R152a Versus R134a in Domestic Refrigerator-Freezer -- Energy Advantage or Energy Penalty!

K. S. Sanvordenker
Tecumseh Products Research Laboratory

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By

Dr. K. S. Sanvordenker
Tecumseh Products Research Laboratory
3869 Research Park Drive
Ann Arbor, Michigan 48108-2290

ABSTRACT

Based on relatively simple computer modeling of refrigerator-freezer systems, R152a has been predicted to offer a degree of energy advantage over R134a as the working fluid. That impression was reinforced by a pamphlet put out by AHAM consortium summarizing laboratory calorimeter data taken by AHAM members, which showed an energy penalty of approximately 8% for R134a over R152a.

The implication of this penalty is that an additional 800 MW power plant will be needed when all the R12 refrigerators in the U.S. are converted to R134a rather than to R152a.

This author, in an internal memo of November 13, 1991, questioned the validity of the AHAM data, and presented calorimeter results showing no energy advantage for R152a over R134a; and in the process identified errors in the thermodynamic data on R152a as printed in the ASHRAE Handbook.

Because those findings were based on one compressor, data have been determined on five additional high efficiency compressors using R12, R134a and R152a. These data show that R152a not only offers no energy advantage but, to the contrary, there may even be an energy penalty under some operating conditions of a refrigerator-freezer, by the use of R152a over R134a.

INTRODUCTION

On October 3, 1991, U.S. Environmental Protection Agency (EPA) held a meeting with industry representatives to discuss an EPA-sponsored Risk Assessment Study on the use of flammable refrigerant 152a in domestic refrigerator-freezers. At that meeting, EPA presented a chart showing the relative efficiencies of R152a versus 134a. This chart was a summary of experimental calorimeter data taken by AHAM consortium member companies and showed that R152a would be 3.6% more efficient than R12 and that R134a would be 4.2% less efficient than R12. Together, there would be approximately 8% energy penalty if R134a were to be the working fluid for future refrigerator-freezers instead of R152a.

The implications of an 8% energy penalty are far reaching. A prominent EPA official states that this translates into an extra 800 MW power plant when all the U.S. refrigerators are converted to CFC alternatives. This not only means extra cost to the consumer, but also means emissions of additional carbon dioxide to exacerbate global warming.

If the AHAM chart is correct, and applicable to compressors for future refrigerator-freezers, then industry must seriously consider the use of R152a rather than R134a. This author, therefore, decided to examine the conclusions of the AHAM chart, and carried out calorimeter tests on a high efficiency reciprocating refrigerator-freezer compressor using R12, R134a and R152a as the working fluids. The results on that compressor are shown in Tables I and II.

SUMMARY OF PRIOR CALORIMETER DATA

Compressor Model: TP 1410 Y
Serial No.: 999 295
Lubricant: Polyol Ester
Thermodynamic Property Source: REFPROP 2.0

Table I: Evaporator - minus 10°F
Condenser - plus 130°F

Refrigerant	Capacity BTU/hr	Efficiency BTU/watt-hr
R12	1,162	5.06
*R134a	1,048	5.08
R152a	939	4.97

* A repeat test, a week after the preceding test showed repeatability within approximately one-half percent both for capacity and for efficiency.

Table II: Evaporator - minus 10°F
Condenser - plus 110°F

Refrigerant	Capacity BTU/hr	Efficiency BTU/watt-hr
R12	1,233	5.67
R134a	1,080	5.64
R152a	997	5.6

Even though the data are on one compressor, it is clear that the projected 8% energy advantage of R152a over R134a is suspect. Data in Tables I and II show that R152a has no energy advantage over R134a, and, if at all, it has an energy penalty over R134a.

ERRORS IN R152a THERMODYNAMIC DATA

During the course of explaining why the AHAM data were so far divergent from our calorimeter data, we identified an error in the boiling point of R152a as printed in the ASHRAE Handbook. It appears that the best measurements put the normal boiling point of R152a to be -25°C rather than -24°C as printed in the ASHRAE Handbook.

Theoretical models which may have used -24°C boiling point, and, for that matter, calorimeter data taken using the ASHRAE tables are likely to err in favor of energy efficiency of R152a, even though the error may not be very large. The thermodynamic data in the ASHRAE Handbook are now being corrected.

ARE DATA ON ONE COMPRESSOR REPRESENTATIVE!

Because of understandable skepticism that data from one compressor may not be reliable or for that matter may not be representative, it was decided to take similar data on ten more compressors of the same model. However, as experimentation progressed, it became evident that the data were remarkably consistent and repeatable. Hence, after testing five compressors, additional experimentation was considered unnecessary. Results of calorimeter data are summarized in Tables III and IV.

