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EFFECT OF GAS COOLER SIZE ON ITS PERFORMANCE AND ENTIRE R744 A/C SYSTEM

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ABSTRACT

This paper presents an investigation of the effect of gas cooler size reduction on the capacity and system COP for a residential a/c system operating with R744 in transcritical mode. The gas cooler consists of three slabs with a multi-serpentine structure. Two of the three slabs were used as the first pass and one of them was operated as the second pass. One of the two slabs in the first pass was eliminated by partially closing the headers on the refrigerant side. As a result, the volume of the heat exchanger was reduced by one third. Experimental results showed that this 33% volume reduction of the gas cooler decreased the capacity slightly, but significantly increased the refrigerant pressure drop. The model prediction was closer to the experimental results for the condition with partially blocked headers than with unblocked headers. The simulation results showed that the capacity ratio between the first and second pass changed due to the reduction in the volume of the first pass. The system model predicted the change of COP and gas cooler capacity with respect to the change of the gas cooler size.

1. INTRODUCTION

Carbon dioxide (R744) has been considered as one of the potential replacements of conventional refrigerants because of its lower global warming potential (GWP) than HFCs. One very important factor for competitiveness of R744 systems is the development of compact heat exchangers with microchannel tubes. The airside and refrigerant side heat transfer issues have been addressed in work by Yin et al. (2001), Beaver et al. (1998, 1999), Richter et al. (2003), and Pettersen et al. (1998) in R744 mobile and residential air conditioning systems with microchannel heat exchangers. In the previous studies, carefully designed microchannel heat exchangers were implemented and their capacity and system effects were examined in detail. However, to date, very little research has been conducted on the effect of changing the heat exchanger size and its subsequent effect on the capacity of heat exchanger and system COP. Hrnjak (2003) presented some effects of heat exchanger sizing for a particular mobile A/C system.

This study was performed with a special heat exchanger for transcritical R744 residential air-conditioning and heat pump systems. The novelty of a heat exchanger design approach is mostly related to the orientation of the air and refrigerant flow, refrigerant circuiting and the header design. Most of the conventional heat exchangers in transcritical R744 systems have cross flow geometry to air and long headers to distribute refrigerant to each tube. The long headers of the heat exchangers cause maldistribution of the refrigerant to each tube and it reduces the capacity of the heat exchanger. The heat exchanger that is investigated in this study has a cross-counter flow geometry to air and short headers to reduce maldistribution in the case of two-phase flow. Furthermore, it decreases or increases the cross sectional area of the refrigerant side by varying the number of slabs along the refrigerant flow direction, so the same heat exchanger could be used as a gas cooler and an evaporator in air conditioning mode. Microchannel tubes were used in the heat exchanger and all components were made from aluminum. The heat exchanger has 3 slabs: 2 of the slabs worked as the first pass and 1 slab as the second pass for gas cooler mode. By partially blocking the header, one slab of the first pass was effectively eliminated. As a result, the heat exchanger had a 33% total volume reduction and a 50% cross section area reduction on the refrigerant side in the first pass. This paper presents the experimental capacity and pressure drop characteristics due to the experimental reduction in volume and refrigerant side area. These results were compared with the unblocked heat exchanger test results. The heat exchanger was operated as the gas cooler in A/C mode.

The comparison of the two test modes: operation with 33% volume reduced and designed heat exchanger, was performed with the experimental and simulation results. The gas cooler model predicted the heat exchanger capacity for the same inlet conditions as those of the experiment. The effect of the gas cooler size on the system COP and the gas cooler capacity was considered with the system simulation model.

2. EXPERIMENTAL FACILITIES

In order to measure the capacity and pressure drop of the heat exchanger as a gas cooler of an R744 system, it was placed in a wind tunnel located in an environmental chamber. The airflow rate was varied. Two independent methods, the airside and refrigerant side energy balance, were applied to measure the gas cooler capacity. Figure 1 is the schematic of the test facilities. The refrigerant flow in the gas cooler is shown for the blocked and unblocked header test conditions. Beaver et al. (1999) presented the facilities in detail. Two independent methods determined the capacity within $\pm$ 2.4% for partially blocked test and $\pm$ 3.2% for unblocked test for the gas cooler respectively. The uncertainty calculation of the measured capacity was based on the guidelines presented by Taylor et al. (1994) and it was performed using EES (Engineering Equation Solver). The maximum uncertainties were 2.9% for the measured gas cooler capacity.

