1994

New High Efficiency R-134a Compressor

R. Schmitz
Tecumseh Products Company

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R-134a COMPRESSOR

Ray Schmitz
Project Engineer
Tecumseh Products Company
Tecumseh, MI 49286
Telephone: (517) 423-8828
Telefax: (517) 423-8426

ABSTRACT

Over the last thirty years, few changes have taken place in reciprocating refrigeration and freezer hermetic compressors. Of course there have been design improvements and many new applications, but the basics have stayed the same.

DOE requirements have brought new energy standards for 1993 and, with most of that work behind them, manufacturers of white goods are concentrating on alternate refrigerant conversions and new energy questions for 1994 and beyond.

Since 1988 the development of hermetic compressor conversions and a new compressor design evolved through many stages from the chlorofluorocarbon (CFC) R-12 refrigerant to the hydrofluorocarbon (HFC) R-134a. Steps in the analysis and development process will be outlined to show how a once common energy efficiency range of 4.0-4.4 escalated to above 5.5 BTU/Watt Hour (Energy Efficiency Ratio= EER). Miscibility and solubility with R-134a and the lubricant of choice will be explored. Compatibility with compressor materials, lubricity and stability will be examined in order to better see chemically, mechanically, and thermodynamically the results of this endeavor.

INTRODUCTION

Even before the Montreal Protocol of 1987 as amended in Copenhagen in 1992 dictated CFC phaseout including R-12, the need for a new compressor to handle the new alternate refrigerants and new lubricants was very apparent. The household refrigeration industry, the target for the application of this new technology, pushed hard to make the transition from R-12 to R-134a. Some of work done here was used to refute R-152a's 8% energy advantage touted by the EPA. The combination of developing a new compressor using new lubricants and charging with a new refrigerant provided a challenge the result of which follows.

COMPRESSOR DESIGN

Since the traditional 4.5 BTU/Watt Hour R-12 compressor is unacceptable in the domestic marketplace, a new design was prepared from the experimental prototype testing employing fundamental design techniques and sound research information. It was hoped that it would fulfill all the requirements during the phaseout of R-12 but also evolve into a 6.0+ EER for the higher standards anticipated by 1998. See Figure 1 for efficiency advancements. Figure 1A is a cutaway drawing of the new design.

Crankcase design was configured for motor up assembly since tests indicated that this was one way to obtain higher efficiencies. Details included one piece main bearing, no outboard bearing, and symmetrical geometry in order to minimize thermal distortion and ease in assembly.

Crankshaft design was made with intentions of reducing the horsepower losses, minimizing crankshaft deflection, and close coupling of the eccentric so that an outboard bearing was not needed. The projected area of the journal was developed with an undercut to provide the ideal geometry to accomplish these goals.

In keeping with the criteria listed above, the calculations were run to select the correct eccentric journal width. This too resulted in the best combination of projected length and stiffness for the shaft and rod design.

Among the design parameters for high efficiencies are minimizing suction and discharge plenum heat exchange, minimizing clearance or re-expansion volume, and improving internal gas dynamics including discharge and suction muffler volumes some of which were cited by Siewert.[1] See the P-H diagram, Figure 2 showing the rating point of the compressor compared to the theoretical thermodynamic design.

The graph depicts the theoretical compressor cycle operating on R-134a with thermal audit data presented. It shows the departure from suction and discharge conditions to the cavity conditions measured based upon internal temperature and pressure information. The dashed lines show actual audit data as opposed to the solid theoretical cycle data. See table below for some key values. Note the temperatures of the cooler, dense gas to the cylinder which is a major contributor to the increased efficiency levels.

CONDITIONS	R-12	R-134a
Enthalpy at suction inlet	93.64	120.69 BTU/LBM
Suction cavity temperature	197.2	143.5 °F
Enthalpy at suction cavity	107.3	132.42 BTU/LBM
Enthalpy at discharge cavity	126.65	150.89 BTU/LBM
Discharge cavity temperature	332.2	248.4 °F
Enthalpy at discharge outlet	98.52	123.49 BTU/LBM

The suction muffler design of the entrance, standpipe design, internal baffles, and suction tube positioning are major contributors to efficiency improvements mentioned above. These considerations were used to make the final decision on the crankcase design including the pump suspension.

