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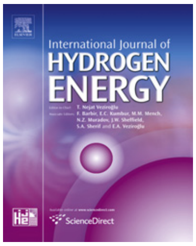

Effects of storage types and conditions on compressed hydrogen fuelling stations performance

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ABSTRACT

In hydrogen fuelling stations hydrogen is usually stored in the high-pressure buffer or cascade storage systems. Buffer storage system includes a single pressure reservoir, while the cascade storage system is usually divided into three reservoirs at low, medium and high-pressure levels. In the present study, first and second laws of thermodynamics have been employed to analyze the filling process associated with these two storage systems. The important parameters such as filling time, filled mass and compressor input work have been examined in detail. Assuming the same final vehicle on-board in-cylinder pressure for both storage systems, the results reveal that filling time of the buffer storage system is much less than the cascade storage system. However, the filled mass related to the buffer system for the same conditions is approximately equal of the cascade system. Furthermore, the buffer system is accompanied with much higher entropy generation as compared to the cascade storage system, which directly reflects in the amount of required compressor input work. Entropy generation minimization has also been employed to determine the optimized low and medium-pressure reservoir pressures for the cascade storage system, which corresponds to the lowest required compressor input work for a specific high-pressure reservoir in the cascade systems.

1. Introduction

The use of hydrogen as a clean alternative to common fuels such as gasoline and diesel in automobiles has a positive impact on environment [1]. Due to the recent developments in hydrogen fuel technology the expansion of hydrogen fuelling stations has gained more attention in the world [2,3]. Previous studies indicate that the high-pressure compressed hydrogen storage has the advantage of being more practical, dependable, durable and admissible [4–6] as compared to the liquid hydrogen storage. According to statistics, about 80–90% of hydrogen is stored using high-pressure compression in hydrogen fuelling stations and vehicle cylinders [7].

Although there are many advantages in using hydrogen, yet it has not been widely accepted as an alternative fuel to gasoline due to the low driving range of hydrogen vehicles, which is partly associated with the hydrogen fuelling stations technology.

The hydrogen vehicles commonly receive hydrogen from high-pressure reservoirs at the fuelling stations during filling.

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Clearly, fuelling time is a matter of practical importance. Fortunately, the hydrogen vehicle industry has made excellent progress in providing hydrogen refuelling systems comparable to that of gasoline dispensers, such that the problem with the long refuelling time has been basically resolved. The fill time of the fast fill or rapid charge systems is basically comparable to the fill time of gasoline powered automobiles.

The on-board storage capacity of the hydrogen vehicles is another issue related to the wide spread marketing of these vehicles. The on-board storage cylinder experiences a rise in the gas temperature (in the range of 70 K or more [8]) during the fast filling due to the compression and mixing processes. This temperature rise reduces the density of the gas in the cylinder resulting in an under-filled cylinder relative to its rated specifications, which may require a transient over-pressurizing the tank. Although hydrogen on-board cylinder volume plays the main role in storage capacity, the pressure of the fuelling station reservoirs has also considerable effect of the amount of the filled mass.

There are two methods for storing hydrogen fuel in the stations including buffer storage and cascade storage systems. In the buffer storage system, there is only one single high-pressure reservoir for storing hydrogen. The cascade storage system is usually divided into three reservoirs, which are referred to as low, medium and high-pressure reservoirs.

It is expected that storage type and its pressure influence the filling time, the amount of the filled mass of the on-board cylinder and the compressor input work. Any improvements in reducing the filling time and the compressor input work, while increasing the filled mass of the on-board cylinder leads to the enhancement of the fuelling station performance.

The effects of storage types and conditions on the performance of a hydrogen fuelling station have been considered in the present study. A theoretical analysis has been performed based on the first and second laws of thermodynamics, conversation of mass and real gas assumptions. The second law analysis has been employed to calculate the amount of entropy generation during the filling process. It is well known that the required compressor work can be optimized by minimizing the related entropy generations. To the authors' best knowledge, there have been no previous studies related to the performance enhancement of the hydrogen fuelling stations. However, there are limited researches available regarding the modelling of filling process in literature.

Yang [9] has developed a thermodynamic and heat transfer analysis of the refuelling process of a hydrogen on-board cylinder. During the refilling process, the cylinder is treated as adiabatic, isothermal, or diathermal. Ideal and real gas behaviours of hydrogen are considered in the analysis. Non-ideality is treated using the newly developed equation of state for normal hydrogen, which is based on the reduced Helmholtz free energy formulation. With the ideal-gas assumption, simple analytical expressions were derived for the tank temperature and pressure during adiabatic, isothermal, and diathermal refuelling conditions. A constant feed-rate is assumed in this study. Comparing to the real gas analysis, lower tank temperatures and pressures and longer filling times are always predicted, when the ideal-gas assumption is invoked in the calculations irrespective of the refilling conditions. In another paper, Mohamed and Paraschivoiu [10] simulated hydrogen release from a high-pressure chamber based on real gas assumption.

Zheng et al. [11] modelled an optimizing control method for a high utilization ratio and fast filling speed in hydrogen fuelling stations. It was shown that the optimizing control method can significantly improve the utilization ratio, while allowing for acceptable refuelling time. Liss and Richards [12], Liss et al. [13], Newhouse and Liss [14], Chan Kim et al. [15] and Liu et al. [16] have studied fast filling of hydrogen cylinder using a number of experiments. They reported a high temperature increase in the cylinder during the process. For the hydrogen fast filling process, Chan Kim et al. [15] have studied thermal characteristics during the filling of a type IV cylinder using computational fluid dynamics (CFD) analysis. The predicted results show reasonable agreements with the experiments especially as the initial in-cylinder pressure increases. Similar CFD analysis has been carried out by Heitsch et al. [17], where fast filling process of hydrogen tanks is simulated using the CFD code CFX. It was found that the local temperature distribution in the tank depends on the materials of liner and outer thermal insulation. Different material combinations (type III and IV) are investigated.

