inlet of both impellers, downstream the leakage path (planes 1.1 and 3.1), the total pressure of the full configuration is higher by 8.3% and 3.6%; this is due to the mixing of the pressurized shroud leakage flow with the main flow. The operation of the impellers at very high, off-design mass flow conditions results in lower head rise at the impeller outlet, which is evident by the lower measured head at planes 2 and 4. The head losses in the first volute and ducts are larger in the clean configuration, while in the second stage ones, no significant differences can be observed.

To better assess the performance variations of the different components, head changes and efficiencies of each of the major components are extracted for both configurations and compared in Fig. 10. The variations are given as percentage changes with respect to the baseline quantities, with (–) and (+) signs denoting detrimental and beneficial changes, respectively, except for the head changes in the volutes, where the opposite holds true. It is evident that the presence of leakage flow is detrimental for all components except for the first volute and connecting ducts. There is a significant drop in the inducer head rise, due to more extended two-phase regions and mismatch with the first stage impeller which is operating at a very high off-design flow rate, both precipitated by the leakage flows. This is also reflected on the efficiency, which drops by 23.1% in the full configuration. The overall contribution of the inducer is rather negligible compared to the impellers, thereby not significantly affecting the overall pump performance metrics. The mismatch is also affecting the first stage impeller, exacerbating the head rise penalties by 4.5% compared to the second stage (1.2%), resulting in similar efficiency drops by 12.6%. The second stage impeller also achieves a higher head rise due to the lower density of the fluid. The impact of the leakage paths translates to a 4% and 13.5% deterioration in the pump's overall head rise and efficiency, respectively, indicating that the pump requires substantially higher energy while producing slightly less head.

3.2 Off-Design Operation. The pump is expected to operate at a range of different mass flows dictated by the instantaneous engine hydrogen supply requirements. As the engine throttles down, the engine fuel requirements will decrease, shifting the operating point of the pump toward the lower mass flow region. The low mass flow operation of the pump is therefore critical from an operability point of view. Higher than design point flow rates are of lesser practical usage, as the pump would not be expected to operate at such conditions. However, a rather unexpected behavior is exhibited; thus, a more in depth investigation is carried out. A range of higher and lower off-design points are obtained at 60%, 80%, and 100%

speed with and without leakage flows. The overall pump head rise and efficiencies are normalized with the corresponding overall maximum recorded values and plotted against the normalized inlet volumetric flow rate in Fig. 11(a). A more in depth analysis is carried out in lower and upper extreme points; the second lowest and highest mass flow points are selected, corresponding to 34% and 156% of the design volumetric flow rate (see Table 3). These along with the design point are annotated on Fig. 11.

The head rise and efficiency curves show a detrimental impact of leakage flow on pump performance across the majority of the investigated operating range regardless of the rotational speed. In terms of head rise, the penalties are more pronounced at higher speeds, reaching approximately 4% and 4.5% at DP and point (1), but tend to diminish as the pump speed reduces. The most noteworthy impact is the efficiency detriment, which amounts to 13.5% and 11% at the aforementioned conditions. Despite the large shroud leakage flows, the pump is still able to deliver the required head rise but requires significantly more power to overcome the losses and the dissipation of the total pressure carried by the leakage flows. Observing Fig. 11(b), it can be seen that the presence of leakage flows essentially shifts the characteristics downwards (lower efficiencies) and toward the right, behaving as a less efficient, up-scaled (in terms of mass flow) version of the baseline model. When the mass flow though exceeds the design point by a certain amount (140% of the design mass flow at 100% speed), the trend reverses and the leakage model is substantially over performing both in terms of head rise and efficiency. At point (2), the full model outperforms the clean one in terms of head rise and efficiency by a striking 43% and 27%, respectively. The mechanisms

underlying the behavior of the pump at low and high mass flows are investigated, focusing on the 100% speed characteristic.