On top of the efficiency improvement, there was another positive side to this development. Traditionally, increases in efficiencies are accompanied by increased sound levels. However, in this case, lower sound levels prevailed in part with the computer-generated housing[2]. The telescoping housing with increased stiffness due to housing thickness has enabled the overall sound level to be reduced from 51 dbA to 48 dbA for a 1000 BTU/HR compressor. The suspension design reduces foot (mounting bracket) vibration to one of the lowest for this size of compressor.

Overall sound power level and suction and discharge pulses are measured while the compressor runs on one side of a coupled load stand in a semi-anechoic room. Six calibrated microphones working at a three foot radius send signals through a multiplexer to a sound power calculator. Computer, analyzer, and plotter provide 1/3 octave band plots, data tables and narrow band plots.

[1] Herbert Siewert, "The Evolution of the High Efficiency Two-Pole Hermetic Compressor", p. 1437 - Purdue Compressor Conference 1992.

[2] Dave Lowery U.S. Patent No. 4,384,635, May 24, 1983.

Pressure pulses too have been kept to a minimum. Discharge pulses are in the range of 1.0-1.2 psi (peak-to-peak). Besides the efficiency improvement with the suction muffler, the design reduces pulses associated with suction cavity gas resonances. Suction pulses are in the range of 0.4-0.6 psi (peak-to-peak). See Figure 3 for a comparison of the overall sound power levels for an R-12 compressor vs. the state-of-the-art compressor of the same capacity.

The valving mentioned earlier was designed to provide the least efficiency penalty due to re-expansion losses. Port sizes and geometry were selected to give the best flow characteristics and minimal re-expansion in part due to design and the other part due to manufacturing. The discharge valve is recessed to a minimum cross section to prevent deflection that could cause gas sealing problems. The valve is assembled with a backer (retainer) and spacer then riveted to the valve plate.

The suction valve is a valve leaf plate. The length of the valve and thickness contribute to the best combination of geometry and valve dynamics within conventional valve stress limits. With all these design features combined in the suction leaf plate and valve plate, there was additional efficiency improvement seen when progressing from prototype stage to production with the result larger capacities translating into higher efficiency per a given displacement. This was due in large part to the consistent fine blank valve plates with the valve seats which required no secondary operations to prevent valve leakage.

MOTOR DESIGN

Improved motor design was obtained from experimentation with the mix of wire size in the main and the auxiliary windings. The 20 frame stator utilizes maximum slot fill with wire lubricant and core bonding epoxy that have been approved for use with R-134a and polyol ester oils.

Rotor construction consists of two different laminations used to provide a counterbore for the rotor to ride upon the thrust washers and thrust bearing. The combination of sintered brass and cast aluminum counterweight provides the proper balance limiting unbalance forces transmitted through the housing and mounting springs to the mounting feet. The acoustic noise is loudest through the oil as verified by internal microphones used to record sound levels by position.

The use of a PTCSCR motor with a 15 μf run capacitor provides 100 psig equalized pressure starting capability and running efficiencies above 5.5 BTU/Watt Hour at -10° F evaporator and 130° F condenser. Since cabinet operating conditions are closer to -10° F and 100° F respectively, actual efficiencies are near 6.8 - 6.9 EER. Some day serious consideration will be given to change to this specified rating point.

LUBRICATION

PAG Problems

Early testing and development of R-134a with polyalkylene glycol synthetic oils (PAGs) was favorable making them the leading lubricant. They showed acceptable lubricity and miscibility. But, serious problems with hygroscopicity (moisture absorption) and partial miscibility with mineral oil used in the assembly of hermetic compressors and, thus a significant retrofit concern all indicated that an alternate lubricant be explored.

On top of these concerns, thermal stability[3] near the range of then traditionally designed R-12 compressors was another negative. At around 400° F, decomposition products including moisture, acidic components, carbon monoxide and carbon dioxide are all available to attack compressor internals. Some of the same concerns exist with esters as seen below.