Due to the similarity between hydrogen and the Compressed Natural Gas (CNG) infrastructures, it is also instructive to review the comparable works on CNG. Thomas and Goulding [18] and Shiply [19] have studied filling process of on-board CNG cylinders. They reported a rise in storage gas cylinder temperature (in the range of 40 K or more) during the process. This temperature rise reduces the density of the gas in the cylinder, resulting in an under-filled cylinder, relative to its rated specification. Shiply [19] also mentioned that ambient temperature could affect the process and storage capacity significantly. He also concluded that, the test cylinder was under-filled every time it was rapidly recharged.

Farzaneh-Gord et al. [20,21] have modelled fast filling process in the CNG stations. They developed a computer program based on Peng-Robinson state equation and methane properties table for single reservoir. They investigated the effects of ambient temperature and initial cylinder pressure on final on-board cylinder conditions. In another study, Farzaneh-Gord et al. [22] presented a thermodynamic analysis of cascade reservoirs filling process, which indicated that ambient temperature has a considerable effect on filling process and final Natural Gas Vehicle (NGV) on-board cylinder conditions.

Farzaneh-Gord et al. [23] have also carried out a theoretical analysis to study the effects of storage type on the performance of CNG filling stations and filling process. It was found that the filling time required reaching the NGV on-board cylinder to its final pressure (20 MPa) in the buffer storage system is about 66% less than that for the cascade storage systems. While the filled mass for the cascade system is about 80% of the buffer system, which is an advantage for the buffer type. Furthermore, it was shown that the entropy generation associated with the cascade system is 50% less than the buffer system for the considered configuration, which directly reflects in the required input power.

As mentioned earlier, the second law has been employed to theoretically calculate the amount of the entropy generation in the present study. Different mechanisms responsible for entropy generation in applied thermal engineering have been identified and well described in literature [24–26]. Generation of entropy destroys the available work of a system; therefore, all the irreversibilities associated with the heat transfer and fluid flow must be examined. Here, entropy generation minimization has been employed as the main tool to determine the optimized pressure level of the low and medium-pressure reservoirs.

2. Hydrogen fuelling station

Fig. 1 shows a typical hydrogen fuelling station, where hydrogen is compressed using a big multi-stage compressor and collect in a storage system [27]. The storage system consists of several large cylinders, which are available in a variety of sizes, typically from 50 L internal capacity to well over 100 L. This system is maintained at a pressure higher than the vehicle cylinder pressure for a continuous gas flow to the vehicle storage. There are two types of storing systems in hydrogen stations commonly called buffer and cascade storage systems, which are described in the following sections.

2.1. *Buffer storage system*

The buffer storage operates in the range of 37 MPa–70 MPa, while the vehicle's maximum on-board cylinder pressure is about 35 MPa. In this type of the storage system, all fuel reservoir cylinders are connected together as shown in Fig. 2 and maintained at the same pressure all the times. In the present study, the reservoir temperature and pressure are assumed to be equal to 300 K and 37 MPa, respectively, unless otherwise stated.

2.2. *Cascade storage system*

The cascade storage system is usually divided into three reservoirs, which are low, medium and high-pressure reservoirs. Each of these reservoirs consists of several large cylinders. In this storage system, reservoir cylinders are put into an order of ascending pressure. Fig. 3 shows a schematic diagram of a cascade storage system.

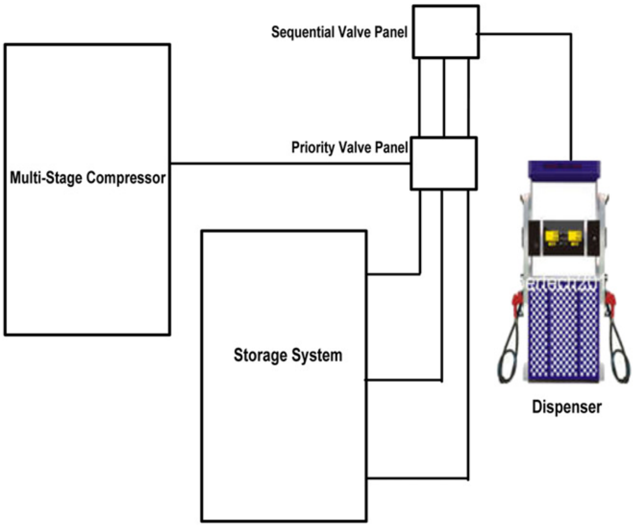

Fig. 1 – A schematic diagram of hydrogen fuelling station.

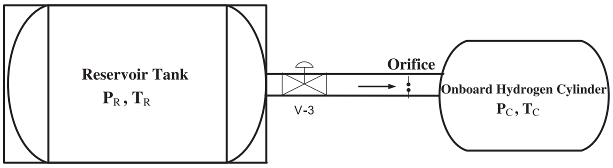

Fig. 2 – A schematic diagram of the buffer storage system.

During filling, the hydrogen on-board cylinder is first connected to the low-pressure reservoir. As the flow rate reaches a pre-set level the system is first switched to the medium-pressure reservoir, and then to the high-pressure reservoir to complete the fill. However, in refilling the station reservoirs the compressor is automatically switched on to fill the high-pressure reservoir first, and then switches to the medium and the low-pressure reservoirs. This ensures that the high-pressure reservoir is maintained at maximum pressure all the times, which in turn guarantees that vehicles are always supplied with the maximum amount of gas available. Correct specification of the compressor capacity and the volume of cascade storage is necessary to ensure that the hydrogen station can deal with the type (buses or trucks) and frequency (peak periods) of vehicles using the facility.

3. Cascade reservoirs parameters

Thermodynamic properties in the cascade reservoirs play important roles on the filling process. Two main properties are pressure and temperature. As shown in Fig. 3, each reservoir has its own temperature (T_R) and pressure (P_R), which are assumed to be constant during the filling process. Typically, the maximum pressure in the high-pressure reservoir (P_{R3}) is in the range of 37 MPa–70 MPa, while the vehicle's maximum on-board cylinder pressure is about 35 MPa.