The amount of leakage flow through the shroud and the hub is essential in the evaluation of their correlation with the overall pump performance. As discussed in the previous paragraphs, the mass flow through the hub leakage paths is negligible since the hub leakage flow of the first stage impeller is impeded by the shaft bearing, while the one of the second stage impeller leads to the motor cavities. Once these fill with hydrogen, the liquid remains trapped and does not interact with the main flow (see Fig. 5). The impeller shroud cavities, however, are creating a closed feedback loop with the main flow path, allowing the backwards traveling of a significant amount of mass flow from the impeller exit to the impeller inlet, as observed at the design point. In absolute values, the shroud leakage mass flow increases with impeller speed but remains approximately invariable throughout the characteristic, i.e., is not a function of pump mass flow. Plotting the volumetric flow rate of the first stage shroud leakage flow normalized with the pump flow rate (inlet) highlights the extend of the leakage flow as a fraction of the overall flow rate (see Fig. 12(a)). At design speed and lowest mass flow setting, the flow rate through the leakage amounts to an astonishing 92.9% (82.5% at point (1)), dropping to 27.6% at DP and 18.5% at point (2),

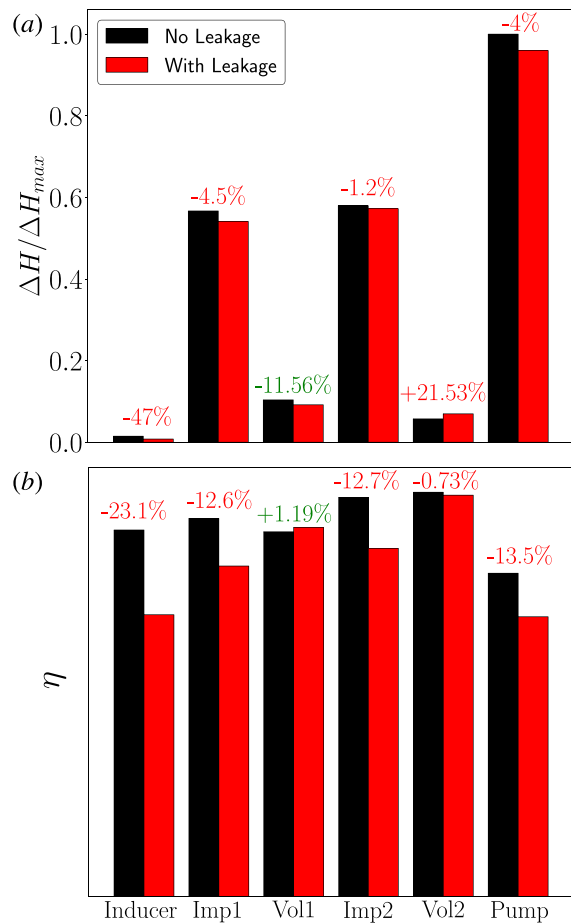


Fig. 10 Comparison and corresponding percental changes of (a) head change and (b) efficiencies of the main components with and without leakage flows

