

# Effect of Flash Gas Bypass on the Performance of R134a Mobile Air-Conditioning System with Microchannel Evaporator

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## ABSTRACT

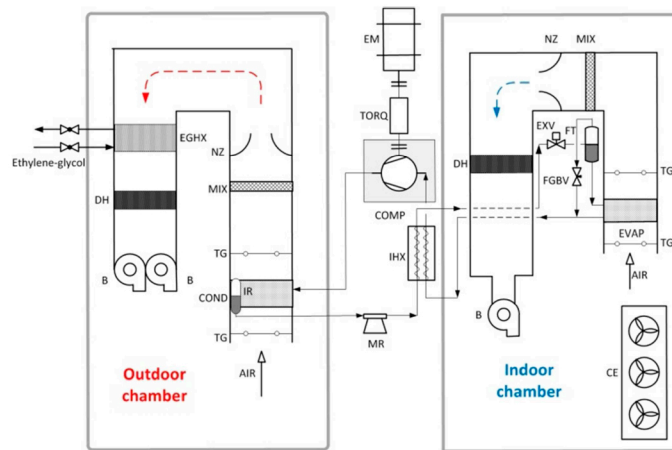
This paper demonstrates that the implementation of Flash Gas Bypass method can improve the performance of conventional direct expansion R134a mobile air-conditioning system with a microchannel evaporator. This method uses flash gas tank after expansion valve to separate and bypass flash refrigerant vapor around the evaporator, and feed the evaporator with only liquid refrigerant. Pressure drop is reduced and refrigerant distribution is significantly improved, resulting in higher evaporator effectiveness and evaporation pressure. Both lower pressure drop and lifted evaporation pressure allows the compressor to work with lower pressure ratio, saving required compressor work. An experimental comparison of the direct expansion system shows that Flash Gas Bypass method increases the cooling capacity and COP at the same time by up to 16% and 11%, respectively. Thus, the implementation of this method is very promising for the system using a microchannel evaporator which is especially very sensitive to the refrigerant side maldistribution.

## INTRODUCTION

In many air conditioning and refrigeration systems, the aluminum parallel microchannel heat exchangers have become widely used, since they aid in reducing system size and refrigerant charge. However, the use of a microchannel heat exchanger faces problems of refrigerant distribution in headers, especially for evaporators where the entering fluid is usually in a two phase state. Previous attempts to correct the problem have relied either on placement of baffles, flow constrictions, or other geometry-specific remedies [1, 2, 3, 4, 5, 6]. However, given the wide range of operating conditions for HVAC systems, particularly in automotive applications, these methods generally fail to produce uniform distribution

of the two-phase flow over the entire operating range and can lead to significant losses in the efficiency of heat exchangers. Some authors [7, 8, 9] suggested that another method is to use an atomizer in order to generate homogenous flow at the inlet of a header, but such solution, though providing improved distribution, may not be practical in a real mobile air conditioning (MAC) system.

Recently, another alternative approach is the implementation of a Flash Gas Bypass (FGB), which was described by Beaver et al. [10], Elbel and Hrnjak [11, 12]. The basic concept is to use a flash gas tank after expansion valve to separate and bypass flash refrigerant vapor generated in isenthalpic expansion process, and feed the evaporator with pure liquid refrigerant. Beaver et al. increased the system efficiency for 20% while maintaining same capacity of the prototype of the R744 residential A/C using flash gas bypass. Further, Elbel and Hrnjak demonstrated this approach was capable of improving transcritical R744 A/C system COP as well as cooling capacity in the following ways: reducing pressure drop in low pressure side, increasing local CO<sub>2</sub> heat transfer coefficient and improving refrigerant distribution. Due to the absence of compressor control capability in the abovementioned system, even larger improvements otherwise were possible in case a variable speed compressor was utilized to match the performance of the conventional DX system. Shikazono et al. [13] developed a series of compact vapor-liquid separators using surface tension for an R410A heat pump system. Results of cycle simulation indicated flash gas bypass is very effective for the improvement of heating capacity at very low temperature. However, these FGB methods were originally designed for residential cooling and heating applications using either R410A or CO<sub>2</sub>. Very recently, Milosevic and Hrnjak [14] investigated effect of using flash gas bypass on evaporator effectiveness and