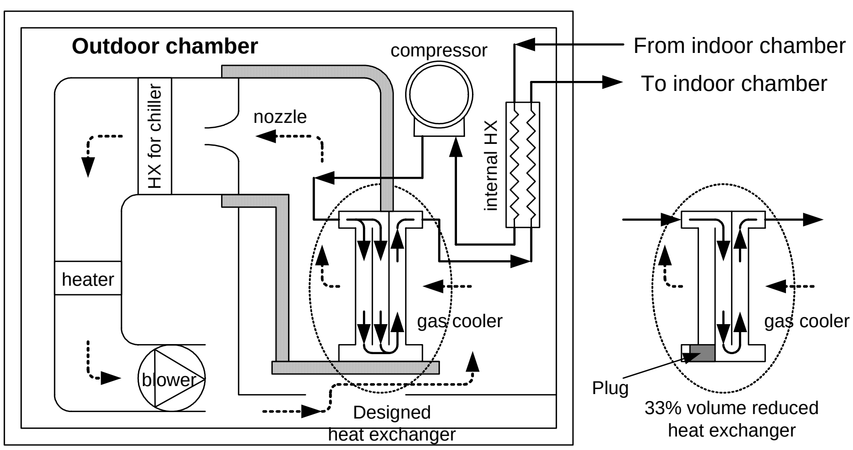

Figure 1. Test facilities

3. HEAT EXCHANGER DESCRIPTION

The heat exchanger in this study has a multi serpentine arrangement with short headers intended to reduce the maldistribution of refrigerant to the microchannels, and cross-counter flow geometry for refrigerant and air. The heat exchanger used as the gas cooler had 3 slabs and each slab consisted of 4 branches. Figure 2 shows the structure of the heat exchanger with refrigerant and air flow direction.

At the top part of the heat exchanger, there is a header. 12 tubes of 3 slabs entering the header. The header has a baffle which separates the inlet and outlet sections. One header connected to the 2 slabs was used as the refrigerant inlet and the other header was used as the outlet as shown in Figure 1. Figure 2 presents the heat exchanger in gas cooler operation. This design was used to compensate for the decrease in the refrigerant specific volume along the gas cooling path. There are two headers at the bottom part of the heat exchanger. Each header is connected to 4 tubes of 2 slabs operated as the first pass, and 2 tubes of single slab used as the second pass. The refrigerant merged from the 4 tubes and was fed to 2 tubes in the header for the unblocked gas cooler mode. The refrigerant, viewed from the air inlet side, was supplied from the backside of the heat exchanger.

Two cylindrical plugs were used to partially block the headers as shown in Figure 2. They were located at the bottom headers of the heat exchanger in the experiments to simulate reduced heat exchanger size. The diameter of the plugs is smaller than the header internal diameter. Two O-rings were used to seal and prevent refrigerant from passing through the plugs. As a result, there was a reduction of 33% in the volume of the heat exchanger and 50% in the refrigerant flow area in the first pass.

The nominal diameter of the microchannel is 0.8 mm and each tube has 10 round microchannels. Louvered folded fins were used in this heat exchanger. More information about the heat exchanger is given in Table 1.