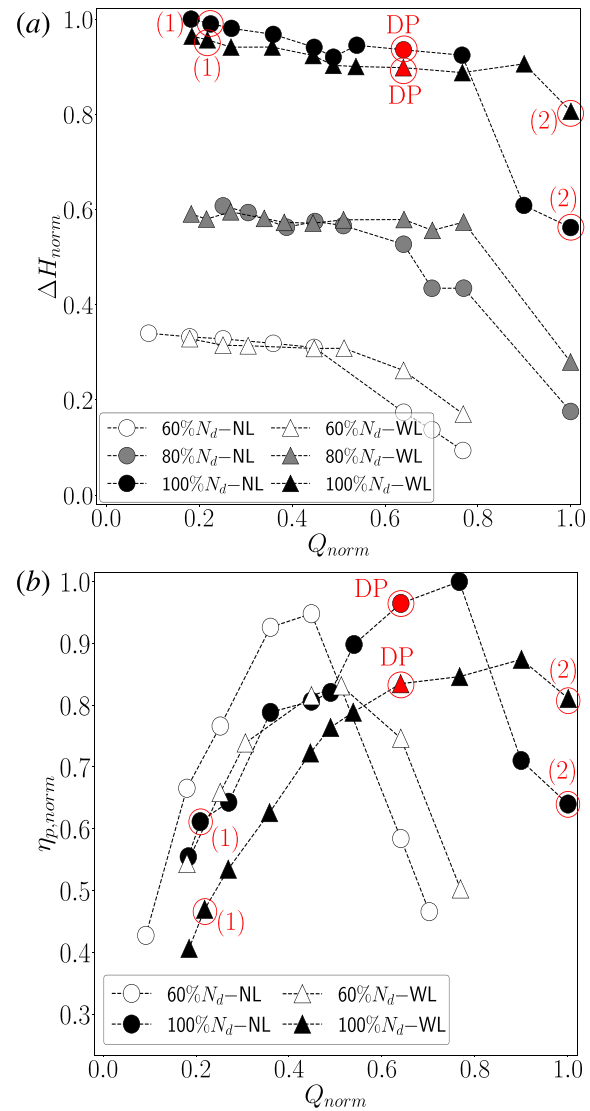


Fig. 11 Comparison of (a) head rise and (b) pump efficiency obtained from the baseline and the full model at 60%, 80%, and 100% speed. Values are normalized with the maximum head rise and efficiency, respectively.

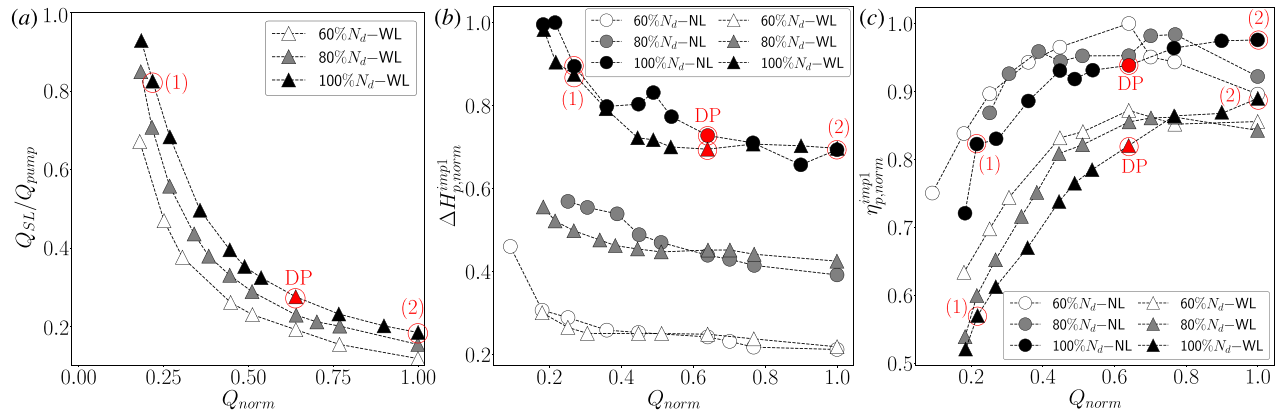


Fig. 12 First stage (a) shroud leakage flow normalized with overall pump mass flow, (b) head rise, and (c) efficiency with and without leakage flows

respectively. The trends are similar for all speeds but the leakage flow rate decreases with speed both in absolute numbers as well as a function of the overall pump flow rate.