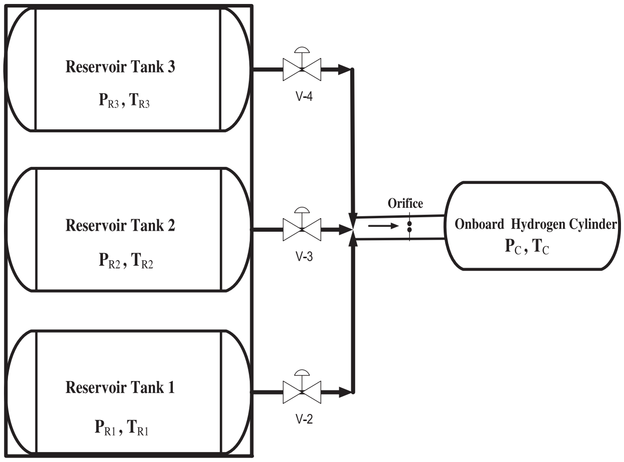

Fig. 3 – A schematic diagram of the cascade storage system.

To maintain the final pressure within the on-board cylinder to its common maximum level, the pressure within the high-pressure reservoir is assumed constant at 37 MPa throughout the present study. The effects of medium and low reservoir pressures on the performance of the fuelling station have been studied by introducing two dimensionless parameters. The ratios of the medium and low-pressure reservoir to the high-pressure, which are defined as:

$$NP1 = P_{R1}/P_{R3} \quad NP2 = P_{R2}/P_{R3} \tag{1}$$

An additional dimensionless number is the fill ratio, which is defined as the mass of charged gas after filling divided by the mass which the on-board cylinder can hold at the rating condition of 300 K, 35 MPa. This parameter is directly related to the driving range of the hydrogen vehicles:

$$FR = \frac{m_c(\text{at end of filling})}{\rho(300\text{K}, 35\text{MPa})V_c} \tag{2}$$

Filling algorithms for the cascade reservoirs is as follows. The on-board cylinder is first connected to the low-pressure reservoir. The cylinder is switched to the medium and then high-pressure reservoir when the pressure difference between the in-cylinder and the reservoir drops to 0.5 Mpa. Filling is stopped, when the on-board cylinder pressure reaches to 35 MPa.

4. Thermodynamic analysis

4.1. *First law analysis*

To model the fast filling process and construct a mathematical method, the hydrogen on-board cylinder is considered as a thermodynamic open system which goes through a quasi-steady process. By applying the continuity and first law of thermodynamics to the on-board hydrogen cylinder as a control volume containing only one inlet the followings are obtained:

$$\frac{dm_C}{dt} = \dot{m}_i \tag{3}$$

In Eq. (3), $\dot{m}_i$ is the inlet mass flow rate, which can be calculated by considering an expansion through an orifice [28]

$$\dot{m}_i = C_d\rho_R A_{orifice}\left(\frac{P_C}{P_R}\right)^{\frac{1}{\gamma}}\left\{\left(\frac{2\gamma}{\gamma-1}\right)\left(\frac{P_R}{\rho_R}\right)\left[1-\left(\frac{P_C}{P_R}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{\frac{1}{2}} \tag{4}$$

for

$$P_C/P_R \le \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$$

$$\dot{m}_i = C_d\sqrt{\gamma P_R\rho_R}A_{orifice}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \tag{5}$$

for

$$P_C/P_R > \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$$

In Eqns. (4) and (5), C_d is discharging coefficient of the orifice and stands for irreversibility. For reversible process, $C_d = 1$.

The first law of thermodynamics for a control volume can be written as follows:

$$\begin{aligned}\dot{Q}_{cv} + \sum\dot{m}_i\left(h_i + V_i^2/2 + gz_i\right) = \sum\dot{m}_e\left(h_e + V_e^2/2 + gz_e\right)\\ + d/dt\left[m\left(u + V^2/2 + gz\right)\right]_{cv}\\ + \dot{W}_{cv}\end{aligned} \tag{6}$$

The work term is zero in the filling process and the change in potential energy can be neglected. The equation is simplified for the present case as:

$$\frac{dU_C}{dt} = \delta\dot{Q} + \dot{m}_i\left(\frac{V_i^2}{2} + h_i\right) \tag{7}$$

Defining $h_R = (V_i^2/2) + h_i$, the above equation can further be simplified as:

$$\frac{dU_C}{dt} = \delta\dot{Q} + \dot{m}_i h_R \tag{8}$$

The heat released from the on-board hydrogen cylinder to the environment is calculated according to:

$$\delta\dot{Q} = -U_{HC}A_C(T_C - T_\infty) \tag{9}$$

U_{HC}, A_C, T_C, and T_∞ are the overall heat transfer coefficient, cylinder surface area, in-cylinder temperature and ambient temperature, respectively. Combining Eqns. (3), (8), and (9), the following equation is obtained:

$$\frac{d(m_C u_C)}{dt} = -U_{HC}AC(T_C - T_\infty) + \frac{dm_C}{dt}h_R \tag{10}$$

Or in the following form:

$$\frac{d(m_C u_C)}{dt} - \frac{d(m_C h_R)}{dt} = -U_{HC}A_C(T_C - T_\infty) \tag{11}$$

The above equation can be rearranged in the following form:

$$d(m_C u_C - m_C h_R) = -U_{HC}A_C(T_C - T_\infty)dt \tag{12}$$

The above equation can be integrated from “start” of filling, up to the “present” time as:

$$\int_s^p d(m_C u_C - m_C h_R) = -\int_o^t U_{HC}A_C(T_C - T_\infty)dt \tag{13}$$

The integration of the above equation for a single reservoir fuelling station resulted to:

$$m_C(u_C - h_R) - m_{Cs}(u_{Cs} - h_R) = -U_{HC}A_C\Delta T_{av}t \tag{14}$$

where m_C and m_{Cs} are the mass of charged gas at “present” and “start” of the filling process, respectively. ΔT_{av} is the average temperature difference between the cylinder and environment defined as:

$$\Delta T_{av} = \frac{1}{t}\int_o^t (T_C - T_\infty)dt \tag{15}$$

The first law of thermodynamic for the on-board hydrogen cylinder finally is written as:

$$u_C = h_R - U_{Hc}A_C\Delta T_{av}t + \frac{m_{Cs}}{m_C}(u_{Cs} - h_R) \tag{16}$$