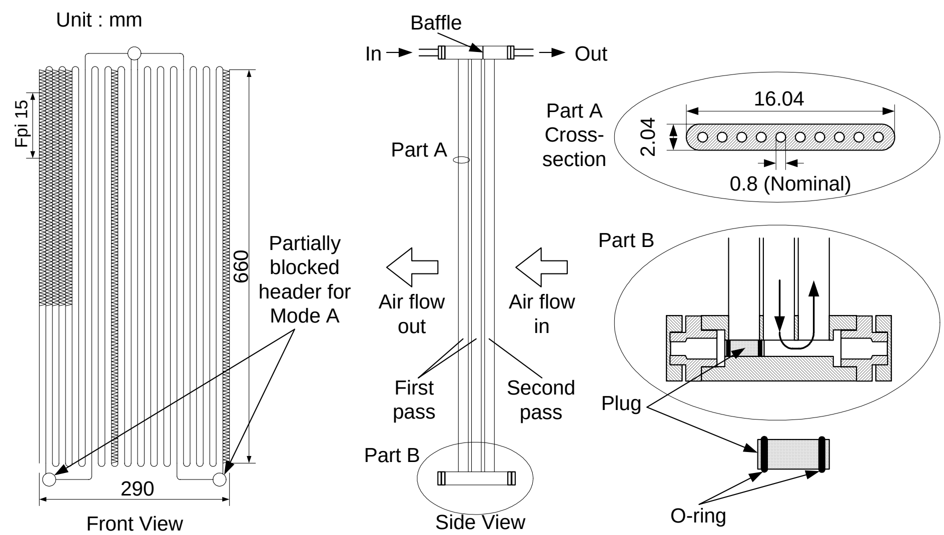

Figure 2. A schematic of heat exchanger and plugs

Table 1. Dimensions and characteristics of the heat exchanger

Gas cooler characteristics		Gas cooler characteristics	
Airside area [m^2]	10.52	Louver angle [deg]	27
Channel diameter [mm]	0.8	Louver entry length [mm]	1.7
Core depth [mm]	49.15	Louver height [mm]	6.0
Core volume [cm^3]	9186	Louver pitch [mm]	1.0
Face area [cm^2]	1914	Number of channel	10
Fin density [fins/in]	15	Number of louvers	2×6
Fin height [mm]	8	Refrigerant side area [m^2]	1.393
Fin thickness [mm]	0.1	Tube major diameter [mm]	16.04
Header tube diameter [mm]	6.5	Tube minor diameter [mm]	2.04

4. TEST CONDITION

The measurement of the capacity and the pressure drop was performed at steady state in ARI A condition (1989) and the air inlet temperature to the gas cooler remained relatively invariant at 35°C. Frontal air velocity for the gas cooler was 0.6, 1.2, and 1.8 m/sec, respectively while the R744 mass flow rate was 15, 20, and 25 g/sec, respectively for both partially blocked and unblocked heat exchanger header conditions. Table 2 shows the refrigerant inlet conditions. The air inlet temperature was controlled identically for all test cases at 35°C.

Table 2. Refrigerant conditions at gas cooler inlet

R744 mass flow rate	Inlet T & P	Partially blocked header condition (Mode A)			Unblocked header condition (Mode B)		
		0.6 m/sec	1.2 m/sec	1.8 m/sec	0.6 m/sec	1.2 m/sec	1.8 m/sec
15 g/sec	T [°C]	104.9	101.8	98.57	106.5	102.6	100.8
	P [MPa]	8.99	8.88	8.71	9.00	8.82	8.71
20 g/sec	T [°C]	110.2	106.8	102.8	111.1	105.9	103.7
	P [MPa]	9.39	9.29	9.06	9.31	9.15	8.98
25 g/sec	T [°C]	113.8	111.3	107.5	112.7	110	107
	P [MPa]	9.77	9.70	9.46	9.44	9.46	9.25

5. MODEL DESCRIPTION

A finite volume model was used to account for the change in the refrigerant properties along the heat exchanger. The model is based on the version described by Yin et al. (2001). The tubes in each slab were divided into 10 elements. Every element was described by equations for cross-flow heat exchanging conditions and the properties of the air and refrigerant at the outlet of each element were determined by the effectiveness-NTU method (1984). The conduction through the fins and tubes was ignored. Air distribution at the heat exchanger inlet surface was assumed to be uniform. The "minor pressure losses" in the headers were also ignored when calculating the pressure drop on the refrigerant side. Table 3 gives other correlations used in this model.

Table 3. Correlations used in the model

Item	Correlation
Refrigerant friction factor	Churchill (1977)
Refrigerant side heat transfer coefficient	Gnielinski (1976)
Airside heat transfer coefficient	Chang and Wang (1997)

The input variables for this model were: refrigerant mass flow rate, refrigerant inlet pressure and temperature, airflow rate, and air inlet temperature. The outputs were: gas cooler capacity, refrigerant exit pressure and temperature, and air exit temperature. The model was solved by EES and the properties of R744 and air were based on REFPROP6 provided by EES.

