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Published in:
Applied Energy

Link to article, DOI:
[10.1016/j.apenergy.2019.113958](https://doi.org/10.1016/j.apenergy.2019.113958)

Publication date:
2020

Document Version
Peer reviewed version

[Link back to DTU Orbit](#)

Citation (APA):
Wen, C., Rogié, B., Kærn, M. R., & Rothuizen, E. D. (2020). A first study of the potential of integrating an ejector in hydrogen fuelling stations for fuelling high pressure hydrogen vehicles. *Applied Energy*, 260, [113958].
<https://doi.org/10.1016/j.apenergy.2019.113958>

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A first study of the potential of integrating an ejector in hydrogen fuelling stations for fuelling high pressure hydrogen vehicles

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Abstract: This present study evaluates the potential of entraining the low-pressure hydrogen to fuel cell vehicles during fuelling processes, which is expected to promote the development of the hydrogen fuel automotive industry. A computational fluid dynamics model is developed to evaluate the potential of the proposed hydrogen fuelling process. A flow behaviour analysis is performed to show the detailed flow structure in the critical and sub-critical processes for the hydrogen ejector. The critical suction pressure and critical back pressure are assessed under various inlet pressures of the primary nozzle. The results show that the high-pressure hydrogen accelerates in the primary nozzle, leading to the decrease of the static pressure, which generates the suction effect in the downstream of the nozzle exit to entrain the hydrogen from the low-pressure tank. The entrainment ratio declines along with the increasing back pressure or decreasing inlet pressure of the suction chamber. This study suggests that the integration of an ejector instead of a reduction valve into the hydrogen fuelling station improves the energy efficiency by utilizing less hydrogen from the high-pressurized hydrogen storage during vehicle fuelling.

Keywords: hydrogen; fuelling process; ejector; entrainment ratio; fuel cell vehicle

1. Introduction

Hydrogen is recognised as a remarkable fuel as it potentially can be produced without pollution and greenhouse gas emissions to the environment by using renewable electricity sources [1]. Hydrogen as an energy carrier for personal transportation is a well-known concept [2, 3] and the fuelling procedure for high pressure hydrogen storage is standardised. Society of Automotive Engineers describes the fuelling procedure of hydrogen vehicles in the standard: Surface vehicle standard J2601 [4]. The hydrogen fuelling station plays an important role for the development of the fuel cell electric vehicle (FCEV), as the fuelling costs and the availability is crucial for the penetration of FCEV into the market for personal transportation.

The design, analysis and optimisation of the hydrogen fuelling station have been carried out for many years. Agll et al. [5] designed a drop-in hydrogen fuelling station, which allowed a streamlined introduction of hydrogen in the vehicle fuelling infrastructure. The modular design approach with the off-the-shelf items is expected to a design with the capability of mass production, and ease in transport and integration. Xu et al. [6] performed a near-term analysis of a roll-out strategy to introduce fuel cell vehicles and hydrogen stations in Shenzhen China, and it was estimated that the total capital investment for 10 hydrogen stations is approximately \$19.7 million. Lin et al. [7] proposed an approach to determine the optimal delivered hydrogen pressure for fuel cell electric vehicles during fuelling process, and it was suggested that the delivered hydrogen pressure of 700 bar showed a better performance than 350 bar or 500 bar, regardless of fuel availability, FCEV adoption, driver types, time values and fuel