4.1.1. *Adiabatic system*

For an adiabatic system, the Eq. (16) can be further simplified to:

$$u_C = h_R + \frac{m_{Cs}}{m_C}(u_{Cs} - h_R) \tag{17}$$

And if $m_{Cs} = 0$, the following relation is valid at any time:

$$u_C = h_R \tag{18}$$

4.2. *The second law analysis*

The second law of thermodynamics for filling process of an on-board hydrogen cylinder is presented as:

$$\dot{S}_{gen} = \frac{dS_C}{dt} - \frac{\delta\dot{Q}}{T_\infty} - \dot{m}_i s_i \geq 0 \tag{19}$$

It is further assumed that all irreversibilities occur from the inlet to in-cylinder position. Therefore, an isentropic expansion is considered from the reservoir to inlet position, that is $s_i = s_R$. In view of this assumption and combining Eqns. (3), (9), and (19), the following equation is obtained:

$$\dot{S}_{gen} = \frac{d(m_C s_C)}{dt} - \frac{dm_C}{dt}s_R + \frac{U_{Hc}A_C(T_C - T_\infty)}{T_\infty} \tag{20}$$

Or rearranging as:

$$\dot{S}_{gen}dt = d(m_C s_C - m_C s_R) + \frac{U_{HC}A_C(T_C - T_\infty)}{T_\infty}dt \tag{21}$$

The above equation can be integrated from “start” of filling to the “present” time as:

$$S_{gen} = \int_s^p d(m_C s_C - m_C s_R) + \int_s^p \frac{U_{HC}A_C(T_C - T_\infty)}{T_\infty}dt \tag{22}$$

For a fuelling station with a single reservoir in which s_R remains constant throughout the filling process, the integration of the above equation resulted in the following simple equation:

$$S_{gen} = m_C(s_C - s_R) - m_{Cs}(s_{Cs} - s_R) + \frac{U_{HC}A_C(T_{av} - T_\infty)}{T_\infty} \tag{23}$$

4.2.1. *Adiabatic system*

Eq. (23) can be more simplified for an adiabatic system as:

$$S_{gen} = m_C(s_C - s_R) - m_{Cs}(s_{Cs} - s_R) \tag{24}$$

If the cylinder is empty at the start of filling process ($m_{cs} = 0$), the following relation is obtained:

$$S_{gen,\max} = m_C(s_C - s_R) \tag{25}$$

It should be noted that Eqns. (23)–(25) are only valid for a single reservoir fuelling station. Calculating entropy generation for a fuelling station with the cascade reservoirs system is more complex as reservoirs conditions vary during filling. Here the non-dimensional entropy generation is introduced to allow for comparing the results for various configurations:

$$NS = \frac{S_{gen}}{S_{gen,\max}} \tag{26}$$

It worth mentioning that *NS* expresses the irreversibility in the system. Minimizing *NS* means reducing the required input work to the system, which is provided by the station compressor.

4.3. *The numerical procedure*

The procedure for calculating the in-cylinder condition of hydrogen starts from the initial conditions. Eq. (4) or (5) is employed to calculate the inlet mass flow rate. Eq. (3) is then utilized to compute the in-cylinder mass and consequently specific volume of hydrogen within the cylinder using first order Euler numerical scheme. Similarly, Eq. (16) is solved to calculate the in-cylinder specific internal energy of hydrogen at the new time step. Upon determination of two independent thermodynamic properties (here specific internal energy and specific volume), other properties can be easily found from the hydrogen property tables.

5. Results and discussion

In this study, an adiabatic hydrogen vehicle on-board cylinder is considered and therefore, the characteristics of the orifice, do not affect the final temperature in the cylinder. The orifice diameter and the cylinder volume are considered to be 1 mm and 150 litters [11], respectively.

Initially, the effects of storage type (buffer and cascade storage systems) on hydrogen filling performance are examined. In this regard the dynamic profile of in-cylinder properties and the important parameters affecting the filling station performance are discussed in Section 5.1. The reservoir pressure of 37 MPa is considered for buffer system. While for the case of cascade storage system, the pressures of 11, 21 and 37 MPa are considered for the low, medium and high-pressure reservoirs, respectively. The optimized pressure values for the low and medium- pressure reservoirs are established for cascade storage system through minimization of the entropy generation of the system in Section 5.2.

5.1. *Comparison between buffer and cascade storage systems*

Fig. 4 shows the dynamic in-cylinder pressure profiles during filling process for buffer and cascade systems. It is clear that the time required to reach the final pressure (35 MPa) in the buffer storage system is about 66% less than the cascade storage system. It should be noted that the filling time could also be reduced by appropriate sizing of the piping equipments (e.g. orifice diameter).

Fig. 5 compares the time variations of the mass flow rate during the filling process for buffer and cascade systems in the constant ambient temperature of 300 K. Figure shows that the mass flow rate profile in the cascade system is not continuous and it contains three separate sections corresponding to the times that the low, medium and high-pressure reservoirs are switched on. While the time variations of the mass flow rate

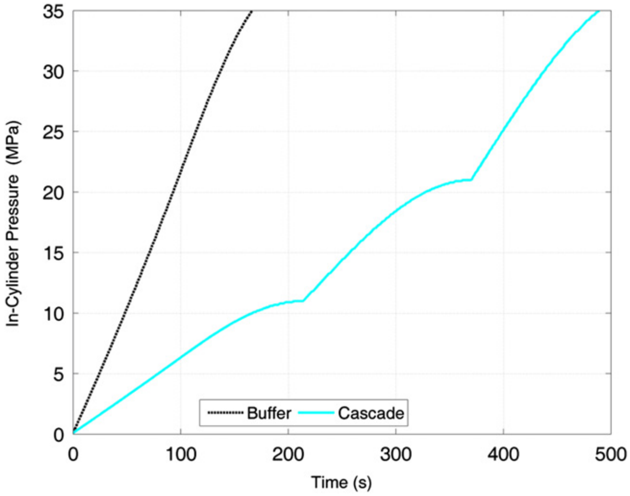

Fig. 4 – The time variation of in-cylinder pressure profiles during filling process for buffer and cascade storage systems.