The head rise and efficiencies of the first stage impeller with and without leakage flows are plotted against normalized pump inlet

flow rate in Figs. 12(b) and 12(c). Due to the primarily flat profile of the head rise curves, the head does not vary significantly despite the increased flow rate through the impellers, with the clean configuration performing slightly better in the lower mass flow region. The efficiency curves though indicate a consistent efficiency drop across the entire operating range, which is exacerbated as the flow rate decreases. The efficiency diminishes by 8.82% at the highest flow rate point, further reducing by 12.6% at DP and by 36.7% at operating point (1). The increasingly higher efficiency loss is associated with the flow rate through the leakage flow, which dramatically increases (as fraction of overall pump flow rate) at lower mass flows, resulting in higher dissipation of the produced head as this travels back through the shroud leakage path.

While the efficiency of the impellers is consistent with the global pump performance quantities at mid to low flow rates, including DP, it does not reflect on the high flow range, where the baseline pump configuration is under performing. The driving mechanism lies in the ducting components downstream the volutes; the 180 deg bend and the diffusing duct at the outlet. Figure 13 shows the head loss across the first stage volute inlet to second stage impeller inlet (top) and the second stage volute inlet to pump outlet. The head loss is normalized with the head at the volute inlets. For clarity, only the 60% and 100% speed curves are shown. The head losses in the ducting components form a loss bucket, reaching a minimum in the vicinity of the DP and increasing at higher and lower off-design conditions. Similar to pump head rise and efficiency curves, the addition of leakage flows shifts the curves upwards (higher losses) and toward the right, which is more evident at lower rotational speeds. In the mid to low flow rate operating range, the differences are relatively minor; thus, the inefficiency of the impellers is primarily driving the pump performance penalties. At high flow rates though, the head loss difference between the two configurations is significant; at point (2), the head loss of the full configuration reduces by 66.3% and 47% in the first and second stage ducts, respectively. These completely offset the higher efficiency of the impellers, reducing the overall pump efficiency to levels lower than those of the full configuration. The overall pump performance at higher mass flows is therefore limited by the performance of the volutes and the ducting components, rather than the performance of the turbomachinery components.

To identify the underlying flow mechanisms driving the reduced losses in the full configuration, total pressure contours are plotted at different planes across the ducting components downstream the first stage volute (Fig. 14(a)). To better visualize the high loss regions, local colorbar limits are used at each plane. The low total pressure regions visualize the extend of the separated/recirculation region, which increases abruptly shortly after the volute outlet, enters the diffusing duct and gradually increases as the flow is guided toward the second stage. Comparing the baseline and the full configuration, it is evident that the separation region is larger in the baseline configuration. The mechanisms triggering the onset of the separation