economies. Jin et al. [8] proposed a novel high multi-stage pressure reducing valve, which is expected to be used for hydrogen stable decompression in hydrogen fuelling station. Their numerical simulation showed that the novel structure enabled a multi-stage regulation of fluid pressure in the valve, less fluid turbulent vortexes, high uniformity of fluid flow and less process fluid loss. Kuroki et al. [9] developed a thermodynamic analytical approach to analyse the transient temperature rise of hydrogen when pre-cooled hydrogen was heated through filling equipment at a hydrogen fuelling station. Their analysis showed that the actual filling equipment can be assumed to be the simple pipeline and the temperature rise of hydrogen heated through actual filling equipment at any time and at any position can be predicted by the proposed analytical approach. Kim et al. [10] focused on the leak risk of a hydrogen fuelling station and employed the computational fluid dynamic modelling to evaluate the risk level for the safety design of the station system. Kodoth et al. [11] evaluated the uncertainty in accident rate estimation at hydrogen fuelling station using a time correlation model, their studies showed that the uncertainty in the estimation increased when the operation time is long owing to the decreasing data. The aforementioned studies present that the hydrogen fuelling station uses well known technology but the most research until now have been on the fuelling procedure and making reliable functioning stations rather than optimizing energy consumption.

The energy consumption of an isolated hydrogen fuelling is app. 0.6-0.8 kWh/kgH₂ for compression and cooling, where the compression work account for approximately 50 % of the consumption. If considering low utilization of the station the energy

consumption can increase to more than 10 kWh/kg_{H2} [12, 13]. The total cost of a hydrogen fuelling, without production and delivery of the hydrogen to the station, is app. 3\$/kg_{H2} [14, 15]. This study proposes a novel concept substituting the conventional reduction valve with an ejector in order to minimize the energy losses during the fuelling process, as more hydrogen at lower pressure can be utilised. The ejector provides an economic and eco-friendly development to consume the energy resource in a sustainable way [16, 17] and improve the quality of the low-grade heat energy by entraining the low-pressure stream [18, 19].

This present study develops a computational fluid dynamics model to predict the performance of the hydrogen ejector during vehicle fuelling. The flow field inside the hydrogen ejector is analysed in the critical and sub-critical fuelling processes. The effects of the inlet pressure of the suction chamber and the back pressure of the ejector are performed based on the evaluation of the entrainment ratio. The results demonstrate the potentials of integrating ejectors into the hydrogen fuelling stations for improved overall energy efficiency. Although in this study the aim is to investigate the concept and find out if there is any potential for further investigation, the result does therefore not resemble a hydrogen fuelling as standardised by SAE but rather a proof of concept using different static pressures in the CFD modelling.

2. Problem description

A new concept for hydrogen fuelling station is proposed by integrating an ejector into the current system, as shown in Fig. 1. The substitution of the reduction valve with the ejector is expected to contribute towards more efficient utilization of hydrogen

energy. On one hand, the high-pressure hydrogen entrains the low-pressure fluid to the mixing section of the ejector, which achieves efficient utilization of the high-pressure energy and expansion work recovery. On the other hand, the ejector can lower the storage pressure of the low-pressure tanks, in which the fuelling station will benefit from the transmission process between the truck and storage tank. The potential of the new throttling concept implemented into the design of the hydrogen fuelling station could potentially contribute to the implementation of hydrogen fuel cell vehicles into the market.

Fig. 1 A new concept integrating ejectors for hydrogen fuelling stations

A typical ejector consists of a primary nozzle, a suction chamber, a mixing section, a constant area section and a diffuser [20, 21], as shown in Fig. 2. A primary nozzle is employed to accelerate the high-pressure fluid, where the static pressure and temperature decrease due to the increase of the gas velocity [22]. The low-pressure fluid will be entrained as a result of the suction effect downstream of the primary nozzle. The high-pressure fluid and the low-pressure fluid are mixed in the mixing section and constant area section. The diffuser is used to recover the pressure energy. The complicated flow behaviour includes transonic flow, mixing process of high-pressure and low-pressure fluids, and shock waves [23]. One of the main parameters to represent

the ejector performance is the entrainment ratio, which is defined as the ratio of the entrained mass flow from the suction chamber to the motive mass flow from the primary nozzle [24]:

$$\gamma = \frac{\dot{m}_s}{\dot{m}_p} \quad (1)$$

where γ is the entrainment ratio, $\dot{m}_s$ is the mass flow from the suction chamber, $\dot{m}_p$ is the mass flow from the primary nozzle.

