JMC/RMS-1401a

R-421A FLOW RATE COMPARISONS

2014.01.13

prepared by

James M. Calm, P.E.
Engineering Consultant
10887 Woodleaf Lane
Great Falls, VA 22066-3003 USA

for

Refrigerant Management Services of Georgia, LLC (RMS of Georgia)
610 McFarland / 400 Drive
Alpharetta, GA 30004 USA

Please refer questions or comments on this document to:

James M. Calm, P.E.
Engineering Consultant
10887 Woodleaf Lane
Great Falls, VA 22066-3003 USA

phone: +1 (703) 636-9500
e-mail: jmc@jamesmcalm.com

RMS of Georgia
610 McFarland / 400 Drive
Alpharetta, GA 30004 USA

phone: +1 (770) 777-0597, +1 (800) 347-5872
e-mail: info@rmsgas.com

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

blank page

R-421A Flow Rate Comparisons

prepared by

James M. Calm, P.E.
Engineering Consultant

INTRODUCTION

The following discussion compares the mass and volumetric flow rates of refrigerants for aftermarket – not included in original equipment manufacturer (OEM) – use for refrigerant retrofits in air conditioners under idealized conditions. These flow rates adjusted by the individual latent heat of vaporization (and to a much lesser extent subcooling) determine the cooling capacity. The refrigerants addressed include R-407C, R-410A, R-421A, R-422B, R-422D, R-427A, and R-438A along with reference data for R-22. The "R-" number designations are standard identifiers as assigned in and according to ASHRAE standard 34.[1] No inference is intended that the seven cited refrigerant blends are the only retrofit options to replace R-22.

Refrigerant flow rates in a system depend primarily on the system design coupled with the density and latent heat of vaporization of the individual refrigerant. The flow rate can be expressed either by mass or volume and for either of them for the system refrigerating capacity (for example system kW or RT) or per unit capacity (per kW or per RT). Inch-pound (also called English units) refrigerating tons (RT or tons) also can be expressed as MBH (also MBtuh, sometimes MBtu/h, or thousand British thermal units per hour). The flow rates addressed herein are normalized to those for R-22 to simplify comparisons and to avoid both the ambiguities between system capacity versus per unit capacity and the need for repetition in dual units – metric (SI) and inch-pound (IP). As such, normalized flow rates of 0.8 and 1.1 indicate 20% lower and 10% higher, respectively, flow compared to R-22; a rate of 1.0 means the same as R-22.

MASS FLOW RATE COMPARISONS

Figure 1 shows the comparative mass flow rates for average evaporating (heat removal or cooling) and condensing (heat rejection) temperatures of 10 °C (50 °F) and 35 °C (95 °F), respectively. As shown, the mass flow per unit capacity (or for full system capacity) is approximately the same (less than 1% difference) for R-407C and R-410A compared to R-22 and 6-42% higher for the other refrigerants shown. Stated another way, similar mass flow rates of R-407 and R-410A yield nearly the same cooling capacity as with R-22, but the other refrigerants shown provide lower capacity for the same mass flow rates.

