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Thermodynamic analysis of a CNG refueling station considering the reciprocating compressor

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ABSTRACT

The reciprocating compressor is the heart of a CNG refueling station with dominating influence on refueling process. The main operating cost of a CNG refueling station arises from the compressor input work. The reciprocating compressor have not been subjected to previous studies which investigated the CNG refueling stations. The main objective of the present work is to apply a thermodynamic analysis for modeling a CNG refueling station, including on-board NGV cylinder, reservoir tanks, connecting pipes and especially reciprocating compressor. Unlike most previous researches, pressure variation of the reservoir tanks during the filling process have been considered which necessary for compressor modeling. The results of the present work have been validated against the previous experimental data and theoretical studies. The additional experimental data, measured by the present authors, is also used to analyze and identify the compressor performance in unsteady conditions. The simulation result of mass flow rate is a good agreement with the experimental values. The results show that variations in the compressor discharge pressure have no effect on the first and intermediate compressor pressure. It has also no effect on the compressor delivery mass flow rate. The energy consumption by the compressor for a cycle (including discharge and charge zones) is found to be 61.86 kWh. The average specific energy consumption to fill a vehicle is found to be 0.250 kWh/kg.

1. Introduction

Compressed natural gas (CNG) and compressed hydrogen gas (CHG) could be considered clean compared to petrol or diesel. Natural gas primarily consists of methane, but also contains other hydrocarbons such as butane, propane and ethane. CNG is stored in high-pressure cylinder on the vehicle (190–250 bar). Natural gas vehicles (NGVs) connect to the high-pressure reservoir tanks (200–235 bar) at refueling stations during filling process. Two main goals could be seen for refueling stations, operating costs at the refueling stations and the time to refuel a vehicle (filling time). The main operating cost of any CNG refueling station belongs to the compressor. So, a simulation to optimize the refueling process is very important. Filling time less than 5 min is referred as a fast fill. In these refueling stations, CNG is stored in reservoir tanks. There are two systems for storing CNG in reservoir tanks, cascade (multi-line) and buffer (mono-line) storage systems. Reservoir tanks in a cascade storage system are divided into three pressure levels, low, medium and high pressures. In a buffer system, reservoir tanks are only in one pressure level. A refueling station with a cascade storage system uses the priority panel to control the outlet flow of compressor to a series of cascade storage tanks (see Fig. 1). Based on the algorithm in the dispenser the on-board NGV cylinder is switched from lower pressure reservoir to a higher one, when the mass flow rate drops to a certain limit. There are many similarities between the CNG and CHG refueling process which makes the present work as a useful research for the CHG refueling stations.

There are various researches about CNG refueling station. These researches could be classified into 4 main groups as: a) the researches which only consider on-board NGV's cylinder, b) the researches which consider the reservoir tanks and on-board NGV's cylinder, c) the researches which consider the reservoir tanks, on-board NGV's cylinder and connection pipes and finally d) the researches which consider the reservoir tanks, on-board NGV's cylinder, connection pipes and reciprocating compressor.

Three have been many researches which only considered on-baord NGV's cylinder. In these researches, the reservoir tanks

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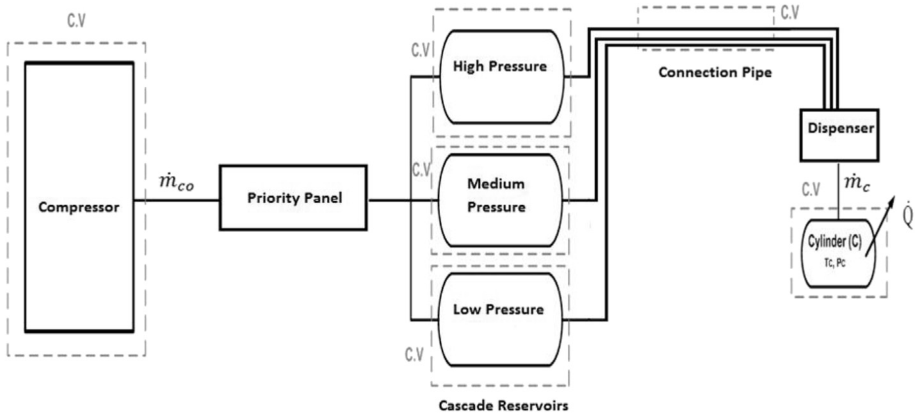

Fig. 1. A Schematic diagram of a CNG refueling station with cascade storage system.