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[3]Sanvordenker, K. S., Compressor Manufacturer's R-134a Applications

Polyol Ester Emergence

Although some of the first testing of the prototype compressors was accomplished using R-12 and mineral oil, the switch was made early to R-134a and 32 cSt polyol ester (POEs) oil for design validation. POEs exhibited hygroscopic rates at a factor of 10X and 4X less than the PAG Diol and Monoether. Moisture content needs to be controlled in part by the oil supplier and the compressor manufacturer in order to meet total moisture specifications. Today, charge values below 30 ppm are acceptable with some oil samples measuring as low as 15 ppm. This is accomplished with the employment of a good production drier system and with careful manufacturer and supplier handling and processing.

The class of POEs offer good lubricity and miscibility. The lubricant alone consists of highly polar compounds which tend to wet metal surfaces. This attribute alone can add some reliability since lubrication is critical right at start-up.

Our Research Lab screened many lubricants and arrived at several good candidates. After thorough testing, six oils were approved for R-134a commercial refrigeration applications. Experience and feedback has been favorable to date. The only low back pressure lubricant approved today is Mobil Arctic 15. See the approved commercial application oils.

ICI EMKARATE RL32S	PENRECO SONTEX SEZ-32, SEZ-22	EMERY 2927-A
MOBIL EAL ARCTIC 22A	WITCO SUNISO SL32, SL22	CASTROL INC. ICEMATIC SW-32

The screening process and testing of oils continues. At this time, manufacturers have not settled all supplier and viscosity issues. For that matter, there is still concern about additives to the base stocks. There have been some successes using 10 cSt oils, but suppliers are leaning toward 15 cSt lubricants to provide the extra margin of safety. One must remember the energy efficiency penalty is only about 0.1 BTU/Watt Hour between these two viscosities.

Stability concerns mentioned with PAGs are still present with POEs. Decomposition of POEs produces carbon dioxide, moisture, and acidity which will attack compressor internals. One difference is that POEs need an iron catalyst for this decomposition to occur. A metal passivator was developed to circumvent this condition. The use of this additive is selective. So, it is very important to specify the type of base stock used in the production of POEs. Incorrect base stock may have undesirable toxic fumes if subjected to very high temperatures.

OTHER COMPATIBILITY ISSUES

Some of the most important compatibility issues that embrace the lubricant and refrigerant are the chemical make-up, miscibility and solubility which have been stated above. Another lubricant-refrigerant relationship that had to be considered was the chemical reactivity. Fortunately, R-134a is relatively inert so decomposition is measured in ppm.

A close watch of the lubricant-refrigerant interplay is the effect on polymeric materials. In the case of this design, motor insulation, magnet wire lubricants, muffler construction, thrust bearing case, cluster block and terminal cover must withstand this environment.

Traditional chlorinated solvents, rust preventives and other processing fluids must not be present not only in the compressor components but also in the refrigeration system. Potential capillary tube plugging presents a most formidable challenge to the conversion to R-134a. One known case of one manufacturer's supplier using an unapproved drawing compound for condenser tube forming caused immediate concern about compressor integrity. Care must be taken to remove such processing materials from all system components.

Though the above example was traced to supplier not related to the compressor manufacturer, another incident was. After the design validation and endurance tests had been completed for the most part, two customer testing programs were interrupted by wholesale tube blockages.

Subsequently, infrared spectra comparisons from the two separate failures plus five independent material evaluations traced the cause to an unwanted retardant added by a plastic molder. Although there still may be other unforeseen compatibility problems as conversion moves ahead, it is hoped that the worst is over.

SUMMARY

All the known technical barriers have been overcome. The remaining issues that need to be answered are the final approvals of the base stock oils, selection of the additive packages and what viscosity limits. The bulk of the work is done. But, it must be clear that those remaining decisions must not jeopardize the integrity and reliability of the design.