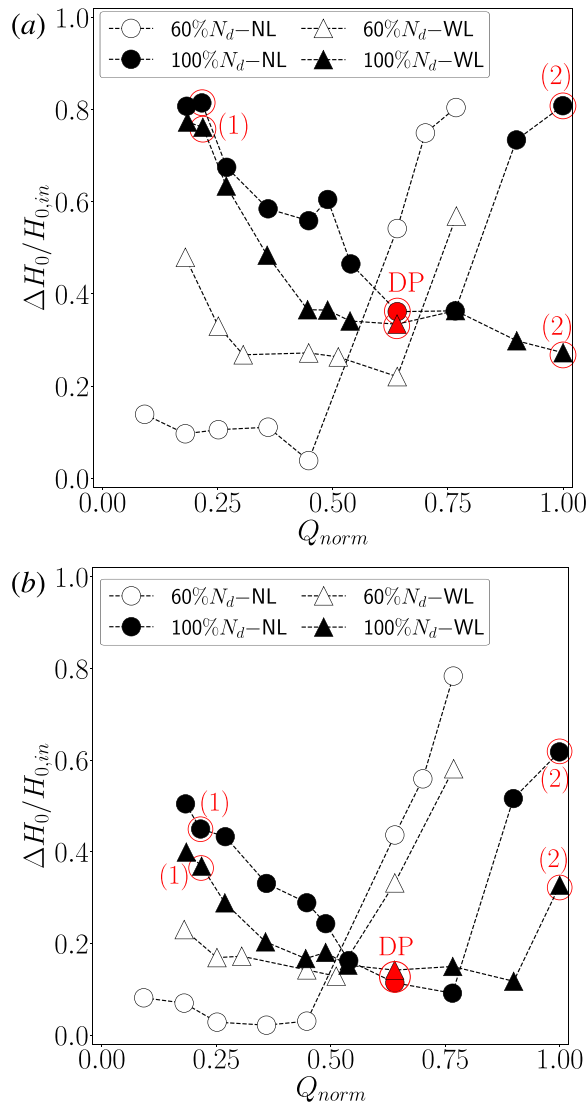


Fig. 13 Head loss normalized with volute inlet head (a) across first stage volute and downstream ducts and (b) second stage volute and downstream ducts

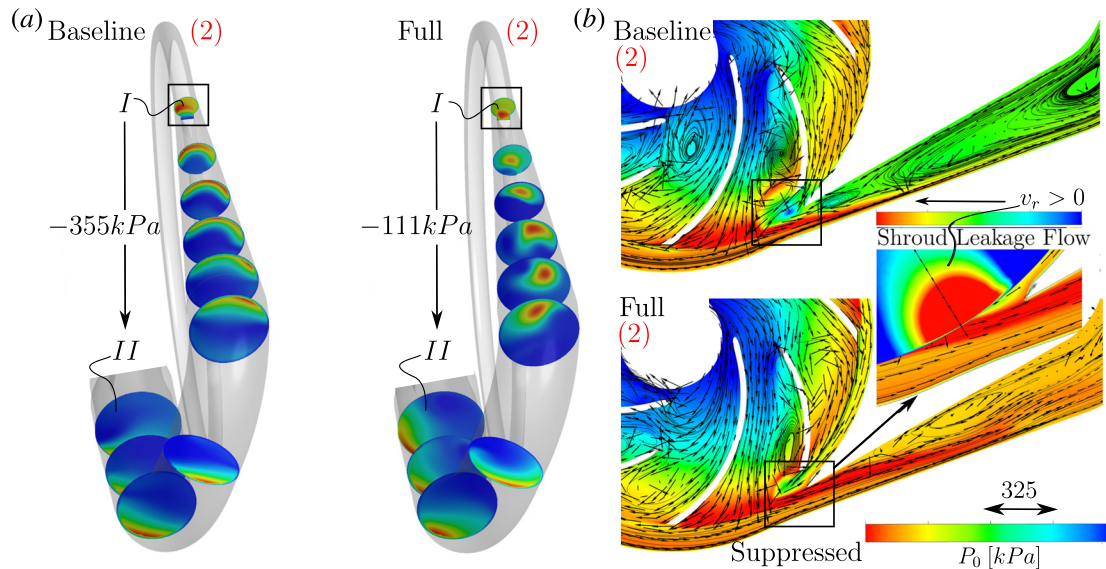


Fig. 14 High flow rate (a) total pressure contours at different planes along first stage ducts using local color bar limits and (b) velocity vectors and total pressure contour comparison of baseline and full configuration. Shroud leakage flow radial velocity contour is also shown.

have been investigated at design conditions and are also underpinning the off-design points. Focusing on Fig. 14(b)), where the impeller streamlines and total pressure distributions are shown, it can be seen that the separation point is shifted further away from the volute tongue, delaying the onset of separation. This occurs due to the shroud leakage flow; the local acceleration of the flow escaping from the blade passage reduces the local pressure. This enables locally part of the shroud leakage flow to reverse and exit at the impeller outlet. This is shown in Fig. 14(b) (bottom right), where contours of positive radial velocity form in the shroud leakage path. This mixes locally with the migrating flow, suppressing it and decreasing the deflection of the main volute flow. This reduces the absolute flow angles compared to the clean configuration near the volute tongue and, by extension, the radial shift of the streamlines which initiates the downstream flow separation. The beneficial effect of the shroud leakage flow manifests only when the local acceleration of the flow is sufficiently high to create that suction phenomenon in the shroud leakage path, which explains its occurrence only in the high flow region.