Fig.2 Schematic diagram of an ejector

In this study, the ejector is designed based on the maximum mass flow rate of 0.06 kg/s for the hydrogen fuelling process, according to the limit set by SAE's fuelling protocol [4]. The inlet pressure of the primary nozzle and the suction chamber is chosen to be 450 bar and 300 bar, respectively, resembling a low and medium pressure tank in a cascade system. The ejector dimensions are listed in Table 1. A converging nozzle is employed as the primary nozzle for this hydrogen ejector. The area ratio of the hydrogen ejector is 12.96, which is defined as the ratio of the mixing section inlet area to the primary nozzle throat area. The nozzle exit position between the primary nozzle outlet and mixing section inlet is fixed to be 0 mm. The 2D axisymmetric geometry is employed in the CFD modelling to evaluate the performance of the hydrogen ejector.

The computational domain is discretized by the structured mesh to reduce the numerical diffusion, as shown in Fig. 3. To ensure that the y^+ is less than 1 for the shear stress transport (SST) $k-\omega$ turbulence models, the mesh in the boundary layer was refined along the steam ejectors including the walls of the primary nozzle, suction chamber, mixing section, constant area and diffuser.

Fig. 3 Grid system of the hydrogen ejector

Table 1. Dimensions of the hydrogen ejector

Geometrical parameters	Value (mm)
Diameter of primary nozzle inlet	6.00
Diameter of primary nozzle outlet	1.00
Length of primary nozzle	6.00
Nozzle exit position	0.00
Diameter of mixing section inlet	3.60
Diameter of constant area	2.12
Diameter of diffuser	4.00
Length of mixing section	5.00

Length of constant area	21.21
Length of diffuser	23.02

3. Mathematical model

3.1. Governing equations

High-speed compressible flow occurs inside the ejector and features complicated phenomena such as supersonic flows, shock waves and turbulent mixing processes. The conservation equations are recognized as the Navier-Stokes equations and coupled to the real gas equation of state by means of Refprop database [25]. The conservation equations include the mass, momentum and energy equations, which are written as follows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = 0 \quad (2)$$

$$\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} \quad (3)$$

$$\frac{\partial}{\partial t} (\rho E) + \frac{\partial}{\partial x_i} (u_i (\rho E + p)) = -\frac{\partial}{\partial x_i} (\lambda_{eff} \frac{\partial T}{\partial x_i}) + \frac{\partial}{\partial x_i} (u_i \tau_{ij}) \quad (4)$$

$$\tau_{ij} = \mu_{eff} \left[\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \frac{\partial u_l}{\partial x_l} \delta_{ij} \right] \quad (5)$$

where ρ , u , p and E are the density, velocity, pressure and total energy, respectively. τ_{ij} is the stress tensor. μ_{eff} is the effective turbulent viscosity. λ_{eff} and T are the effective heat conductivity and temperature, respectively.

3.2. Turbulence model

Violent turbulence has been observed in the ejectors with large Reynolds numbers [26, 27], and therefore, a turbulence model is needed to calculate the complicated fluid

flows. Croquer et al. [28] compared and evaluated the two-equation turbulence models including the standard k - ε , RNG k - ε , realizable k - ε and shear stress transport (SST) k - ω turbulence models. The comparison results based on the static pressure distribution and entrainment ratio showed that the SST k - ω turbulence model is the best one for the prediction of the complicated flows inside an ejector. As a consequence, the SST k - ω turbulence model [29] is used in this CFD modelling due to the good accuracy to predict the ejector flows. The turbulent kinetic energy, k , and the specific dissipation rate, ω , equations in the SST model are as follows:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j} \left(\Gamma_k \frac{\partial k}{\partial x_j} \right) + \bar{G}_k - Y_k + S_k \quad (6)$$

$$\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_j}(\rho \omega u_j) = \frac{\partial}{\partial x_j} \left(\Gamma_\omega \frac{\partial \omega}{\partial x_j} \right) + C_\omega - Y_\omega + D_\omega + S_\omega \quad (7)$$

where k is the turbulent kinetic energy, ω is the specific dissipation rate. Γ_k and Γ_ω are the effective diffusivity of k and ω , respectively.