REFERENCES

Heating, Ventilating, Air Conditioning Guide 1952, Chapter 36

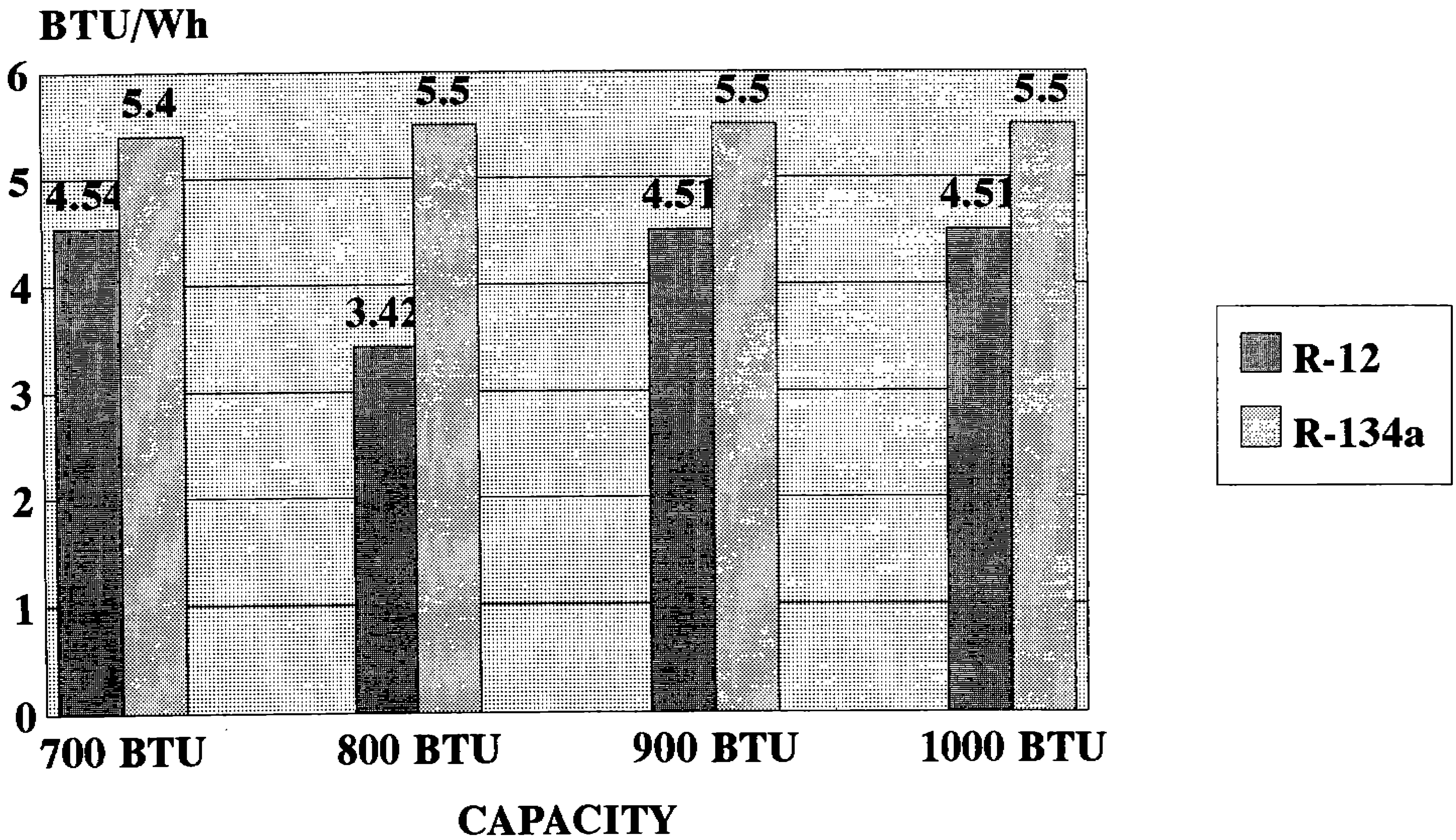
Modern Refrigeration And Air Conditioning, Althouse, Turnquist And Bracciano, Chapter 9