6. RESULTS AND DISCUSSION

6.1 Capacity of gas cooler

Figure 3 shows the capacity for the case of a 33% reduction in volume and the designed gas cooler. The effect of the blocked slab on the capacity has the same trend and nearly the same gradient, except for the case when the frontal air velocity is 1.8 m/sec in the volume reduced heat exchanger. The capacity is reduced by 5 - 8% due to the gas cooler volume reduction for a frontal air velocity 0.6 and 1.2 m/sec, while the capacity reduction is about 9 - 12% for a frontal air velocity of 1.8 m/sec. Figure3 shows that the gas cooler capacity increased slightly with an increase of air velocity form 1.2 to 1.8 m/sec for the volume reduced heat exchanger, while obvious capacity increase was observed for the designed heat exchanger. This trend can be explained by the refrigerant temperature at the gas cooler exit. For the test with a 1.8 m/sec air velocity for the volume reduced heat exchanger, the refrigerant outlet temperatures for the gas cooler were from 35.4 - 36.0°C. Considering the air inlet temperature varied from 35.2 - 35.3°C, it means that the temperature difference between refrigerant and air was extremely small. As a result, the capacity did not increase significantly as the air flow velocity was increased from 1.2 to 1.8 m/sec for the volume reduced heat exchanger.

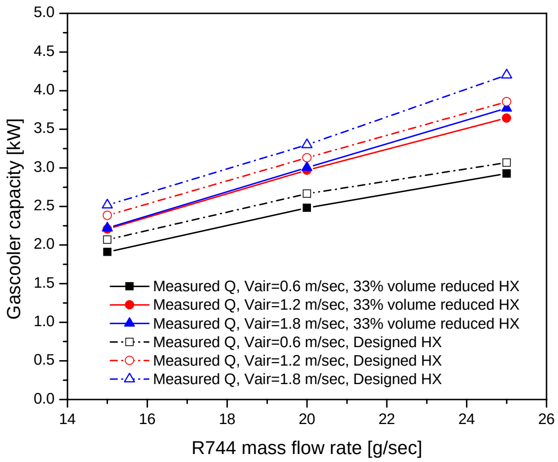

Figure 3. Experimentally determined gas cooler capacity for the volume reduced and designed heat exchanger

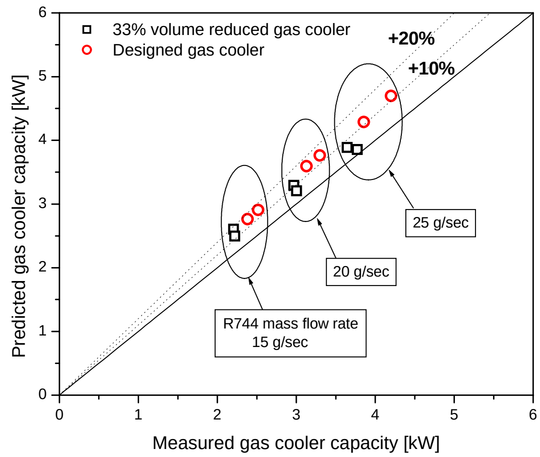

Figure 4. Measured vs. predicted gas cooler capacity for the volume reduced and designed heat exchanger

Figure 4 shows the comparison of the measured and predicted capacity for the volume reduced and designed heat exchanger. The models for each mode over-predicted the capacity of the gas cooler by 2.2 - 18.0% and 11.2 - 15.9% and the average deviation of the model results from measures capacity were 9.5% and 13.9% for the tests with the volume reduced and designed heat exchanger respectively. The prediction error for volume reduced heat exchanger is smaller than for designed heat exchanger because the volume reduced heat exchanger has a simpler configuration to model.

6.2 Analysis of gas cooler with simulation results

The analysis of the simulation results can give useful information for a better understanding of the gas cooler, although the model over predicted the gas cooler capacity. Figure 5 presents the refrigerant temperature and air inlet temperature to each segment and Figure 6 shows the change of capacity which was covered by each segment with respect to the gas cooler circuit length. The simulation conditions for the volume reduced and the designed heat exchanger were: R744 mass flow rate: 20 g/sec and frontal air velocity: 1.2 m/sec.

In Figure 5, the R744 temperature at the gas cooler inlet was 106.8 and 105.9°C, and the temperature at the outlet was 39.1 and 38.9°C for the volume reduced and designed gas cooler, respectively. The refrigerant temperature drops very quickly just after entering the gas cooler, and the temperature change gradient decreases as the R744 flows through the channels. For both test conditions, the refrigerant temperatures at the inlet and outlet were similar, while the temperature change has a different gradient especially in the first pass. Due to the larger refrigerant and airside area for designed gas cooler without volume reduction, the refrigerant can be cooled more quickly. This trend is apparent in Figure 5. The air inlet temperature for the first pass is higher than 35°C because the air heated in the second pass is supplied to the first pass.