modeled as a constant properties reservoir. Firstly, Kountz (Kountz, 1994), presented a simple thermodynamic model to modeling the fast filling process. The model was simple with only considering the on-board NGV cylinder. The heat transfer from the on-board NGV cylinder to the environment is investigated in his work. Kountz et al. (Kountzet al, 1997; Kountz et al., 1998a, 1998b, 1998c), have also developed a dispenser control algorithm. Farzaneh-Gord et al. (Farzaneh-Gord, 2008) modeled the process of fast filling for one storage system (buffer) in a CNG refueling station. The effects of ambient condition and initial reservoir tanks pressure on final on-board NGV cylinder pressure, temperature and filling time are considered in their works. Their model based on the on-board NGV cylinder, without considering the connection pipes. They modeled fast filling process while the pressure of reservoir tanks, remain constant during the process. Farzaneh-Gord et al. (Farzaneh-Gord et al., 2008) presented a thermodynamic analysis of cascade storage bank. Their model was similar to previous models. The results of this study indicated that ambient temperature and initial conditions have an enormous effect on final on-board NGV cylinder conditions and filling time and mass. Their results indicated that there was a temperature rise of at least 40 K for real gas through the fast filling process. They have also developed a theoretical analysis based on real gas assumptions to compare two storage systems. The results show both of storage systems have advantages over the other. It has been found that the entropy generation in the cascade storage system is 50% less than the buffer storage system. They presented, the less entropy generation which probably cases lower compressor input work, but energy consumption by the compressor has not been calculated, directly. To calculate the energy consumption by the compressor, it should be modeled. They showed that filling time in the buffer storage system is about 66% less than the other. Their model based on two cylinders without considering pipe, similar to the former ones. In a study, Bang et al., (Bang and et al., 2014) has modeled a CNG residential refueling system. Their model was able to generate electric power consumption profiles versus refueling time. The model is appropriate for simulating residential refueling systems (filling time is about 4 h) rather than public systems (fast fill). A theoretical analysis of an adiabatic system, has been developed to study the effects of the natural gas compositions on the performance of refueling stations by Farzaneh-Gord et al. (Farzaneh-Gord et al., 2014). The results showed that the compositions of natural gas have great effects on the final in–cylinder conditions at the end of the process. Furthermore, the gas with less methane percentage in its composition is more suitable for faster filling process. For example, the filling time for Pars natural gas type (with about 87% methane) is about 15 s less than Khangiran natural gas (with about 98% methane). Deymi-Dashtebayaz et al. (Deymi-Dashtebayaz and et al., 2014) presented an equation for foretelling heat transfer rate in on-board NGV cylinder based on a numerical simulation using the CFD code fluent. By using this equation and a thermodynamic model, the final conditions of the on-board NGV cylinder at the end of the filling process, were determined. Their results had been compared with an experimental research and showed a good agreement. There are similar studies which only consider the on-board vehicle cylinder for CHG (Olmos and Manousiouthakis, 2014; Hosseini and et al., 2012; Okawa and et al., 2012; Bourgeois and et al.; Monde and et al., 2012; Wang and et al., 2014; Zhao and et al., 2012; Suryan et al., 2012; Zheng and et al., 2012; Zheng and et al., 2013).

To the author knowledge, the limited researches have been carried out which considered the reservoir tanks and on-board NGV's cylinder. Deymi-Dashtebayaz et al. (Deymi-Dashtebayaz et al., 2012) studied the effects of charging dynamic reservoir tanks conditions on NGV cylinder filling time. Their results show that the initial pressure of reservoir tanks has a big effect on the reservoir tanks volume for bringing up the NGV cylinder to 200 bar. Failure to correctly predict the mass flow rate is evident in their results. The reason is the lack of connection pipes modeling. A thermodynamic model for a tank filling process is presented for CHG by Striednig et al. (Striednig and et al., 2014). Their simulation is used to predict the hydrogen properties during the fast filling process. They considered the reservoir tanks and on-board NGV's cylinder in their model but the mass flow rate hasn't been calculated and it has been assumed constant during filling.

There is also only one research which considered the reservoir tanks, on-board NGV's cylinder and connection pipes. A mathematical model has been developed to calculate mass flow rate of CNG considering connecting pipe by Khadem et al. (Khadem et al., 2015). The natural gas flow in connecting pipe was modeled as a Fanno flow (ideal and adiabatic flow where the effect of friction is considered) by them. In their work, the reservoir tanks modeled as a constant properties reservoir, similar to many of the previous research. In their work, the mass flow rate of gas was calculated by a trial and error procedure.

Having a complete model for a CNG refueling station is of paramount importance. It should be noted that in all previous work (Kountz, 1994; Farzaneh-Gord, 2008; Farzaneh-Gord et al., 2008, 2014; Deymi-Dashtebayaz and et al., 2014; Olmos and Manousiouthakis, 2014; Hosseini and et al., 2012; Okawa and et al., 2012; Bourgeois and et al.; Monde and et al., 2012; Wang and et al., 2014; Zhao and et al., 2012; Suryan et al., 2012; Zheng and et al., 2012; Zheng and et al., 2013; Deymi-Dashtebayaz et al., 2012; Striednig and et al., 2014; Khadem et al., 2015), only a few components of the refueling station is modeled. The above literature review shows that, there have not been any studies which modeled the reservoir tanks, on-board NGV's cylinder, connection pipes and reciprocating compressor simultaneously. The reciprocating compressor is the heart of any CNG refueling station, which has dominated influence on refueling process. Reciprocating compressors are used in a lot of industrial applications. They can provide high pressure ratios (ractical Guide to Comp, 2006). There are a number of methods have been developed for these tapes of compressors thermodynamic analysis (Stouffs et al., 2001; Sun and Ren, 1995; Elhaj and et al., 2008; Ndiaye and Bernier, 2010). There isn't any report about reciprocating compressor performance in unsteady conditions, too.

The main objective of the present study is to apply a thermodynamic analysis for a complete modeling of a CNG refueling station. The natural gas flow between reservoir tanks and on-board NGV cylinder is modeled as a one-dimensional and isothermal flow in the present work. the results of the present work have been validated against the experimental data and compared with the results of a previous work (Farzaneh-Gord et al., 2008), to investigate the effect of connecting pipes. On the other hand, unlike many of the previous researches (Kountz, 1994; Farzaneh-Gord, 2008; Farzaneh-Gord et al., 2008, 2014; Deymi-Dashtebayaz and et al., 2014; Olmos and Manousiouthakis, 2014; Hosseini and et al., 2012; Okawa and et al., 2012; Bourgeois and et al.; Monde and et al., 2012; Wang and et al., 2014; Zhao and et al., 2012; Suryan et al., 2012; Zheng and et al., 2012; Zheng and et al., 2013; Deymi-Dashtebayaz et al., 2012; Striednig and et al., 2014; Khadem et al., 2015), pressure variations of the reservoir tanks during the filling process have been considered in the present research. It is necessary for compressor modeling. The reciprocating compressor of CNG refueling station has been modeled in the present work. The experimental data, measured by the present authors, is used to identify the compressor performance in unsteady conditions.