3.3. Numerical setup

In this numerical simulation, the computational platform is based on the commercial package ANSYS FLUENT version 18 [30], which employs the finite volume method to solve the governing equations. The implicit density-based solver is employed to calculate the high-speed compressible flows in the hydrogen ejector. The second-order upwind scheme is adopted for an accurate prediction. The pressure inlet conditions are assigned for the inlets of the primary nozzle and suction chamber, while the outlet of the ejector employs the pressure outlet condition. No-slip and adiabatic boundary conditions were used for the ejector walls. Table 2 represents an example of

the boundary conditions for the numerical simulation.

Table 2. Boundary conditions for hydrogen ejectors

Hydrogen ejector	Boundary conditions	Pressure (bar)	Temperature (K)
Inlet of the primary nozzle	Pressure inlet	450	298.15
Inlet of the suction chamber	Pressure inlet	300	298.15
Outlet of hydrogen ejector	Pressure outlet	300	298.15
Wall	No-slip	N/A	N/A
	adiabatic wall		

4. Results and discussion

4.1. Validation and verification of CFD model

4.1.1 CFD validation

The ejector in Nikiforow et al. [31] was employed to validate the developed CFD model with experiments at smaller pressures than the ones for hydrogen fuelling station. The ejector was designed with a high area ratio of 16, which is defined as the ratio of the cross-sectional area of the mixing chamber to that of the nozzle throat. The diameters of the nozzle throat and the mixing chamber were 0.50 mm and 2.00 mm, respectively. The quadrilateral grids were employed for this simulation based on the assumption of the 2D axisymmetric geometry of the fuel cell ejector. The mesh was refined by using the adaptive refinement mesh strategy in ANSYS FLUENT based on the numerical results, such as the pressure gradient for this case. Fig. 4 shows the comparisons between the numerical simulation and experimental data of the mass flow

rate and entrainment ratio for the fuel cell ejectors. It is shown that the numerical simulation predicts higher mass flow rates than the experimental measurements with a maximum relative error of approximately 2.96%. For the entrainment ratio, the numerical entrainment ratios are smaller than the experimental ones, and the maximum relative error is about 8.58%. Consequently, it can be concluded that the numerical model accurately predicts the flow behaviour in the ejectors for hydrogen at these mass flows and pressures.

Fig. 4 Numerical and experimental results of mass flow rate and entrainment ratio for different primary inlet pressures for hydrogen ejectors in the fuel cell system.

4.1.2 Similarity

The CFD model agrees very well with the low-pressure experiments. However, for the hydrogen fuelling station, the primary pressure is much higher. In order to study the effect of having higher pressures, a similarity investigation was carried out. Table 3 shows the computational conditions for the back pressure of hydrogen ejectors and the calculated entrainment ratios with various pressures from 4.5 bar to 450 bar. The pressure ratios among the primary inlet, suction inlet and ejector outlet are fixed to 4.5-1-1 for the three similarity cases. It was expected that the increasing primary inlet

pressure induces a higher entrainment ratio as the pressure difference between the primary inlet and suction chamber was much greater at high-pressure operation. Fig. 5 describes the Mach numbers inside three hydrogen ejectors at the primary inlet pressures of 4.5 bar, 45 bar and 450 bar, respectively. The Mach number was less than unity in the constant area of the hydrogen ejector at the primary inlet pressure of 4.5 bar, which means that the ejector goes into the sub-critical conditions. On the contrary, for both the other two cases under high-pressure conditions, the hydrogen ejectors work in the critical conditions although the 45 bar primary inlet pressure presents a larger Mach number drop. This shows that the Mach number will be high when used for high pressure hydrogen fuelling. In general, the CFD model is reasonable to perform the numerical simulation of hydrogen ejector under high-pressure conditions from the view of the local fluid structure, entrainment ratio and similarity aspects.