SUMMARY OF CURRENT CALORIMETER DATA

Table III: T evaporator: - 10°F
T condenser: + 130°F
Lubricant: Polyol Ester, ISO32 grade
Thermodynamic Property Source: REFPROP 2.0
Compressor Model: TP 1410 Y

Compressor		Capacity BTU/hr			Efficiency BTU/watt-hr		
Serial No.	Code	R12	R134a	R152a	R12	R134a	R152a
999685	A	1,201	1,098	980	5.22	5.24	5.24
449162	B	1,146	1,052	956	5.19	5.18	5.23
449163	C	1,115	1,024	925	5.09	5.12	5.07
999463	D	-	1,016	953	-	5.13	5.01
999386	E	-	1,048	963	-	5.34	5.17
Average		1,154	1,048	956	5.16	5.2	5.14
Normalized		1.1	1	0.91	1	1	1

Table IV: T evaporator: - 10°F
T condenser: + 110°F
Lubricant: Polyol Ester, ISO32 grade
Thermodynamic Property Source: REFPROP 2.0
Compressor Model: TP 1410 Y

Compressor		Capacity BTU/hr			Efficiency BTU/watt-hr		
Serial No.	Code	R12	R134a	R152a	R12	R134a	R152a
999685	A	1,257	1,169	997	5.89	5.93	5.64
449162	B	1,216	1,111	977	5.85	5.97	5.52
449163	C	1,203	1,096	947	5.83	5.73	5.37
999463	D	-	1,099	975	-	5.74	5.34
999386	E	-	1,130	993	-	5.9	5.55
Average		1,255	1,123	978	5.86	5.85	5.48
Normalized		1.09	1	0.87	1	1	0.935

A NOTE ON EXPERIMENTATION

The experimental setup was a simple "Secondary Refrigerant Calorimeter Method" of test conforming to ASHRAE Standard 23. The calorimeter was a prefabricated, packaged unit designed for small capacity (1500 BTU/hr max). The Standard rating conditions for determining the capacity and EER are -10°F evaporator, 130°F condenser, and 90°F subcooling and superheating entering the expansion valve and the compressor, respectively.

This test method has some peculiarities. For example, the suction and discharge conditions are controlled not by temperature, but by the saturation pressures of the refrigerant corresponding to those temperatures. Accurate PVT data are a must. Likewise, the suction pressure gages are compound gages which require the measurement of the barometric pressure (zero gage), which can fluctuate between a fair and a stormy day. The return gas temperature is typically close to the ambient (90°F); however, the subcooling to 90°F is not always easy to maintain. A correction factor is therefore applied - which requires accurate liquid heat capacity data. There has been some concern that the heat capacity data on R152a as printed in the ASHRAE Handbook may not be quite as accurate as desired.

The test sequence consisted of calorimeter-testing the compressor first with R134a, then, after a thorough evacuation, testing it with R152a, and again after evacuation testing it with R12. After this sequence, the compressor was again tested with R134a to assure that nothing went wrong with the compressor during the test sequences. Typically, the repeat results were well within 2% of the original.

DISCUSSION OF RESULTS

The purpose of this paper is to present new laboratory data on R134a versus R152a for domestic refrigerator-freezer applications. This discussion, therefore, will be kept to the minimum.

Tables III and IV summarize the calorimeter data. At both condensing temperatures, R152a shows approximately 10% lower capacity than R134a, which in turn shows a 10% drop as compared to R12. The capacity difference is consistent for all five compressors, although it appears to be more than predicted by simplified theoretical models.

At the rating conditions (Table III), the Energy Efficiencies for all three refrigerants is just about the same. The 8% penalty (projected from the AHAM data) for R134a over R152a is not substantiated. There is no energy penalty one way or the other.

Somewhat disturbing are data at 110°F condenser (Table IV). A refrigerator-freezer in the U.S. is more likely to operate at 110°F condenser than at rating conditions of 130°F. R152a shows a substantially lower energy efficiency than either R12 or R134a. The overall difference of 6.5% is far more than can be attributed to errors in measurement.

Viewed differently, comparison between Table III and Table IV shows that both R12 and R134a respond positively to the lower condensing temperature, approx. 13%, whereas R152a does not respond as well - only 6.5%.

It is not the intent of this paper to explain experimental data by using model predictions. However, some clues to the lower efficiency of R152a can be found in a recent report by P. Domanski, D. Didion and J. Doyle titled "Evaluation of Suction Line - Liquid Line Heat Exchange in the Refrigeration Cycle", where they model the effect of superheat for 25 refrigerants including R12, R134a and R152a.

The calorimeter measurement with 90°F liquid subcooling and 90°F vapor temperature entering the compressor is simulated by the Domanski model of suction line/liquid line heat exchange, and is also representative of a domestic refrigerator-freezer design.

The Domanski model results show that whereas R152a may be marginally more efficient than R134a without suction line/liquid line heat exchange, that marginal superiority disappears when superheat is factored in and, what is more, R134a becomes substantially more efficient than R152a. The same model also shows that the improvement of R134a is also affected by the lift, i.e., the difference between the condenser and evaporator temperatures.

The data shown in Tables III and IV, which show equivalent efficiency at rating conditions, and which show R134a to be significantly more efficient at the lower condenser temperatures are valid experimental results.

CONCLUSIONS

The earlier findings based on one compressor, that R152a has no energy advantage over R134a for domestic refrigerator-freezer applications, have been confirmed by additional experimental data with five more compressors.

The new data also show that under conditions where a U.S. domestic refrigerator-freezer is more likely to operate, i.e., at -10°F evaporator, 110°F condenser, there may be a significant energy penalty if R152a is used as the working fluid rather than R134a.