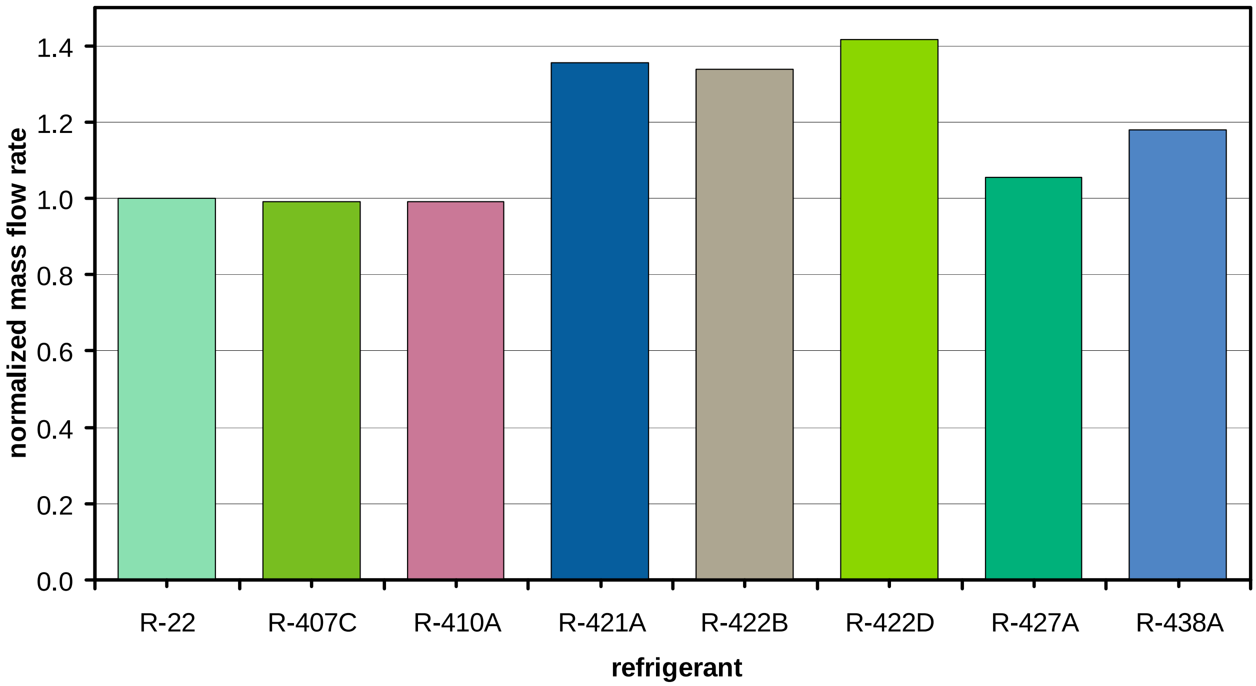

Figure 1: Mass Flow Rates at 10 °C (50 °F) Evaporating, 35 °C (95 °F) Condensing

VOLUMETRIC FLOW RATE COMPARISONS

That comparison may be misleading for retrofits as it implies systems designed (optimized) individually for each refrigerant. Retrofit is different. The compressor and other component sizes as well as connecting tubing diameters were selected for R-22. While R-407C and R-410A appear to require nearly the same mass flow as R-22, that is not the case without system redesign. The compressor displacement determines the volumetric flow rate (limited by other factors notably including refrigerant starvation by improper expansion valve sizing or control) for refrigerant replacement. The statement is true for all positive displacement compressors (for example rotary rolling piston, scroll, reciprocating piston, or screw), but may be easiest to visualize for a reciprocating piston compressor. When a piston drops in its cylinder, it opens a volume under vacuum that sucks in refrigerant through one or more suction valves. When the piston rises, the suction valves close and the piston compresses the refrigerant vapor to a higher pressure. The effective volume of the piston cylinder and the compressor speed determine the volumetric flow rate. The mass flow rate is related by suction pressure and density, but typically is not the same even on a normalized basis.