2. Theory and modeling

A CNG refueling station could be divided into six components (Fig. 1); compressor, priority panel, reservoir tanks, connecting pipe, dispenser and on-board NGV cylinder. A simplified analysis of a CNG refueling station could be performed by modeling these components. The compressor is used to fill the reservoir tanks, when reservoir tank pressure is decreased to a certain limit. In a cascade storage system, reservoir tanks are filled on a priority identified by an algorithm. Based on another algorithm, the on-board NGV cylinder is switched from the low pressure reservoir to the higher one, when the mass flow rate drops to a certain limit (in dispenser). Reservoir tanks in a cascade storage system are divided into high, medium and low pressures (HP, MP and LP).

The theory used to model a CNG refueling station has been described in this section. The section involves four parts; on-board NGV cylinder, reservoir tanks, connecting pipe and compressor.

2.1. On-board NGV cylinder

The on-board NGV cylinder is assumed as a simple thermodynamic open system. The system has a uniform pressure and temperature. Applying the mass conservation principle to the control volume:

$$\frac{dm_c}{dt} = \dot{m}_i \tag{1}$$

where $\dot{m}_i$ is inlet mass flow rate, described in the following section. The general form of the first law of thermodynamics yields:

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

The change in kinetic and potential energy and the work term could be neglected as there is only one inlet, yields:

$$\frac{dU_c}{dt} = \dot{Q}_{cv} + \dot{m}_i \left(h_i + \frac{V_i^2}{2}\right) \tag{3}$$

The above equation could be more simplified by replacing enthalpy of the reservoir tanks:

$$\frac{dU_c}{dt} = \dot{Q}_{cv} + \dot{m}_i (h_r) \tag{4}$$

To simplify, the heat transfer of the on-board NGV cylinder to the environment is negligible. For an ideal gas behavior:

$$h = c_p T, \quad u = c_v T, \quad PV = mRT \tag{5}$$

where c_p and c_v are constant pressure specific heats and constant volume specific heats, respectively. Knowing that the reservoir tank temperature, the on-board NGV cylinder volume and specific heats are constant, Equation (4) could be simplified as follows:

$$\frac{d}{dt}\left(\frac{P_c V_c}{R T_c} \cdot c_v T_c\right) = \dot{m}_i c_p T_r \tag{6}$$

Finally, the first law of thermodynamics could be written as for the on-board NGV cylinder:

$$\frac{d}{dt}(P_c) = \dot{m}_i T_r \left(\frac{c_p}{c_v} \frac{R}{V_c}\right) \tag{7}$$

Using Equation (7), the on-board NGV cylinder pressure in the new time step could be calculated. Using Equation (1) and the on-board NGV cylinder pressure in the new time step and ideal gas equation of state, the on-board NGV cylinder temperature in the new time step could be detected.

2.2. Reservoir tanks

Similar to the previous section, applying the mass conservation equation to the new control volume (reservoir tanks) yields:

$$\frac{dm_r}{dt} = -\dot{m}_e \tag{8}$$

That $\dot{m}_e$ is outlet mass flow rate, described in the following section. General form of the first law of thermodynamics for the new control volume (Equation (2)) yields:

$$\frac{dU_r}{dt} = \dot{Q}_{cv} - \dot{m}_e (h_r) \tag{9}$$

To simplify, the heat transfer of reservoir tanks to the environment is negligible. Similar to the on-board NGV cylinder, the first law of thermodynamics could be written as for the reservoir tanks:

$$\frac{d}{dt}(P_r) = \dot{m}_e T_r \left(\frac{c_p}{c_v}\frac{R}{V_r}\right) \tag{10}$$

Using Equation (10), the reservoir tank pressure in the new time step could be calculated.

2.3. Connecting pipe

The natural gas flow between reservoir tanks and on-board NGV cylinder is modeled as a one-dimensional and isothermal flow. Transient process is a quasi-steady process. To calculate the mass flow rate of gas through the connecting pipe, continuity, momentum and equation of state must be solved for a pipe.

The governing equations (Continuity, momentum and state equation) for pipe flow are as follows, (Oosthuizen and Carscallen, 1997):

$$\frac{d(\rho V)}{dx} = 0 \tag{11}$$

$$\frac{d(\rho VV)}{dx} + \frac{dP}{dx} + \frac{f}{2D}\rho V^2 = 0 \tag{12}$$

$$P = z\rho\frac{R}{M}T \tag{13}$$

where, z, ρ, V, P, f, D, R, M and T are compressibility factor, density, velocity, pressure, Fanning friction factor, diameter of pipe, gas constant, molecular weight and temperature, respectively.

Applying the mass Continuity equation:

$$\rho V = \frac{\dot{m}}{A} \tag{14}$$

The combination of Equations (12) and (14) gives:

$$\underbrace{\rho^2 VdV}_{I} + \underbrace{\rho dP}_{II} + \underbrace{\frac{fdx\dot{m}^2}{2A^2D}}_{III} = 0 \tag{15}$$

The kinetic energy term (I) is negligible when compared with the other terms (II and III).

To integrate term (II) of the previous equation and by using Equation (11):

$$\int_{P_1}^{P_2} \rho dP = \int_{P_1}^{P_2} \frac{MP}{ZRT}dP = \frac{M}{Z_{avg}RT_{avg}}\int_{P_1}^{P_2} PdP = \frac{M\left(P_2^2 - P_1^2\right)}{2Z_{avg}RT_{avg}} \tag{16}$$

Which (Weymouth, 1912):

$$T_{avg} = \frac{T_1 + T_2}{2} \quad and \quad P_{avg} = \frac{2}{3}\left[P_1 + P_2 - \frac{P_1P_2}{P_1 + P_2}\right] \tag{17}$$

Z_{avg}, could be calculated based on the T_{avg} and P_{avg} for pure methane by an equation of state (Soave-Redlich-Kwong equation of state). This equation is convenient for thermodynamic modeling. So Z_{avg} is given by the Soave-Redlich-Kwong equation of state (Vidal):

$$Z^3 - Z^2 + \left[A^* - B^*\left(1 + B^*\right)\right]Z - A^*B^* = 0 \tag{18}$$

With, A^* and B^* are obtained by pressure and temperature, (Vidal).