Table 3. Numerical conditions and results for hydrogen ejectors at various primary pressures

Numerical cases	Inlet pressure of primary nozzle (bar)	Inlet pressure of suction chamber (bar)	Back pressure of ejector (bar)	Entrainment ratio (-)
Case 1	4.5	1.0	1.0	0.56
Case 2	45	10	10	0.62
Case 3	450	100	100	0.68

Fig. 5 Mach numbers in hydrogen ejectors at various primary pressures

4.1.3 Mesh sensitive analysis

The mesh density is one of the crucial factors for the developed CFD model to predict the hydrogen ejector performance for the fuelling station. In this study, both the local quantity, static pressure, and global performance, entrainment ratio, are employed to evaluate the influence of the mesh sizes on the ejector behaviour. The geometrical dimensions of the ejectors for the mesh independent test are shown in Table 1. The coarse mesh of 36 000 cells, the medium mesh of 72 000 cells and the fine mesh of 187 500 cells are used for the mesh sensitive analysis, respectively. The comparison results are shown in Fig. 6 and Table 4, respectively.

Fig. 6 shows the distribution of the local static pressure at the axis along the flow direction inside the hydrogen ejectors under the three different mesh sizes. Both the medium and fine meshes predict very similar static pressure even in the low-pressure

region in the constant area of the hydrogen ejector due to the gas expansion. For the coarse mesh, the tiny difference of the local static pressure is observed in the constant area compared to the medium and fine meshes. All of the three mesh sizes calculate almost the same static pressure in the other parts of the hydrogen ejector. Table 4 lists the predicted mass flow rate through the primary nozzle and the entrainment ratio of the hydrogen ejectors under various mesh densities. It can be seen that the coarse mesh predicts the maximum relative errors of approximately 0.6% and 1.67% for the mass flow rate and entrainment ratio, respectively. The refinement of the numerical mesh sizes does not significantly contribute to the improvement of the numerical accuracy of the global quantities of mass flow rate and entrainment ratio. Therefore, in view of the local quantity of the static pressure and global performance of the entrainment ratio, the medium mesh size of 72 000 cells is employed for the numerical simulation considering the balance between the numerical accuracy and computational cost.

Fig. 6 Static pressure in ejectors for hydrogen fuelling stations with various numerical meshes

Table 4. Mesh sensitive test for entrainment ratio of hydrogen ejectors

Cases	Mesh size	Mass flow rate (Kg/s)	Relative error	Entrainment ratio (-)	Relative error
Coarse mesh	36 000 cells	0.0166	0.60%	2.56	1.16%
Medium mesh	72 000 cells	0.0167	0	2.57	0.77%
Fine mesh	187 500 cells	0.0167	-	2.59	-

4.2. Flow features in hydrogen ejectors

There are two representative operation modes of the hydrogen ejectors, namely, the critical operation and sub-critical operation, as shown in Fig. 7. In the critical operation, the entrainment ratio is constant with the increase of the back pressure of the hydrogen ejector. In the sub-critical operation, the increasing back pressure reduces the entrainment ratio until the appearance of reverse flow in the suction. The flow field inside the hydrogen ejectors is analysed in detail for these two typical operation modes. Table 5 lists the pressure conditions for the primary nozzle, suction chamber and outlet of the ejector for the numerical simulations.

Table 5. Static pressure for hydrogen ejector simulation

Hydrogen ejector	Temperature (K)	Static pressure (bar)	Working process
Inlet of the primary nozzle	298.15	450	From critical to

Inlet of the suction chamber	298.15	300	reverse flows
Outlet of hydrogen ejector	298.15	100-380	

Fig. 7 shows the resulting entrainment ratio of the hydrogen ejector. It can be observed that the hydrogen ejector works in the critical operation when the back pressure increases from 100 bar to 250 bar, and in the sub-critical operation when the back pressure increases from 250 bar to 340 bar, respectively. The ejector will result in reverse flows if the back pressure increases further.