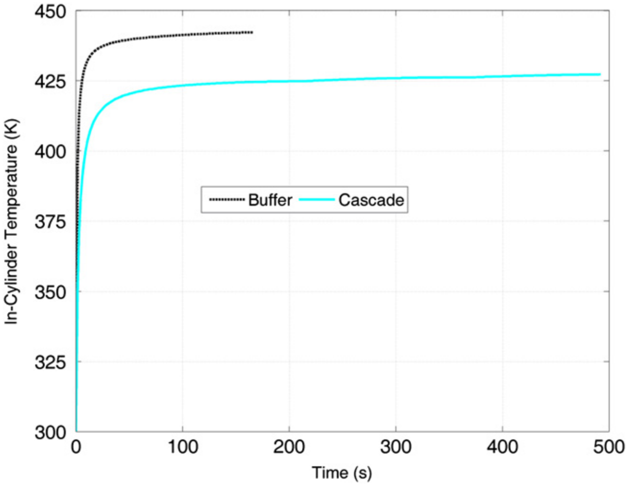

Fig. 6 – The time variations of the dynamic hydrogen in-cylinder temperature profiles for buffer and cascade storage systems.

for the buffer system reveals a similar trend to the first section of the cascade profile. In the early times of the filling process the mass flow rate is constant due to the choking condition of orifice. Clearly, the mass flow rate for the buffer system is much higher than the cascade system, which makes the filling time for buffer system to be much smaller than the cascade system.

Fig. 6 shows the time variations of the dynamic hydrogen in-cylinder temperature profiles for both cases. Both storing systems show a rapid increase in temperature from the ambient temperature in the early stages of charging, while the temperature remains basically constant during the rest of the filling process. However, the final in-cylinder temperature for buffer system is about 15 K higher than the cascade storage system.

Farzaneh-Gord et al. [20–23] have also presented similar results for CNG filling process. In their results, the in-cylinder temperature in buffer system drops during the early stages of filling before rising to its final value. It was found the final temperature of the buffer system is about 20 K less than the cascade system due to this initial drop in temperature. It was explained that the reason for the initial drop in temperature, is because of the Joule-Thompson cooling effect, since the gas undergoes an isenthalpic expansion through the orifice from 20 MPa supply pressure to the initially low cylinder pressure of 0.1 MPa. The expanded cold gas mixes with the existing gas in the tank lowering the gas temperature initially. However, later on the compression and conversion of the flow work into the internal energy overcomes the Joule-Thompson cooling effect and the cylinder gas temperature increases.

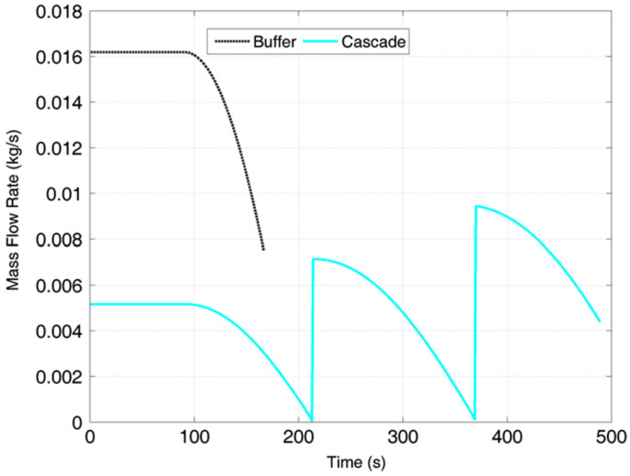

Fig. 5 – The time variations of the mass flow rate during the filling process for buffer and cascade storage systems.

It must be noted that for hydrogen filling process, the Joule-Thompson coefficient is negative and therefore, the gas temperature increases during an isenthalpic filling expansion. This causes the initial temperature rise to be higher for the buffer system as compared to the cascade one.

The results presented in this section are valid for an adiabatic hydrogen vehicle on-board cylinder. In practice, the cylinders are not adiabatic due to the heat lost during the filling process. This makes final the in-cylinder temperature to be lower than the values reported here. The actual final in-cylinder temperature depends on the charging time but for safety considerations it should be lower than 85 °C. The charging time is controlled by the station dispenser algorithm through regulating inlet mass flow rate in actual condition.

In Fig. 7, the time variations of the filled mass are shown for buffer and cascade systems during filling process for initial temperature of 300 K. Clearly, the filled mass for the cascade system is only 0.1 kg more than the buffer system. In the cascade system, about 35%, 29% and 36% of filled mass are supplied by low, medium and high-pressure reservoirs, respectively. For a CNG filling station, Farzaneh-Gord et al. [23] found that the filled mass is higher for the buffer system instead of the cascade system, which is directly related to the final in-cylinder gas temperature. The filled mass is higher for

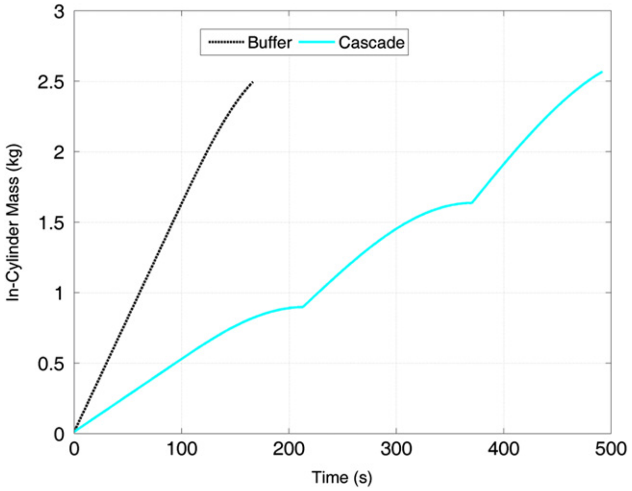

Fig. 7 – The time variations of the in-cylinder mass during filling process for buffer and cascade storage systems.