To integrate term (III) of the Equation (15):

$$\int_{x_1}^{x_2} \frac{f\dot{m}^2}{2A^2D}dx = \frac{f\dot{m}^2}{2A^2D}(x_2 - x_1) = \frac{f\dot{m}^2}{2A^2D}(L) \tag{19}$$

where L is the length of pipe between reservoir tanks and on-board NGV cylinder. Substituting equations (16) and (19) in Equation (15), mass flow rate is given by:

$$\dot{m} = A\frac{1}{\sqrt{f}}\left[\frac{MD\left(P_1^2 - P_2^2\right)}{LRT_{avg}Z_{avg}}\right]^{0.5} \tag{20}$$

By Spitzglass (medium pressure) equation (Menon, 2014):

$$\frac{1}{\sqrt{f}} = \left(\frac{88.5}{1 + 0.09144/D + 1.1811D}\right)^{0.5} \tag{21}$$

So, mass flow rate could be obtained by Equations (17), (18), (20) And (21).

2.4. Compressor

The compressor type used in a CNG refueling station is reciprocating compressor (usually multi-stage). This type of compressor use piston–cylinder combinations to compress gas. Fig. 2 shows the schematic diagram of three-stage compressor with inter cooler between stages and after cooler at the outlet of the compressor. The most important parameters in compressor modeling are outlet mass flow rate and actual work of the compressor.

In the next section (Analysis and discussion) it will be shown that the outlet mass flow rate of compressor remains constant during the operation of the compressor based on experimental measurements. The outlet mass flow rate of the compressor is calculated by the following equation:

$$\dot{m}_{co} = \rho_{std} \times Q_{std} = \left(\frac{Mw_{gas}}{Mw_{air}}\right) \times \rho_{air,std} \times Q_{std} \tag{22}$$

where Mw_{gas}, Mw_{air}, $\rho_{air,std}$ and Q_{std} are molecular weight of gas (Methane in the present work), molecular weight of air, air density at standard conditions and the capacity of the compressor at standard conditions, respectively.

The adiabatic compressor work is calculated by the following equation (Hanlon, 2001):

$$W_{adi} = \frac{k}{k-1}(Z_{su}T_{su}R)\left[\left(\frac{P_{dis}}{P_{su}}\right)^{\frac{k-1}{k}} - 1\right] \tag{23}$$

Total adiabatic compressor work for three stages is calculated by:

$$\begin{aligned} W_{adi} = {} & \frac{k_1}{k_1-1}(Z_{su1}T_{su1}R)\left[\left(\frac{P_{dis1}}{P_{su1}}\right)^{\frac{k_1-1}{k_1}} - 1\right] \\ & + \frac{k_2}{k_2-1}(Z_{su2}T_{su2}R)\left[\left(\frac{P_{dis2}}{P_{su2}}\right)^{\frac{k_2-1}{k_2}} - 1\right] \\ & + \frac{k_3}{k_3-1}(Z_{su3}T_{su3}R)\left[\left(\frac{P_{dis3}}{P_{su3}}\right)^{\frac{k_3-1}{k_3}} - 1\right] \end{aligned} \tag{24}$$

The actual compressor work is calculated by the following equation:

$$W_a = \frac{W_{adi}}{\eta_c \times \eta_m} \tag{25}$$

where η_c and η_m are compression efficiency and mechanical efficiency, respectively. These parameters are commonly reported by compressors companies.

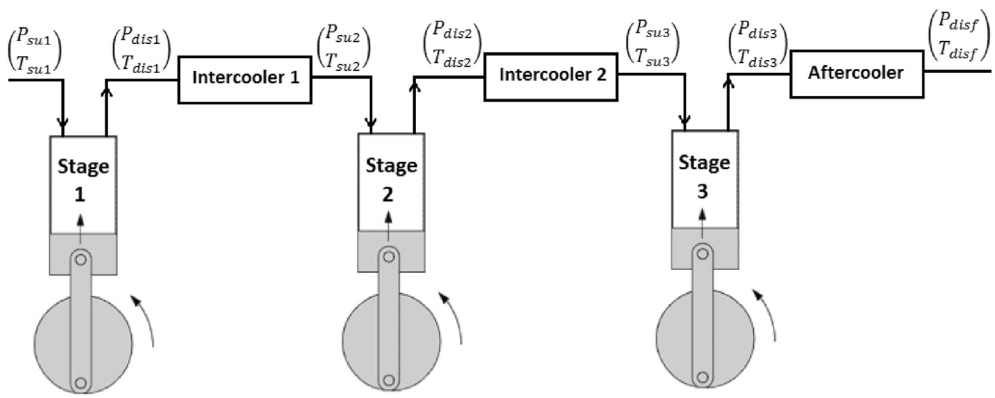

Fig. 2. Schematic diagram of three-stage reciprocating compressor.