Fig. 7 Effect of back pressure on entrainment ratio of hydrogen ejectors

Figs. 8 - 9 describe the static pressure and Mach number contours inside the hydrogen ejectors for critical and sub-critical processes with a back pressure of 100 bar and 320 bar, respectively. The high-pressure hydrogen both accelerates in the primary nozzle for these two processes, where the static pressure declines along the flow direction. It generates the suction effect in the downstream of the nozzle exit to entrain

the hydrogen from the suction chamber. For the back pressure of 100 bar, the hydrogen continues to accelerate in the constant area, where the critical condition is reached with the Mach number of unity and thus choked flow. As a result of the lower pressure at the ejector outlet, the hydrogen further expands in the diffuser with the Mach number up to 2.94 and the static pressure decreases sharply to 12 bar. In this condition, the shock wave is observed, which results in energy losses for ejectors. On the contrary, the acceleration of the hydrogen only occurs in the front part of the constant area, when the back pressure increases to 320 bar at the ejector outlet. The maximum Mach number is approximately 0.79, and far from the choked flow. It indicates that the high back pressure restricts the expansion of the hydrogen in the downstream of the primary nozzle. It also can be observed that the effective suction is extremely small, which reduces the suction capacity and leads to lower entrainment ratios of the hydrogen ejectors.

Fig. 8 Contours of static pressure in hydrogen ejectors

Fig. 9 Contours of Mach number in hydrogen ejectors

4.3. Effect of suction inlet pressure

As described in the introduction, the ejector is employed to entrain the hydrogen from the low-pressure storage tank. The hydrogen fuelling station will benefit from the new design if the gas pressure can be as low as possible herein. In this section, the effect of the inlet pressure at the suction chamber will be evaluated to achieve the effective use of the hydrogen ejector. The computational conditions are listed in Table 6 at various back pressures of hydrogen ejectors.

Table 6. Computational conditions for evaluation of the suction inlet pressure

Computational cases	Inlet pressure of primary nozzle (bar)	Back pressure of ejector (bar)	Inlet pressure of suction chamber (bar)
Case 1	450	150	40 - 60
Case 2	450	200	80 – 120

Case 3	450	250	150 - 190
Case 4	450	300	210 - 250

Fig. 10 shows the effect of suction inlet pressure on the mass flow rates at the primary nozzle, the inlet of the suction chamber and ejector outlet, when the inlet pressure of primary nozzle and back pressure of ejector are fixed at 450 bar and 300 bar respectively. In these conditions, the mass flow rate through the primary nozzle increases with the decrease of the inlet pressure at the suction chamber. It indicates that the increasing pressure difference between the primary nozzle and suction chamber plays a positive role in the mass flow through the primary nozzle. On the contrary, the mass flow rates through the suction chamber and ejector outlet decline with the decrease of the inlet pressure of the suction chamber. In this simulation case, the mass flow rate through the suction chamber is approximately zero when the inlet pressure of the suction chamber reaches 230 bar. It is evident that the reverse flow will appear if we continuously reduce the inlet pressure of the suction chamber below 230 bar.

Fig. 11 presents the effect of the inlet pressure of the suction chamber on the entrainment ratio for the four simulation cases with the different back pressure of the hydrogen ejectors. For each case, the entrainment ratio declines with the reducing inlet pressure of the suction chamber. For instance, the entrainment ratio decreases from 0.38 to -0.40 when the inlet pressure of the suction chamber changes from 250 bar to 210 bar at a back pressure of 300 bar. This means that there is an inlet pressure of the suction chamber for which the entrainment ratio is equal to zero. We defined this condition as the critical suction pressure, P_{cs} . We also can observe from all of the four computational

cases that the decreasing ejector back pressure induces the decrease of this critical suction pressure.