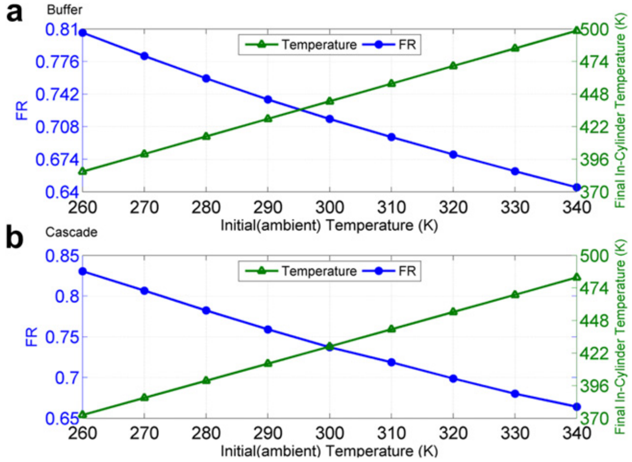

Fig. 9 – Effect of initial (ambient) temperature on fill ratio and final in-cylinder temperature for a) buffer and b) cascade storage systems.

the case where the final in-cylinder temperature is lower. The final in-cylinder temperature itself is related to the joule-Thomson effects.

Fig. 8 shows the effect of initial or the ambient temperature variations on the filled mass of an empty hydrogen on-board cylinder. As mentioned earlier, the filled mass has a direct impact on the driving range of a hydrogen vehicle, which is one of most important problems associated with hydrogen vehicles industry. As figure shows, the filled mass decreases for both systems as the ambient temperature increases, which is slightly larger for the cascade system.

Fig. 9 shows how fill ratio and final in-cylinder temperature vary with ambient temperature. It is seen that as the initial or ambient temperature increases fill ratio decreases. This means that driving range of a hydrogen vehicle will decrease in hot weather condition as compared to the colder weather. Similar conclusion can be made by considering the effects of ambient temperature on in-cylinder temperature variations. Figure also shows that the final in-cylinder temperature increase as ambient temperature increases. Yang [9] has carried out a theoretical analysis and calculated the final in-cylinder temperature to initial temperature ratio for a buffer system. The ratio was found to be 1.48 as compared to the present study results, which vary between 1.467 and 1.485 depending on the initial temperature.

As mentioned previously, entropy generation is associated with thermodynamic irreversibilities, which waste the available work provided by the compressor in the fuelling station. Therefore, any reduction in the entropy generation during the filling process is associated with the decrease in compressor input work. Fig. 10 compares the entropy generation for the buffer and cascade storage systems, which clearly indicates that the entropy generation for the buffer system is about two times larger than the cascade system. This can be justified by the lower work required to store the gas in cascade system comparing to the buffer one.

Fig. 11 shows a normalized comparison among important parameters for hydrogen vehicle industries. The normalized values in this figure are obtained based on the higher value in each case. Considering the important factors associated with filling process and fuelling station that are filling time, filled mass and compressor input work, it can be concluded that the

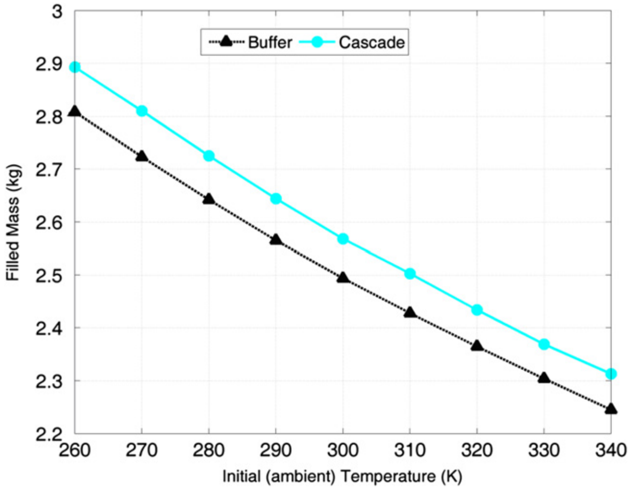

Fig. 8 – Effect of initial (ambient) temperature on filled mass for buffer and cascade storage systems.

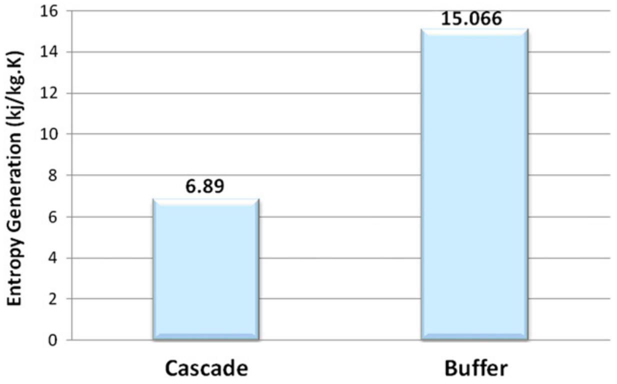

Fig. 10 – Entropy generations for buffer and cascade storage systems.

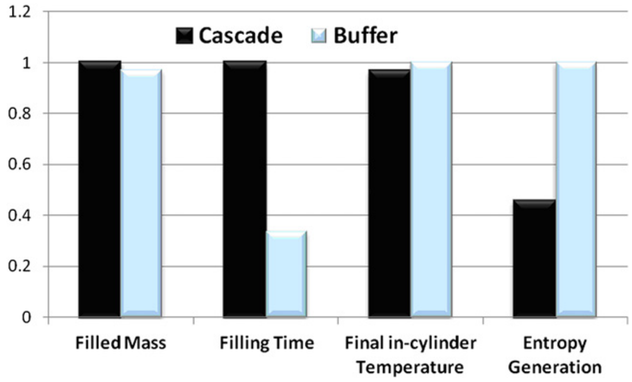

Fig. 11 – **Dimensionless comparisons among important parameters in hydrogen station for buffer and cascade storage systems.**

filled mass is slightly higher and entropy generation is lower for the cascade system compared to the buffer system. The main disadvantage of the cascade system is the considerable larger filling time. However, the filling time can be also be reduced by appropriate sizing of piping equipments (e.g. orifice diameter), and therefore, the cascade storage system may be considered as the more appropriate configuration for the storage system.