3. Results and discussion

3.1. Validation

The comparison between the results of present work, a previous work (Farzaneh-Gord et al., 2008) and experimental values of George (George and Mayeaux, 2014) are presented in this section. The initial pressure of the on-board NGV cylinder is 72 bar and the cylinder volume is 52 litters. There is a cascade storage system in experimental procedure. The initial pressure of the reservoir tanks is 251 bar. The mass flow rate varies between 0.29 and 0.0 kg/s and when it drops to 0.04 kg/s, the on-board NGV cylinder is switched to the other reservoir tanks by dispenser algorithm. Fig. 3 shows the mass flow rate vs. time. Fast filling process has started after 24 s and has ended after 36 s from the start time. It has started with exhausting the reservoir tank 1 and after 26 s, has switched to another reservoir tank. The reason of peak at time~ 55 is increased upstream pressure by switching to another reservoir tank. A good agreement could be seen between experimental and the simulation values of the present work. The deviation may be due to the considered assumptions. The simulation values show the filling time have only 2 s differences from experimental values. The difference is about 10 s for the previous work (Farzaneh-Gord et al., 2008). It should be noted that, in the previous work (Farzaneh-Gord et al., 2008), to determine the mass flow rate between the reservoir tanks and on-board NGV cylinder, the ideal mass flow rate is multiplied by a discharge coefficient. The discharge coefficient is usually selected using matching experimental and numerical values.

In Fig. 4, the on-board NGV cylinder pressure variations pending fast filling process are shown for the experiment, present and the previous work (Farzaneh-Gord et al., 2008). Note from the figure, the on-board NGV cylinder pressure for the experimental work is less than the present work and previous works. It may be caused by the assumption of ideal gas and adiabatic model in the on-board NGV cylinder. So, the results of the present work are more accurate than the previous works.

3.2. Experimental analysis of compressor

As mentioned, the compressor type used in CNG refueling stations is reciprocating compressor. Such as NGV cylinder and reservoir tanks, compressor performance is in unsteady conditions. The third stage output pressure changes with time caused unstable conditions for compressor. Two challenges appeared when reciprocating compressor performance in unsteady conditions was

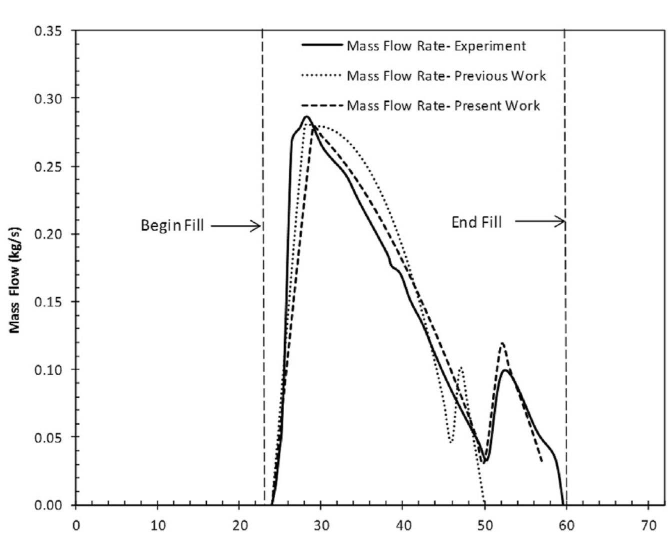

Fig. 3. Comparison between the simulations and experimental results for mass flow rate.

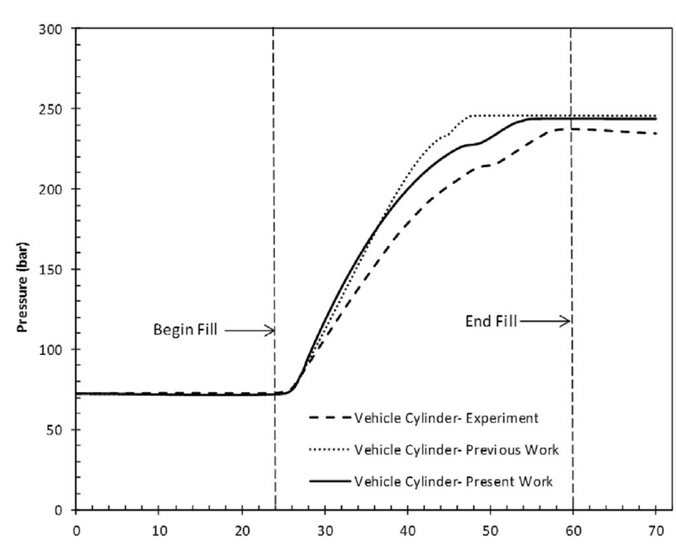

Fig. 4. Comparison between the simulations and experimental values for pressure variations of NGV cylinder.

considered. First: how does pressure distribution change with changes in the output pressure of the third stage? Pressure distributions include inlet pressure compressor, the first stage output pressure and second stage output pressure (Fig. 2). Second: how does mass flow rate change with changes in the output pressure of the third stage?

The experiments have been carried out on a typical CNG refueling station (Baghdar CNG station No.1043, Mashhad, Iran). The CNG refueling station has 4 dispensers, 8 refueling nozzles and one compressor. The main characteristics data for the compressor are given in Table 1.

The schematic of the experimental setup is shown in Fig. 5. The pressure sensors mounted externally (PT) shown in the figure. Some locations of compressor (four points) and reservoir tanks (three points) are instrumented with pressure sensors to monitor the pressure during the variation. Inlet pressure changes during the time could be detected by a pressure sensor located at compressor inlet. The first, second and third stage output pressure changes could be detected by using the other pressure sensors located on the compressor. Reservoir tanks pressure changes could be observed by using the sensors installed on them. Experimental procedure, measured by present authors has been carried out for a period of time to determine the compressor performance in unsteady conditions. Compressor performance is unsteady because the compressor outlet pressure changes during the time.

The pressure distributions of the gas vs. time curve for the compressor are shown in Fig. 6. It could be realized that the compressor has started after 6 min and has stopped after 35 min from start time. It is evident that the compressor inlet pressure during the operation is always constant, about 15 bar. This value is the inlet gas pipeline pressure. The output pressure of the first stage and second stage, during the compressor operation are also almost constant, about 42 and 88 bar respectively. The output flow of the third stage, after passing into the priority panel, enters to reservoir tanks. According to the increase in pressure vs. time in reservoir tanks, the output pressure of the third stage increases, too. Also, the output pressure of the third stage decrease because of the empty tanks. So, unlike in previous cases, the output pressure of the third stage varies with time. This is evident in the figure. Therefore, it is concluded that, pressure variations of the third stage, have no effect on the inlet and intermediate stage pressure. As is shown in this figure, the natural gas is compressed in the first stage to 42 bar, then transferred to the second stage and compressed to 88 bar, then transferred to the third stage of compression where the natural gas is finally compressed to a pressure of reservoir tanks. This important consequence is the answer for the first challenge expressed previously.