Fig. 10 Effect of suction chamber pressure on the mass flow rates

Fig. 11 Effect of suction chamber pressure on the entrainment ratio

4.4. Effect of ejector back pressure

One of the features of the ejector integrated into the fuelling station is that the back pressure of the hydrogen ejector is increasing during the fuelling process. Hence, it is necessary to evaluate the effect of increasing back pressure on the working performance of the hydrogen ejectors. This section will test the influence of the back pressure of the hydrogen ejectors. Table 7 lists the computational conditions.

Table 7. Computational conditions for the back pressure of hydrogen ejectors

Computational cases	Inlet pressure of primary nozzle (bar)	Inlet pressure of suction chamber (bar)	Back pressure of ejector (bar)
Case 1	450	300	100 – 400
Case 2	600	300	350 – 420
Case 3	700	300	390 – 440
Case 4	800	300	410 – 450
Case 5	900	300	360 – 480

Fig. 12 shows the effect of ejector back pressure on the mass flow rates and entrainment ratio, when the inlet pressures of the primary nozzle and suction chamber were fixed at 450 bar and 300 bar, respectively. Fig. 12 (b) is an extension of Fig. 7, which provides more details of the effect of various back pressures. The primary mass flow rate is constant regardless of the back pressure, which means that the flow behaviour in the primary nozzle is not affected by the back pressure of the hydrogen ejector. The mass flow rate through the ejector and the entrainment ratio keeps constant

when the back pressure varies from 100 bar to 250 bar. It indicates that the ejector operates in the critical operation region. As the back pressure further increases, the operation enters the subcritical region. The reverse flow appears if the back pressure is more than 345 bar. Accordingly, the ejector cannot entrain the hydrogen from the suction chamber. We defined this condition as the critical back pressure, P_{cb} , in which the entrainment is zero.

Fig. 13 depicts the effect of the ejector back pressure on the entrainment ratio under various inlet pressures of the primary nozzle. It can be observed that increasing back pressure reduces the entrainment ratio for all of the computational cases. Furthermore, the critical back pressure can be improved by increasing the inlet pressure of the primary nozzle. For instance, the critical back pressure is improved from 345 bar to 480 bar when the inlet pressure of the primary nozzle rises from 450 bar to 900 bar. Further evaluation of the results illustrate that the pressure difference ratio, λ , equals a constant value of 0.3 at the critical back pressure, where λ is defined as follows:

$$\lambda = \frac{P_b - P_s}{P_p - P_s} \quad (9)$$

where P_p and P_b are the inlet pressure of the primary nozzle and the back pressure of the hydrogen ejector, respectively.

Table 8 lists the pressure difference ratio at the critical back pressure condition for all simulated inlet pressures of the primary nozzle. The results suggest that we can estimate the critical back pressure of the hydrogen ejector during the fuelling process by using this simple constant value.

Fig. 12 Effect of ejector back pressure on the mass flow rates and entrainment ratio

Fig. 13 Effect of ejector back pressure on entrainment ratio and critical back pressure as a function of primary nozzle pressure.

Table 8. The relationship among critical back pressure, inlet pressures of the primary nozzle and suction chamber

Inlet pressure of primary nozzle (bar)	Inlet pressure of suction chamber (bar)	Critical back pressure (bar)	$\lambda = \frac{P_b - P_s}{P_p - P_s}$ (-)
450	300	345	0.3
600	300	390	0.3
700	300	420	0.3