As discussed in Section 3 for cascade system, the cylinder is switched to the higher pressure reservoir when the in-cylinder and the reservoir pressure difference drops to 0.5 Mpa. If the switching occurs at a higher pressure difference, the inlet mass flow rate increases and consequently the filling time decreases. It will probably have also impact on the final in-cylinder condition. The future study should investigate these effects.

It should be also noted that in most of the hydrogen fuelling stations with cascade configuration, the pressure of the low and medium-pressure reservoirs are brought up to the highest possible pressure (even up to the pressure of the high-pressure reservoir) when the compressor has free time. In such cases, the fuelling station behaviour is similar to the buffer system.

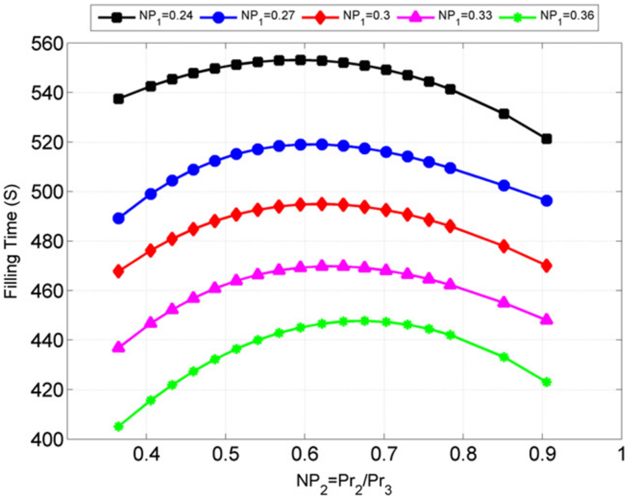

Fig. 12 – **Effects of varying low-pressure (NP1) and medium-pressure reservoir pressure (*NP*2) on filling time for cascade storage system.**

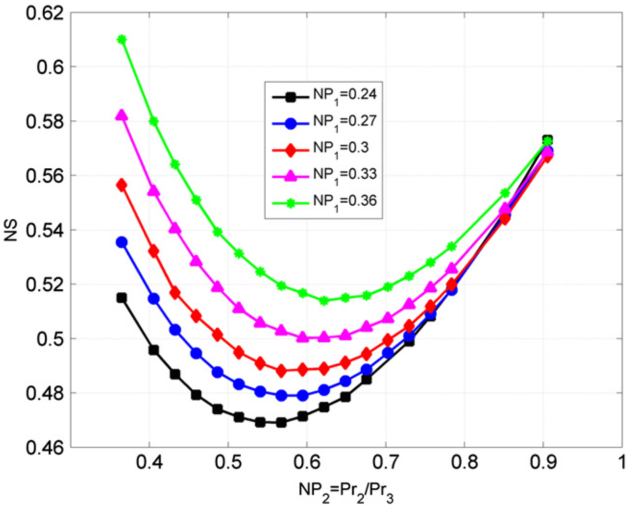

Fig. 13 – **Effect of varying low-pressure (*NP*1) and medium-pressure reservoir pressure (*NP*2) on non-dimension entropy generation for cascade storage system.**

5.2. *Optimizing cascade storage system*

As indicated above, the cascade storage system shows a better performance over the buffer systems. The next question would be about the optimized values for the pressure of the low and medium-pressure reservoir. The optimised values are determined based on the smaller filling time and lower generated entropy as discussed below.

Fig. 12 shows the effects of pressure levels of the low and medium-pressure reservoir on the filling time. The objective is finding a combination of *NP*1 and *NP*2 in which the filling time is minimized. Note from the figure that filling time decreases as *NP*1 increases. Furthermore, for any value of *NP*1 there is a specific value of *NP*2 in which the filling time is maximized. Apparently, it can be concluded that such a combination of *NP*1 and *NP*2 that minimizes the filling time cannot be obtained. Yet as mentioned earlier, it is also possible to reduce the filling time through appropriate sizing of the piping equipments (e.g. orifice diameter).

As for the entropy generation, in Fig. 13 the effects of the pressure level of the low and medium-pressure reservoirs on the normalized entropy generation are presented. It must be emphasized that it is assumed that the cylinder is empty initially. Figure indicates that the minimum entropy generation for the present case occurs for the pressure ratios of *NP*1 ≈ 0.24 and *NP*2 ≈ 0.55 corresponding to the pressure values of 8.88 MPa and 20.35 MPa for the low and medium-pressure reservoirs.

Comparing Fig. 12 with 13, it can be realized that the normalized entropy generation and filling time profiles show opposite trends, such that as the entropy generation decreases the filling time increases. Therefore, as expected the minimum entropy generation corresponds to the maximum filling time for any pressure ratio.

6. Conclusion

First and second laws of thermodynamics have been employed to analyze the filling process associated with the buffer and cascade storage systems. The important parameters such as filling time, filled mass and compressor input work have been examined for the these two storage systems. It is found that the time required to bring up the hydrogen vehicle on-board cylinder to its final pressure of 35 MPa in the buffer storage system is about 66% less than that of the cascade storage system. However, the filled mass related to the buffer system for the same conditions is about 97% of the cascade system which is not a major disadvantage of this system. Furthermore, buffer storage system is accompanied with 55% higher entropy generation as compared to the cascade storage system, which directly reflects in the amount of required compressor input work. Since the filling time can be reduced through appropriate sizing of the piping equipments, it seems that the cascade storage system is more promising in the hydrogen fuelling stations technology.

Entropy generation minimization has also been employed to determine the optimized low and medium-pressure reservoir pressures for the cascade storage system, which corresponds to the lowest required compressor input work for a specific high-pressure reservoir.

Some limitations associated with the present study comes from the employed simplifying assumptions such as adiabatic condition for the on-board cylinder and fixed conditions for reservoirs, which will be relaxed in our future work.

REFERENCES

[1] White CM, Steeper RR, Lutz AE. The hydrogen-fuelled internal combustion engine: a technical review. International Journal of Hydrogen Energy 2006;31:1292–305.