Pressure versus time curve is shown in Fig. 7 for HP reservoir tank. As is shown in this figure, since the beginning until 6 min, the pressure is reduced. The reason for this decrease is related to the increase in NGV cylinder pressure. The compressor has started after 6 min and the high pressure reservoir tank is selected by priority panel, immediately. The pressure is increased from 206 bar to 230 bar by compressor during the 30 s. Pressure is continuously increased to 30 s. Therefore, Output pressure of the third stage also increases continuously in 30 s. After the pressure was increased to 230 bar, the priority panel selects another tank to fill. So, the pressure of the HP reservoir tank is reduced, because the gas comes out to the dispenser. The process of filling and empty the reservoirs continue until the compressor is turned off again. As is shown in this figure growth rate for pressure is nearly constant during the 30 s and this growth rate is nearly constant at other times, too. Pressure growth rate could be considered equivalent to the mass flow rate changes. Therefore, it is concluded that, pressure variations of the third stage, have no effect on the mass flow rate of the compressor. This important consequence is the answer for the second challenge expressed previously.

Table 1
Characteristics of compressor used for experiments.

Number of stages	3
Cylinder Double Acting/Single Acting	Single Acting
Cylinder Bore 1st Stage	110 mm
Cylinder Bore 2nd Stage	66 mm
Cylinder Bore 3rd Stage	42 mm
Stroke 1st Stage	127 mm
Stroke 2nd Stage	127 mm
Stroke 3rd Stage	127 mm
Capacity	1500 Nm^3/hr
Compressor Shaft Speed	800 rpm
Gas	Methane
Compression efficiency	0.85
Mechanical efficiency	0.95

3.3. Simulation results

Natural gas primarily consists of methane, so in this work CNG was assumed pure methane. The thermodynamic model is used to predict the dynamic pressure variation during fast filling process for a cascade storage system. Information about CNG refueling station and the NGV is presented in Table 2. It is obtained from a typical CNG refueling station (Baghdar CNG station No.1043, Mashhad, Iran). The following assumptions have been assumed for extracting the results:

- The initial pressure of each NGV cylinder is 1 bar.
- Environment Temperature remains constant (293 K).
- The final pressure of NGV cylinder is 200 bar, at least.
- Time for switching between vehicles is 1 min.
- There is only one dispenser in the CNG refueling station.
- During the fast filling process, the compressor is off.
- Initial pressure and temperature for cascade storage system are 235 bar and 293 K, respectively.

Variations of pressure during the time for the cascade storage system are shown in Fig. 8. It should be noted that a total of 26 vehicles could be filled without the compressor having to recharge the reservoir tank. After 105 min (discharge time) the HP reservoir tank pressure has dropped down to nearly 200 bar. Cascade storage reservoir recovery time is about 17 min (charge time). A complete cycle (discharge and charge) for a cascade storage system takes about 122 min. For the first vehicle the reservoir tank pressure is 235 bar, after the first vehicle is filled, the LP reservoir tank pressure will be 224 bar. The first three vehicles could be filled from the LP reservoir tank without having to switch to the next reservoir tank. The fourth vehicle could not be filled to the 200 bar level from the LP reservoir tank. The fourth vehicle will initially be filled from the LP reservoir tank until the mass flow rate drops to a certain limit. The MP reservoir tank will now have to be used to fill the vehicle. The twelfth vehicle could not be filled to the 200 bar level from the LP and MP reservoir tanks. The HP reservoir tank will now have to be used to fill the vehicle. There is an algorithm in the dispenser to manage the filling of NGV cylinders (discharge zone). This algorithm applied to fill process from a cascade storage system. According to the algorithm, a NGV will first be filled from the low pressure reservoir tanks, which will result an incomplete fill, perhaps to 220 bar. The medium and high pressures reservoir tanks will bring the vehicle's tank closer to 220 bar. If this algorithm is continued, 26 vehicles could be filled before cascade system will need to be charged. The average filling time for a cascade storage system, is about 242 s.

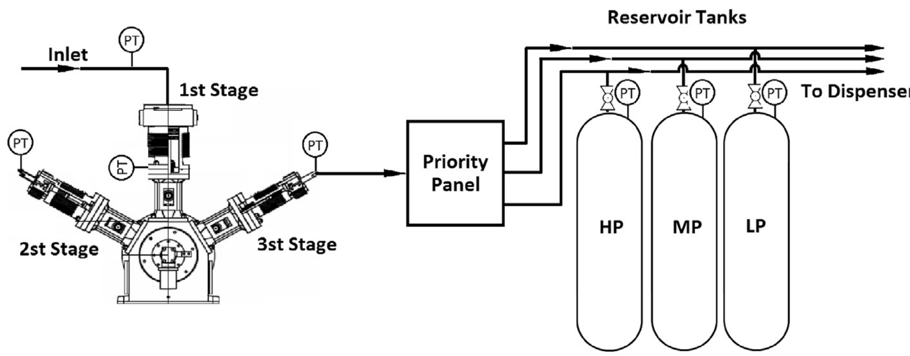

Fig. 5. Schematic of the experimental procedure.