**Symbols:** B-Blower; DH-Duct heater; CE-Ceiling fan; COMP-Compressor; COND-Condenser; DH-Duct heater; EGHX-Ethylene glycol heat exchanger; EM-Electric motor; EVAP-Evaporator; EXV-Electric expansion valve(Manual Control); FGBV-Flash gas bypass valve; FT-Flash gas tank; IHX-Internal heat exchanger; IR-Integrated receiver; MIX-Mixture; MR-Mass flow rate meter; NZ-Nozzle; TG-Thermal couple grid; TORQ-Torque meter.

**Figure 1. Layout of the experimental facility**

system efficiency for low pressure refrigerant R134a. Also, original compact design of flash gas separator was proposed, and its separation performance was tested under wide range of conditions. To the author's best knowledge, no study of effect of FGB method on performance of MAC system is available in open literature.

## EXPERIMENT SETUP

Figure 1 shows a schematic diagram of the MAC system in the experimental facility used in this project. It is the same as described in several SAE papers [15, 16].

MAC system consists of variable displacement compressor, a condenser with an integrated receiver, microchannel evaporator. In order to provide opportunity to explore effects of FGB, a single pass evaporator is installed in the system, accompanied with a tube-in-tube internal heat exchanger, a manual-controlled electric expansion valve, a compact flash gas separator, and flash gas bypass valve. Figure 2 shows the detailed configuration of microchannel evaporator with flash gas bypass control. The system can operate as conventional Direct Expansion (DX) mode when bypass valve is fully closed. When bypass valve is appropriately open, flash vapor is bypassed around the evaporator. The system operates as FGB mode. Ideally only pure vapor is flowing through bypass line, and liquid refrigerant is fed to the evaporator. The purpose of internal heat exchanger is mainly to protect the compressor from slugging when refrigerant exit evaporator in two phases and for the FGB mode some liquid refrigerant may be entrained with vapor due to inappropriate adjustment of bypass control valve. If system runs with evaporator outlet superheated and effective control strategy is applied to bypass valve, internal heat exchanger is not

necessary. More work to address this issue is being carried out but so far not included in this study. System performance of two modes are tested under the identical conditions and compared. The realization of the improvement potentials of FGB method is evaluated. In order to visualize the refrigerant distribution inside the evaporator, an infrared camera is used to record the surface temperature profiles of the aluminum evaporator. Quantitative results of refrigerant maldistribution are obtained by using refrigerant distributing rating method presented by Bowers et al. [17].

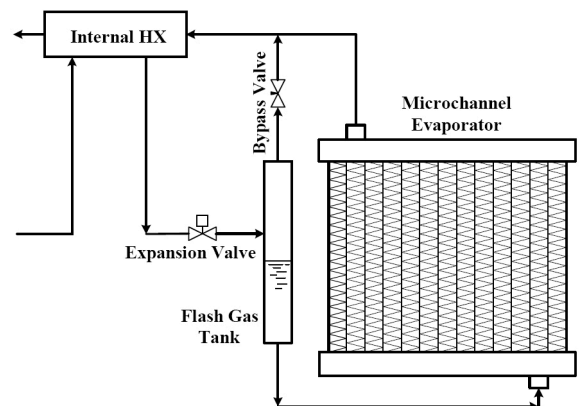
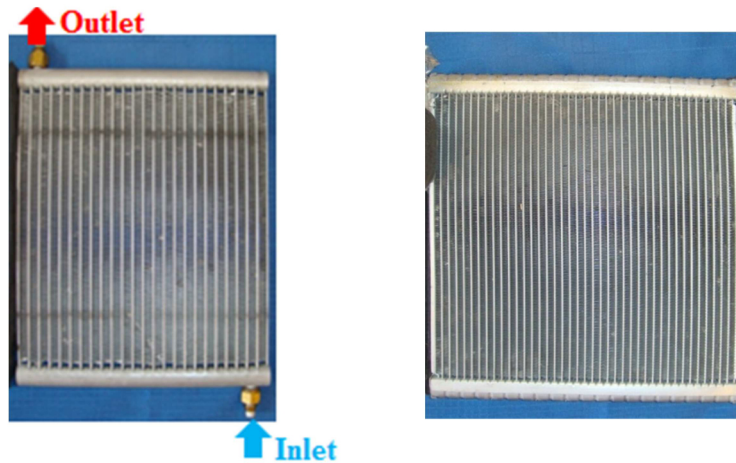