[2] Rigas F, Sklavounos S. Evaluation of hazards associated with hydrogen storage facilities. International Journal of Hydrogen Energy 2005;30:1501–10.

[3] Schoenung S, Ridell B, Maack M, Miles S, His S. A comparative study of hydrogen refuelling station experience. In: World hydrogen energy conference; 2006.

[4] Tzimas E, Filiou C. Hydrogen storage: state-of-the-art and future perspective. Petten: European Communities; 2003.

[5] Zheng JY, Li L, Chen R. High pressure steel storage vessels used in hydrogen refuelling station. Journal of Pressure Vessel Technology 2008;130:1–3.

[6] Maus S, Hapke J, Na Ranong Ch, Wuchner E, Friedlmeier G, Wenger D. Filling procedure for vehicles with compressed hydrogen tanks. International Journal of Hydrogen Energy 2008;33:4612–21.

[7] Zhang YJ, Mao ZQ, Xie XF. Research and application of regenerative fuel cells. Progress in Chemistry 2006;18(5): 635–40.

[8] Dicken CJB, M'erida W. Measured effects of filling time and initial mass on the temperature distribution within a hydrogen cylinder during refuelling. Journal of Power Sources 2007;165:324–36.

[9] Yang JC. A thermodynamic analysis of refuelling of a hydrogen tank. International Journal of Hydrogen Energy 2009;34:6712–21.

[10] Mohamed K, Paraschivoiu M. Real gas simulation of hydrogen release from a high-pressure chamber. International Journal of Hydrogen Energy 2005;30(8): 903–12.

[11] Zheng J, Ye J, Yanga J, Tang P, Zhao L, Kern M. An optimized control method for a high utilization ratio and fast filling speed in hydrogen refuelling stations. International Journal of Hydrogen Energy 2010;35:3011–7.

[12] Liss WE, Richards M. Development of a natural gas to hydrogen fueling station. Topical Report for U.S. DOE, GTI-02/0193; Sept., 2002.

[13] Liss WE, Richards ME, Kountz K, Kriha K. Modelling and testing of fast-fill control algorithms for hydrogen fuelling. 2003 National Hydrogen Association Meeting, March, 2003.

[14] Newhouse NL, Liss WE. Fast filling of NGV fuel containers. SAE paper 1999-01-3739.

[15] Chan Kim S, Hoon Lee S, Bong Yoon K. Thermal characteristics during hydrogen fuelling process of type IV cylinder. International Journal of Hydrogen Energy 2010;35:6830–5.

[16] Liu YL, Zhao YZ, Zhao L, Li X, Chen HG, Zhang LF, et al. Experimental studies on temperature rise within a hydrogen cylinder during refuelling. International Journal of Hydrogen Energy 2010;35:2627–32.

[17] Heitsch M, Baraldi D, Moretto P. Numerical investigations on the fast filling of hydrogen tanks. International Journal of Hydrogen Energy 2011;36:2606–12.

[18] Thomas G, Goulding J, Munteam C. Measurement approval and verification of CNG dispensers. NWML KT11 Report; 2002.

[19] Shipley E. Study of natural gas vehicles during the fast fills process. Thesis for Master of Science, 2002. College of Engineering and Mineral Resources at West Virginia University.

[20] Farzaneh-Gord M, Eftekhari H, Hashemi S, Magrebi M, Dorafshan M. The effect of initial conditions on filling process of CNG cylinders. The second International conference on Modelling, Simulation, And Applied optimization, Abu Dhabi, UAE, March 24–27 2007.

[21] Farzaneh-Gord M. Compressed natural gas single reservoir filling process. Gas International Engineering and Management July/August 2008;48(6):16–8.

[22] Farzaneh-Gord M, Hashemi SH, Farzaneh-Kord A. Thermodynamics analysis of cascade reservoirs filling process of natural gas vehicle cylinders. World Applied Sciences Journal 2008;5(2):143–9.

[23] Farzaneh-Gord M, Deymi Dasht-bayaz M, Rahbari HR. Studying effects of storage types on performance of CNG filling stations. Journal of Natural Gas Science and Engineering 2011;3:334–40.

[24] Bejan A. Second-law analysis in heat transfer and thermal design. Advanced Heat Transfer 1982;15:1–58.

[25] Bejan A. Entropy generation minimization. Boca Raton, NY: CRC; 1996.

[26] Bejan A. A study of entropy generation in fundamental convective heat transfer. Journal of Heat Transfer 1979;101: 718–25.

[27] Baur S, Sheffield J, Enke D. First annual university students' hydrogen design contest: hydrogen fuelling station. National Hydrogen Association, U.S. Department of Energy and Chevron Texaco; 2004.

[28] Oosthuizen PH, Carscallen WE. Compressible fluid flow. McGraw-Hill; 1997.

Glossary

c_p,c_v: Constant pressure &volume specific heats, kj/kg K
g: Gravitational acceleration, m/s^2
h: Specific enthalpy, kj/kg

$\dot{m}$: Mass flow rate, kg/s
u: Internal energy, kj/kg
h: Enthalpy, kj/kg
t: time, seconds
s: Entropy, kj/K
$\dot{s}$: Entropy Rate, kj/Ks
v: Specific volume, m^3/kg
z: Height, m
A: area, m^2
C_d: orifice discharge coefficient
M: Molecular weight, kg/kmol
P: Pressure, bar or Pa
$\dot{Q}$: Heat transfer rate, kW
T: Temperature, K or °C
V: Volume, m^3
W: Actual work, kj/kg
$\dot{W}$: Actual work rate, kW or MW
NP: Non-dimensional pressure ratio
NS: Non-dimensional entropy ratio
ρ: Density, kg/m^3
γ: Isentropic Exponent

Subscript

1: reservoir tank 1
2: reservoir tank 2
3: reservoir tank 3
i: initial or inlet condition
e: exit condition
max: maximum
p: present time of filling process
s: start of filling process
av: average
gen: generation
C: hydrogen on-board cylinder
CV: Control Volume
R: reservoir tank
HC: heat transfer coefficient
∞: ambient