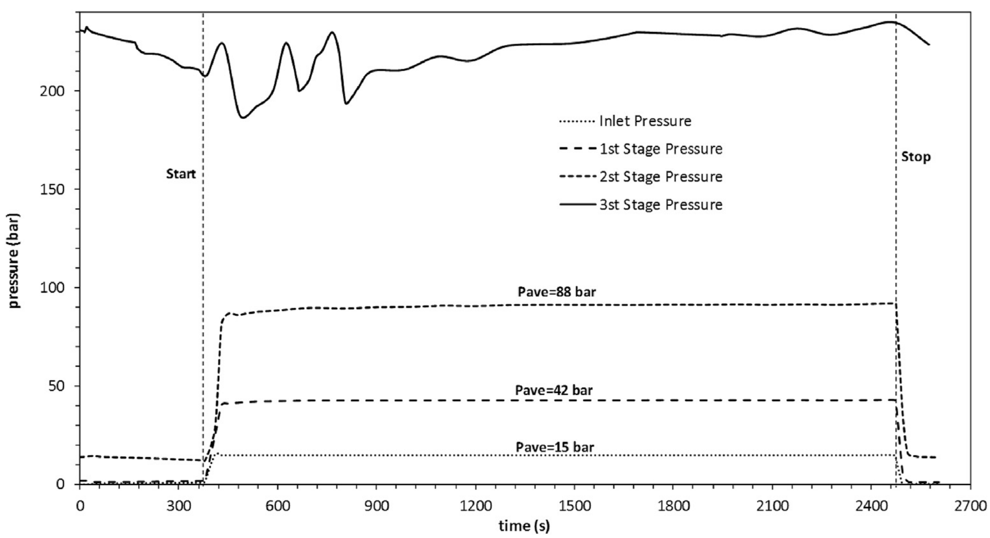

Fig. 6. Pressure measurement and monitoring of the compressor.

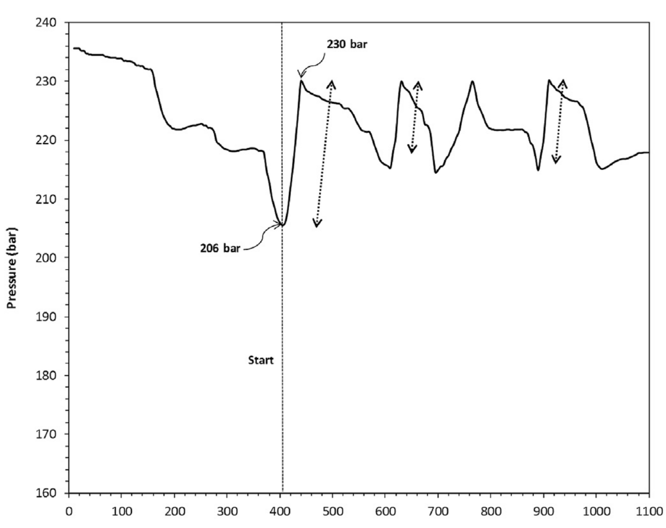

Fig. 7. Pressure variations of HP reservoir tank during the time.

There is another algorithm in the priority panel to manage the filling of reservoir tanks which means that one reservoir tank is filled after another (charge zone). Firstly, HP reservoir tank, then MP reservoir tank and finally, LP reservoir tank are filled by the compressor.

Table 2
Some information about CNG refueling station and NGV on-board cylinder.

Storage capacity of on-board NGV cylinder	101 lit
HP reservoir tank capacity	1920 lit
MP reservoir tank capacity	1920 lit
LP reservoir tank capacity	1920 lit
Priority panel	3 lines
Length of pipe	50 m
Inside diameter of pipe	0.009 m
Initial pressure of on-board NGV cylinder	1 bar
Initial pressure of HP, MP and LP reservoir tanks	235 bar

To see more details, the mass flow rate of the gas for cascade storage system is shown in Fig. 9. The results are presented for the first, tenth and twenty-fifth vehicles. Based on dispenser algorithm, the on-board NGV cylinder is switched from lower pressure reservoir to a higher one, when the mass flow rate drops to certain limit (0.02 kg/s for present work). According to the figure, the mass flow rate of gas profile for the tenth and twenty-fifth vehicles are divided into two and three separate parts, respectively. As shown in the figure, filling time for the first, tenth and twenty-fifth vehicle are 110, 155 and 210 s. As mentioned, one of the advantages of the present work is a more accurate prediction of CNG mass flow rate than the previous works.

The final results of the CNG refueling station are shown in Table 3. It should be noted that 26 vehicles could be refueled in a

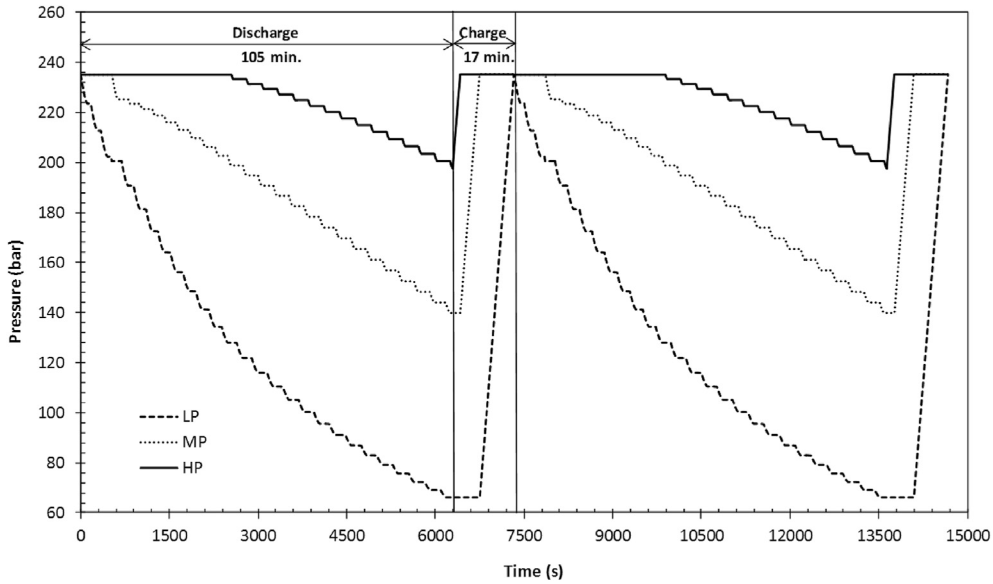

Fig. 8. Pressure profile of reservoir tanks for the cascade storage system.