**Figure 2. Microchannel evaporator implemented with flash gas bypass control**

The evaporator is of aluminum microchannel tube design. In order to explore the adverse effects of evaporator maldistribution, a single pass and single slab evaporator is used in current system test, as shown in Figure 3(a). While this configuration does not meet the core volume or capacity requirements of the normal operating system, it is selected



(a) Tested single pass single slab evaporator (b) Denso multi-pass evaporator

Figure 3. Detailed structures of microchannel evaporators

because in single pass design distribution is completed in the inlet header. To compare with tested evaporator, a typical full size microchannel evaporator designed for MAC system is also shown in Figure 3(b), and their specifications are listed in Table 1. The air side area of tested evaporator is only about half of original Denso evaporator. Under identical operating condition, it needs larger heat transfer temperature difference than denso evaporator to produce the same cooling capacity, thus pulling down evaporation temperature. Even the extent to which FGB improves system COP with full sized evaporator is less than for undersized, the principle still holds. In addition, conventional microchannel evaporators with multi-pass design (like in Figure 3(b)) typically have higher pressure drop due to reduced refrigerant flow area. A desirable evaporator for FGB system should be a single pass and single slab with the same air side but maybe smaller channels than conventional MAC evaporator. The authors were not in position to have an appropriate design for this experiment.

Table 1. Evaporator Specifications

| Items           | Unit           | Denso Evap. | Tested Evap. |
|-----------------|----------------|-------------|--------------|
| Tube Length     | mm             | 255         | 255          |
| Overall width   | mm             | 267         | 233          |
| No. of channels | -              | 39          | 23           |
| Flow depth      | mm             | 32          | 20           |
| Fin height      | mm             | 5           | 8            |
| Fin pitch       | mm             | 1.25        | 1.56         |
| Fin thickness   | mm             | 0.05        | 0.1          |
| No. slab        | -              | 2           | 1            |
| Air side area   | m <sup>2</sup> | 3.2477      | 1.4876       |

## UNCERTAINTY ANALYSIS

Applicable standards require at least two independent energy balances to experimentally determine steady-state system performances. The test facility for this system was designed for even three independent methods, namely a refrigerant-side energy balance ( $Q_{indoor,ref}$ ), an air-side energy balance ( $Q_{indoor,air}$ ) according to SAE J2765 [18] and ASHRAE 41.2 [19] and a calorimetric chamber energy balance ( $Q_{indoor,cha}$ ). In this study, three energy balances are measured and calculated as following:

$$Q_{indoor,ref} = \dot{m}_{ref}(h_{ero} - h_{eri}) \quad (1)$$

$$Q_{indoor,air} = \dot{m}_{air}(h_{eai} - h_{eao}) \quad (2)$$

$$Q_{indoor,cha} = W_e - Q_{loss} \quad (3)$$

At the exit of the evaporator, refrigerant is still in two phase, thus based on energy balance of internal heat exchanger, refrigerant enthalpy at the evaporator outlet  $h_{ero}$  can be indirectly calculated as following:

$$h_{ero} = h_{rcpi} - (h_{cro} - h_{xri}) \quad (4)$$