As refrigerant temperature decreases more quickly, the temperature difference between the air and refrigerant becomes smaller for the gas cooler without the volume reduction as shown in Figure 5. Due to the decrease of temperature difference between the refrigerant and air for the designed gas cooler, its segment capacity cannot be much greater than that for the volume reduced gas cooler as shown in Figure 6. In the second pass, the capacity for each segment is larger for the volume reduced gas cooler because the temperature difference between the air and refrigerant is larger. The airside and refrigerant side area are the same for both test modes in the second pass. The sudden increase of the first segment capacity at the second pass inlet in Figure 5 is due to the increase of the temperature difference between refrigerant and air at this location.

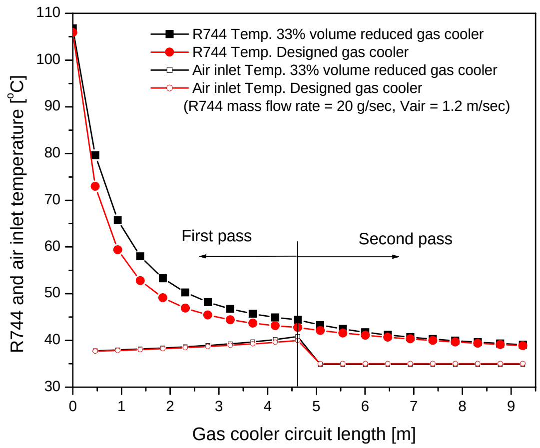

Figure 5. R744 temperature change in the gas cooler with respect to the circuit length

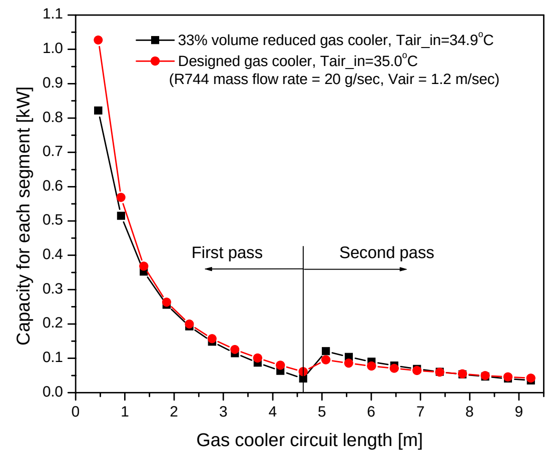

Figure 6. Capacity change for each segment in the gas cooler with respect to the circuit length

From the results of the model, the capacity ratios of the first and second pass are 3.7 and 4.6 for the volume reduced and designed gas cooler respectively. Although the gas cooler without the volume reduction has twice the larger air and refrigerant side area in the first pass, there was no significant difference for the two test modes in terms of the ratios of the capacity covered by the first pass to the total capacity (79% and 82% respectively) for the volume reduced and designed gas cooler.

6.3 Refrigerant pressure drop

Under the test condition of 15 g/sec R744 mass flow rate and 1.2 m/sec frontal air velocity, the measured refrigerant pressure drop was 694 kPa and 415 kPa for the volume reduced and designed gas cooler, respectively. This shows that the 33% volume reduction of heat exchanger gives 67% increase in refrigerant pressure drop, while the capacity decreases by 7.1% in this test condition. In this experimental study, a 50 - 67% refrigerant pressure drop increase and 5 - 12% capacity decrease occurred due to the volume reduction of the gas cooler. As a result, it can be concluded that the 33% volume reduction of the heat exchanger gave a relatively small capacity change compared with the capacity of the heat exchanger without the volume reduction, but it resulted in a much larger pressure drop.

6.4 Effect of gas cooler size on the system COP and gas cooler capacity

The experimental results show that the 33% gas cooler size reduction resulted in a 5 - 12% capacity decrease and a 50 - 67% pressure drop increase. Considering the volume reduction effect on the gas cooler capacity, the size reduction can be a reasonable way to manufacture a cheaper heat exchanger without a significant loss of capacity. However, an increase in refrigerant pressure drop causes a decrease in the system COP by increasing the compressor work.