800	300	450	0.3
900	300	480	0.3

4.5. Discussion

Currently, and to the author's best knowledge, there is no available experimental data on the ejectors for the hydrogen fuelling stations with high-pressures. The ejector used for the CFD validation was designed by Nikiforow et al. [31] and was utilized in a proton exchange membrane fuel cell systems to realize the hydrogen recirculation. The big challenge between these two systems is the huge differences of the low pressure (4 bar for the fuel cell) and high-pressure (up to 900 bar for the fuelling station), and correspondingly the mass flow rates. For instance, the mass flow rate of the hydrogen through the ejector in the fuel cell system ($\sim 10^{-5} - 10^{-4}$ kg/s) is much smaller than that in the fuelling stations ($\sim 10^{-2}$ kg/s). For this purpose, we performed three similarity simulation cases to clarify the effect of the CFD simulation results under various pressure conditions. Moreover, we found similar local fluid structure, entrainment ratio and similarity aspects under different operating pressures. Therefore, the CFD model was found to be reasonably accurate to predict the ejector performance at high operation pressures too.

The current investigation serves to provide some first results of the proposed concept to employ ejectors for hydrogen fuelling stations. The investigation shows that the ejector does entrain much hydrogen from the lower pressure tank, when it operates in the critical process. On the contrary, the entrainment ratio or suction mass flow decrease while entering the sub-critical process. Finally, a reverse flow situation occurs in the

suction, in case the suction does not operate with a check valve. For this case, the ejector simply acts as a valve, expanding the same mass flow from the high-pressure tank to the hydrogen vehicle. The most promising case is with 900 bar primary nozzle pressure as the ejector function in a back pressure range from 360 bar to 480 bar and the average entrainment ratio is 0.75. Considering a 7 kg fuelling from 5 bar to 800 bar in the vehicle it corresponds to approximately 10 % of the fuelling, utilizing 0.7 kg of hydrogen from the low pressure storage which otherwise would have been drawn from the high pressure tank. This could potentially be larger if pressures are optimised and a more advanced control with more storage tanks.

All the CFD simulations has been carried out with static pressures, as the computational time increased from hours to days when simulating one second dynamically. The study therefore shows how a given ejector will perform for the specific primary pressure, suction chamber pressure and back pressure. Considering figure 13 with different primary pressures is shows that the potential of the ejector increases with increasing primary pressure. However, a hydrogen fuelling is typically carried out with several pressure stages at the station, varying from 200 bar to more than a 1000 bar, and it is not given that the station would use a 900 bar storage at the station to fill at 400 bar in the vehicle. The potential of the ejector is therefore limited to the pressure difference of the primary pressure and suction chamber pressure. A simpler model of an ejector which can be deployed into a dynamic simulation software would be needed to investigate the full fuelling process under different conditions. However, it is beyond the scope of this initial study of the ejector for high pressure hydrogen. These matters are to be conducted

in future.

5. Conclusions

A new concept is proposed for hydrogen fuelling stations, in which an ejector is integrated into the fuelling process to improve the efficient utilization of the high-pressure hydrogen. The computational fluid dynamic modelling is validated and employed to evaluate the performance of the hydrogen ejector. The hydrogen ejector shows a good entrainment capacity when it works in the critical process. The entrainment ratio declines when it goes into the sub-critical process. The reverse flow occurs as the back pressure and suction pressure reach a critical value, which should be optimised in future studies by investigating the internal ejector geometry. The maximum back pressure increases from 345 bar to 480 when increasing the primary nozzles pressure from 450 bar to 900 bar and keeping the suction chamber pressure at 300 bar. The study shows that the potential mass which can be entrained from the low-pressure storage is 0.7 kg corresponding to 10 % of the total mass, this potential could increase for other fuelling station pressures designs. This study shows promising results for using an ejector in hydrogen fuelling stations to optimise energy consumption. Further studies should be conducted to investigate the behaviour of the ejector during dynamic conditions to reflect a real hydrogen fuelling.

Conflicts of interest

There are no conflicts to declare.

Acknowledgements

This paper is a part of the research project: EUDP17-II: HyScale. The authors

would like to thank the Danish Energy Agency for financial support and our industrial partner, NEL Hydrogen, for their collaboration and technical support.

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