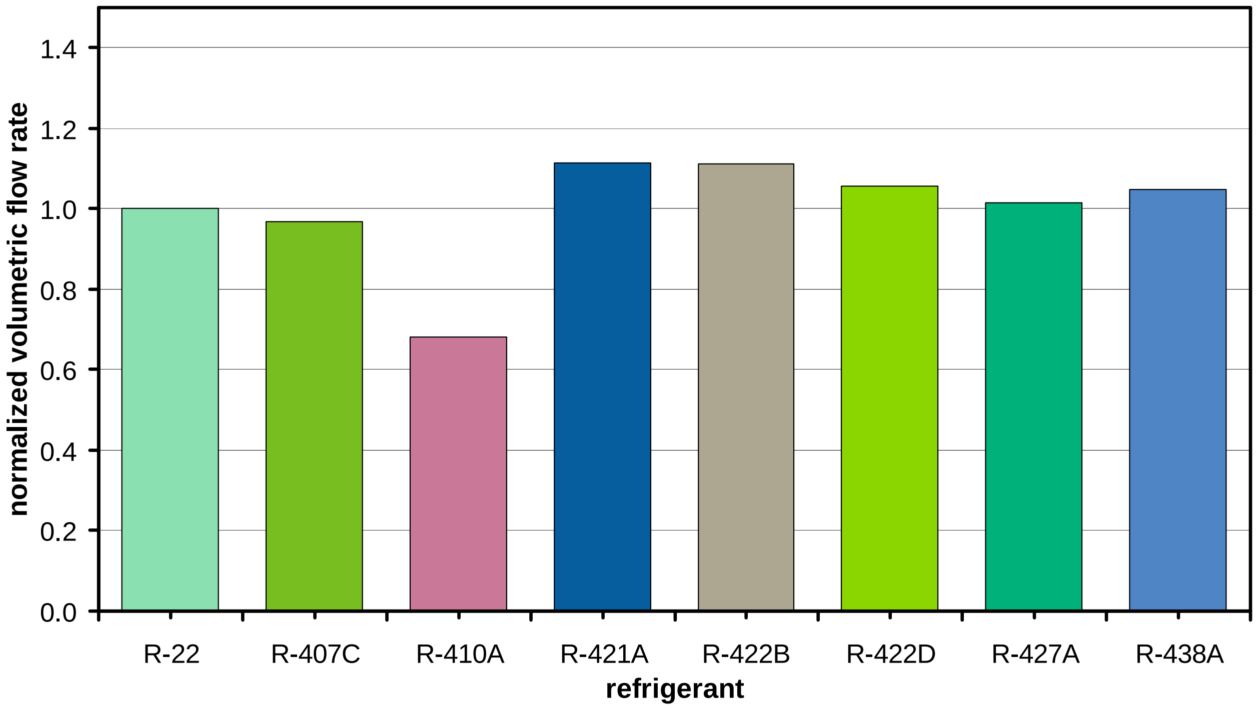

Figure 2: Volumetric Flow Rates at 10 °C (50 °F) Evaporating, 35 °C (95 °F) Condensing

While the required flow rate for R-410A, as an example, appeared nearly the same as R-22 on a mass basis, it is dramatically lower on a volumetric basis. That is the primary reason compressors had to be redesigned with 30-40% (32% for the conditions shown) smaller displacement for R-410A, the primary successor to R-22 in new air conditioners. The motor size remains unchanged for the same efficiency and capacity, but the piston diameter or stroke of the compressor is different. So without changing the compressor and other components in simple refrigerant retrofit, volumetric flow rate is more indicative for retrofits. As shown in Figure 2, the differences in flow rates for R-421A, R-422B, R-422D, R-427A, and R-438A are 1-11% by volume compared to R-22 – significantly lower than the 6-42% previously indicated by mass. The actual differences may be slightly higher or lower depending on the equipment and retrofit charge (and resulting refrigerant subcooling and superheating), but it is not impossible that the capacity (essentially flow multiplied by the corresponding latent heat of vaporization) would be the same or even higher particularly if the R-22 system was undercharged prior to retrofit.

Lower and higher capacity would translate to shorter or longer run time in compressor cycling for single-speed compressors for cooling loads below full capacity, but the energy use except at low loads would be more directly related to efficiency than small differences in run time. At high ambient conditions resulting in cooling loads exceeding the design capacity, the differences in refrigerant volumetric flow rate generally suggest similar differences in comparative cooling capacity, but as for R-22 in a properly designed system only for short periods of the cooling season.

ANALYSES CONDITIONS

The preceding flow calculations are based on ideal cycle analyses, using the National Institute of Standards and Technology (NIST) Cycle_D program, for conditions consistent with those used and discussed in prior studies by Calm and Domanski.[2,3] Theoretical cycle analyses for ideal conditions indicate the limits or comparative limits to attainable performance for simple cycles without regard to differences in equipment and component deviations from ideal, heat transfer, additional thermophysical properties, and lubricant differences. The resulting calculations indicate thermodynamic limits to what is attainable for different refrigerants in comparably optimized systems, but do not imply either that all systems will achieve such performance or that performance rankings of different refrigerants would not change in order of preference for systems not comparably optimized for the individual refrigerants.[3,4]

Table 1 summarizes the cycle conditions used for the analyses addressed herein. The evaporator and condenser temperatures, 10 °C (50 °F) and 35 °C (95 °F) respectively, reflect common rating conditions.

Table 1: Conditions for Performance Comparison

parameter	theoretical cycle limit for air conditioning
average evaporating temperature	10 °C (50 °F)
superheat	0 °C (0 °F) more rigorously expressed as 0 K (0 °F)
average condensing temperature	35 °C (95 °F)
superheat	0 K, 0 °C (0 °F)
compressor and motor efficiencies	
isentropic	100%
volumetric	100%
motor	100%
piping losses (drop)	
suction line	none: 0 K, 0 °C (0 °F)
discharge line	none: 0 K, 0 °C (0 °F)
suction line / liquid line heat exchanger	none (0%)

REFERENCES

1 *Designation and Safety Classification of Refrigerants*, ANSI/ASHRAE Standard 34-2013, American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), Atlanta, GA, USA, 2013

2 P. A. Domanski, D. A. Didion, and J. S. W. Chi, *CYCLE_D: NIST Vapor-Compression Design Program* (version 5.0), Standard Reference Database 49, National Institute of Standards and Technology (NIST), Gaithersburg, MD, USA, 2011.10.20 (and updates thereto)

3 J. M. Calm and P. A. Domanski, “R-22 Replacement Status,” *ASHRAE Journal*, 46(8):29-39, 2004.08; erratum, 46(10):8, 2004.10

4 J. M. Calm and D. A. Didion, “Trade-Offs in Refrigerant Selections: Past, Present, and Future,” *Refrigerants for the 21st Century* (proceedings of the ASHRAE/NIST Refrigerants Conference, Gaithersburg, MD, 1997.10.06-07), 6-19, American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), Atlanta, GA, USA, 1997