In order to investigate the effect of the gas cooler size reduction on the system COP, a system model was developed and it was based on the heat exchangers tested in this study. The simulation was performed for the condition in which the R744 mass flow rate was fixed as 15 g/s. Two frontal air velocities, 1.2 m/s and 1.8 m/sec, were applied to the system model and the gas cooler inlet pressure was 9.1 MPa and 9.0 MPa for each frontal air velocity. Figures 7 and 8 show the variation of system COP and gas cooler capacity with respect to the change of the gas cooler size.

Figure 7 shows that the system COP was reduced between 8 - 9% due to the 33% volume reduction of the gas cooler for the two simulation conditions. The system COP decreases significantly as the volume reduction ratio becomes greater than 33%, and the COP does not increase as the gas cooler size is larger than the designed size. The 33% volume reduction caused a 10 - 11% capacity reduction for the gas cooler and this figure is within the range of

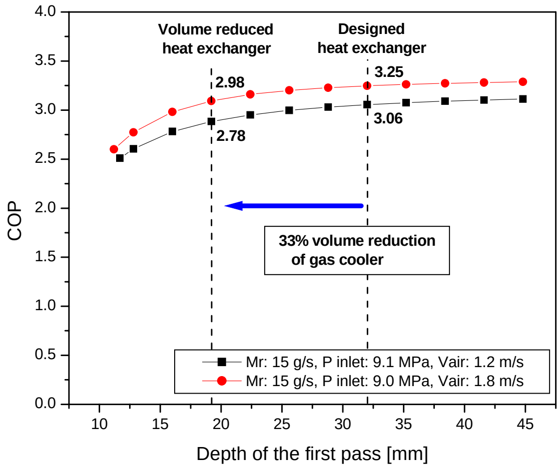

Figure 7. Effect of the gas cooler size on the system COP

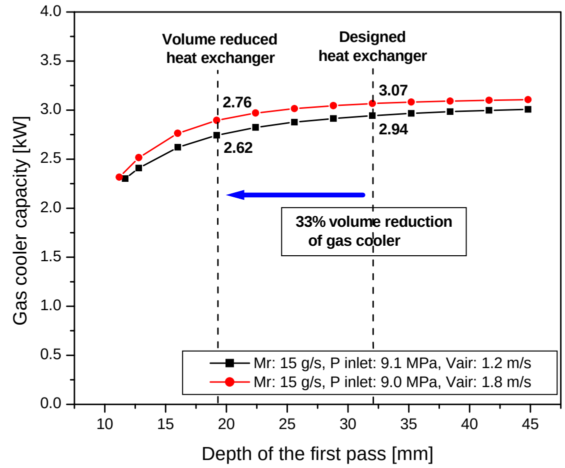

Figure 8. Effect of the gas cooler size on its capacity

the capacity reduction experimentally measured: 5 - 12%. The variation of the gas cooler capacity with respect to the change in size has similar a trend as that of the COP, and it is suggested in Figure 8.

The designed heat exchanger, which has the 2 slabs in the first pass, can be considered as appropriately designed. The heat exchanger volume, which is larger than that of the designed, cannot give a much better system COP or gas cooler capacity but the manufacturing cost increases. However, there is a possibility to reduce the heat exchanger size because the 33% volume reduction of the gas cooler caused the loss of 8 - 9% COP and 10 - 11% gas cooler capacity. These losses can be a small figure, considering the significant volume reduction of the heat exchanger.

7. CONCLUSIONS

An experimental and numerical study was performed to explore the effect of the reduction of the gas cooler size on its capacity and system COP. The 33% volume reduction of the gas cooler resulted in a 5 - 12% capacity decrease and a 50 - 67% increase in refrigerant pressure drop in comparison to those of the designed gas cooler without volume reduction. Gas cooler simulation results show that about 80% of the total gas cooler capacity was covered in the first pass. It means that the heat transfer at the close part of the gas cooler inlet significantly affects the performance. System simulation results show that the heat exchanger size was appropriately designed. They also present that the 33% volume reduction of the gas cooler caused a loss of 8 - 9% COP and 10 - 11% gas cooler capacity. As a result, there is possibility to reduce the volume of the originally designed heat exchanger.

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