4 Summary and Conclusions

The effect of leakage flow on a two-stage, hydrogen pump is assessed and quantified via CFD simulations by comparing the global and component level performance metrics of two configurations: one without leakage flows (baseline) and one with (full). The main conclusions are summarized below:

- (1) The large changes in the compressibility factor between the inlet and the outlet require RGP tables for proper modeling. A reduction in density is recorded, due to the compressibility of hydrogen and due to the presence of regions where vapor would form. Such regions are captured in the leakage paths and the impeller LE of the first stage, thereby highlighting the necessity of explicit two-phase modeling.
- (2) The flow rate through the hub leakage paths and the motor cavities is negligible as these fill with hydrogen and do not interact with the main flow. The acceleration of the flow is sufficient to decrease the pressure below the saturation one.
- (3) The shroud leakage flows create a closed loop system with the impeller flow path, increasing the flow rate through the impellers by 27.6% at design point and up to 92.9% at the lowest flow rate. Presence of vapor is predicted due to the acceleration of the backwards traveling shroud leakage flow.
- (4) Leakage flows detrimentally affect both the pump and turbomachinery components head rise and efficiency. The

head loss does not exceed 4.5% across the entire operating range, but substantially higher power input is required, reflected on the large efficiency drop of 13.5% and 12.6% for the pump and the impellers at design conditions.

- (5) At mid to low flow rates, head rise and efficiency penalties are primarily driven by the dissipation of the head, as a large fraction of the flow is recirculated through the shroud leakage paths.
- (6) At high flow off-design conditions, the leakage model is outperforming the baseline one, by 43.4% and 26.7% in head rise and efficiency, respectively. This is due to reduced losses in the volutes and ducting components which completely offset the more inefficient operation of the impellers.
- (7) The losses in the volutes are governed by the radial shift of the streamlines near the volute tongue. This is triggered by the passage separations, which deflect the flow, allowing it to migrate toward the adjacent blade through the TE, mixing with the incoming flow from the volute. This increases the absolute exit flow angle at the volute tongue promoting flow separation. At higher mass flows, the acceleration is large enough to allow locally the shroud leakage flow to travel forward, toward the impeller outlet, suppressing that effect and reducing separation losses.
- (8) The study has quantitatively described the deviation of performance metrics in the realizable engineering hardware with leakage flow paths, as compared to the typical characterization practice, which is based only on main flow path components. Therefore, this study provides margins to be considered in the baseline design practice for aircraft representative low-specific speed LH₂ pumps that need to operate with large turn-down ratios. The form of the characteristics and their range of deviation could be used as generic scalable formats in lower order fuel system modeling to enable synergistic cycle innovation studies for hydrogen aircraft.

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Data Availability Statement

The authors attest that all data for this study are included in the paper.

Nomenclature

Latin Letters

F = force
 g = gravitational acceleration
 H_0 = head using stagnation pressure
 $\dot{m}$ = mass flow
 P_{sh} = shaft power
 P_0 = stagnation pressure
 Q = volumetric flow rate
 r = radius
 t = time
 W = relative velocity magnitude

Greek Symbols

α_θ = absolute flow angle with respect to tangent
 η_p = efficiency
 Λ = generic scalar quantity
 $\bar{\Lambda}$ = averaged generic scalar quantity
 Λ' = fluctuating component of a generic quantity
 ρ = density
 τ = torque
 ω = angular velocity

Subscripts

ex = extrapolated property
imp = impeller property
in = inlet property
norm = normalized property
out = outlet property

Abbreviations

DP = design point

LE = leading edge
LH₂ = liquid hydrogen
NL = no leakage
RGP = real gas property
TE = trailing edge
WL = with leakage

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