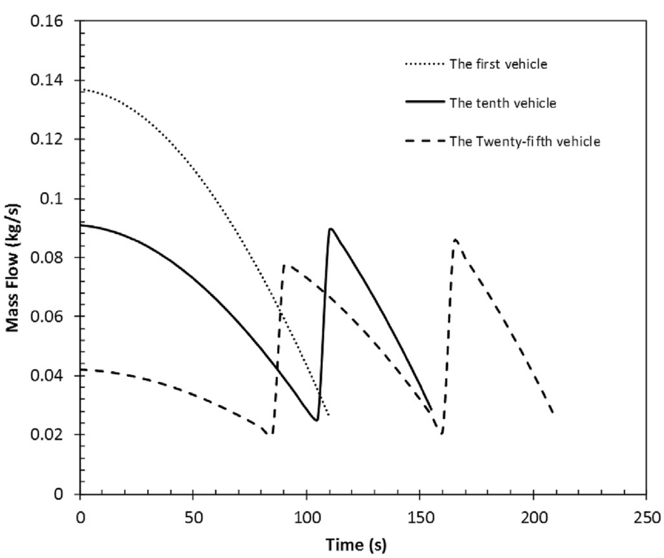

Fig. 9. Samples of mass flow rate of the gas for cascade storage system.

Table 3
The final results of the cascade storage systems.

Required time for a cycle	122 *min.*
Number of vehicles that could be fueled in a cycle	26
Number of cycles per day	12
Number of vehicles that could be fueled in a day	307
Energy consumption for a cycle	61.86 kWh
Average filling time for a vehicle	242 s.
Average specific energy consumption for a vehicle	0.250 kWh/*kg*

cycle for the cascade storage system. The number of cycles per day is equal to 12. Thus, the total numbers of on and off the compressor are almost 12. This is an important parameter because by increasing this number, the compressor maintenance costs increase. The energy consumed by the compressor for a cycle, determined by using of the previous section (Theory and modeling) is equal to 61.86 kWh. The average specific energy consumption is equal to 0.250 kWh/kg for a vehicle. Bang, H.J., et al. (Bang and et al., 2014) has calculated average specific energy consumption to be around 0.19 kWh/kg for slow filling process (Residential Refueling System). As the energy consumption for slow filling is expected to be less than fast filling, the calculated value for specific energy consumption could be considered valid. This is an important value as it could give a rough estimation for calculating energy consumption by the compressor in a CNG station. The value could be also used to design a more optimized CNG refueling station.

The best pattern for a system of storage reservoir could be selected by balancing the two parameters, filling time and energy consumption by the compressor. This could be the subject of future works.

4. Conclusion

The review of literature shows there isn't any research about of a CNG and CHG refueling station which considers the reservoir tanks, on-board NGV's cylinder, connection pipes and reciprocating compressor. It should be noted that in all previous work, only the fast filling process is modeled. The fast filling process is a part of a CNG refueling station. The compressors have not been considered in all previous studies which investigated the CNG stations. The main objective of the present work is to apply a thermodynamic analysis for a CNG refueling station, including on-board NGV cylinder, reservoir tanks, connecting pipes between them and reciprocating compressor. The results of the present work have been validated against the experimental data and compared with the results of a previous work to investigate the effect of connecting pipes. On the other hand, unlike all previous research, pressure changes of the reservoir tanks during the filling process have been considered in the present work. The reciprocating compressor of CNG refueling station has been modeled in the present work. The experimental data, measured by the present authors, is used to identify the compressor performance in unsteady conditions. The modeling of connecting pipes at present work cause the better dynamic pressure profile than the previous works. It is concluded that variations in the compressor discharge pressure have no effect on the first and intermediate compressor pressure. It has also no effect on the compressor delivery mass flow rate. It is found that 26 vehicles could be refueled in a cycle for a cascade storage system.

The energy consumed by the compressor for a cycle is equal to 61.86 kWh. The average specific energy consumption is equal to 0.250 kWh/kg for a vehicle. The best pattern for a system of storage reservoir could be selected by balancing the two parameters, filling time and energy consumption by the compressor. This could be the subject of future works.

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Nomenclature

A	area, (m^2)
c_p	Constant pressure specific heats (kJ/kg K)
c_v	Constant volume specific heats (kJ/kg K)
D	Diameter (mm)
f	Fanning friction factor
g	Gravitational acceleration, (m/s^2)
h	enthalpy, (kJ/kg)
k	Isentropic expansion factor
kWh	kilowatt hour
l	Liter
L	Length (m)
$\dot{m}$	Mass flow rate, (kg/s)
M	Molecular weight, (kg/kmol)
P	Pressure, (bar or kPa)
$\dot{Q}$	Heat transfer rate, (kW)
R	Gas constant (J/mol K)
T	Temperature, (K or ^oC)
u	Internal energy, (kJ/kg)
t	time, (s)
V	Velocity (m/s)
V	Volume, (m^3)
W	Actual work, (kJ/kg)
$\dot{W}$	Actual work rate, (kW)
z	Height, (m)
ρ	Density, (kg/m^3)

Subscript

a	actual
avg	average
c	on-board NGV cylinder
co	compressor
cv	control volume
i	inlet condition
dis	discharge
r	reservoir tank
su	suction
std	standard

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