Sanvordenker, K. S., Energy Efficiencies Of HFC152a vs. HFC134a, November 1991

Sanvordenker, K. S., Materials Compatibility Concerns For Refrigerator - Freezers - A Historic Perspective And A Look To The Future, December 1992

ENERGY EFFICIENCY

FIGURE 1. FREQUENCY-60HZ



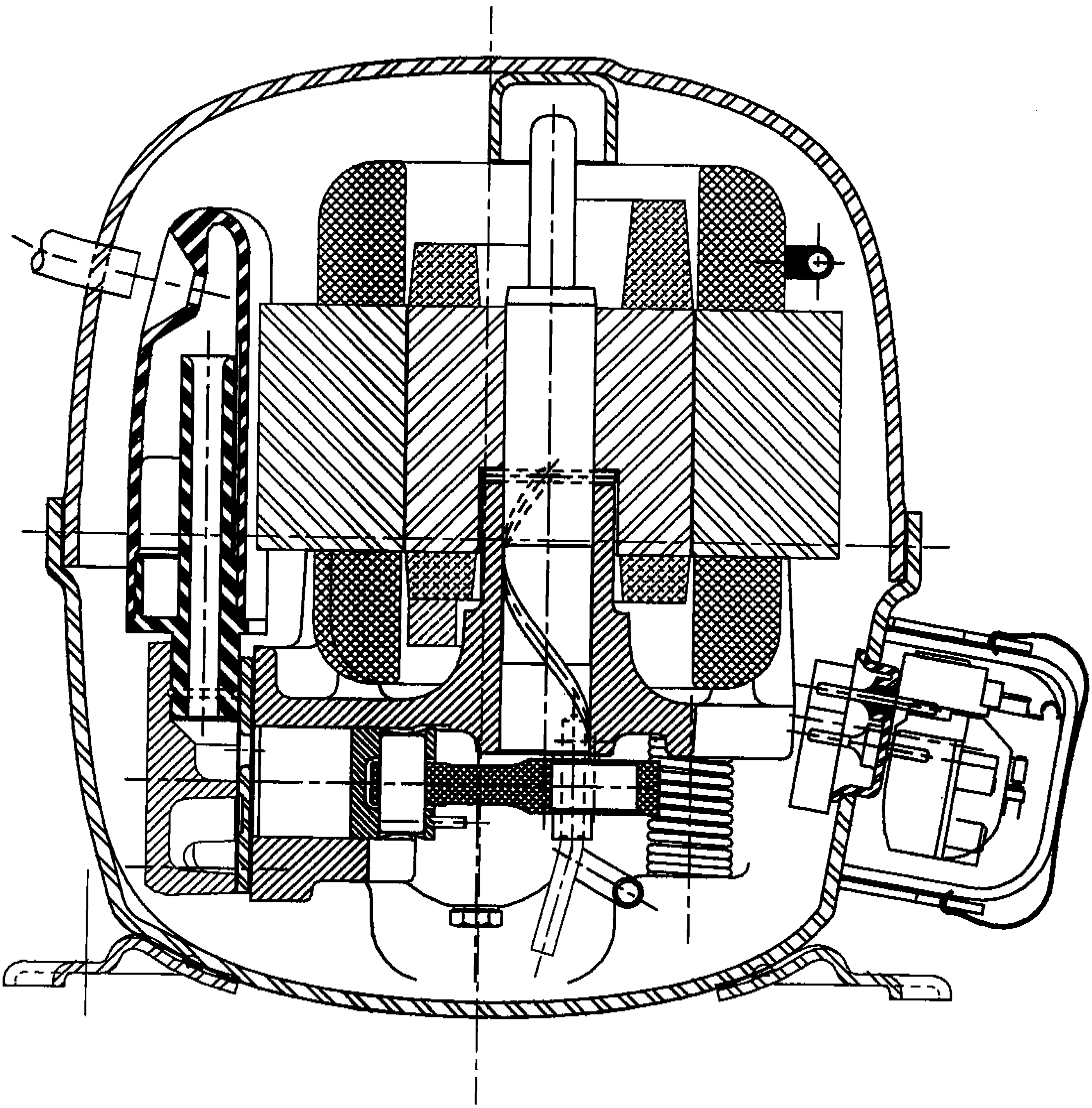
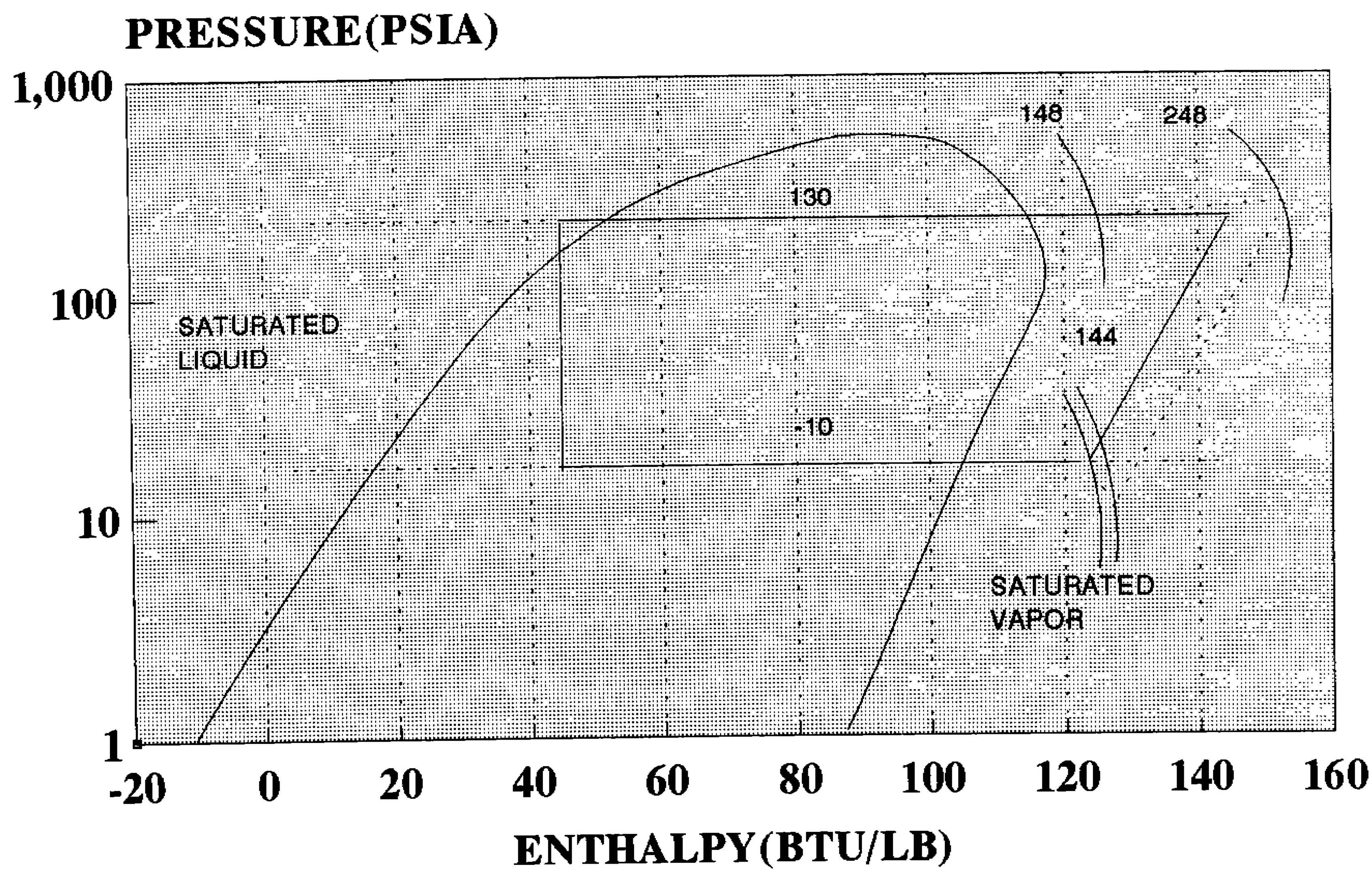


FIGURE 1A

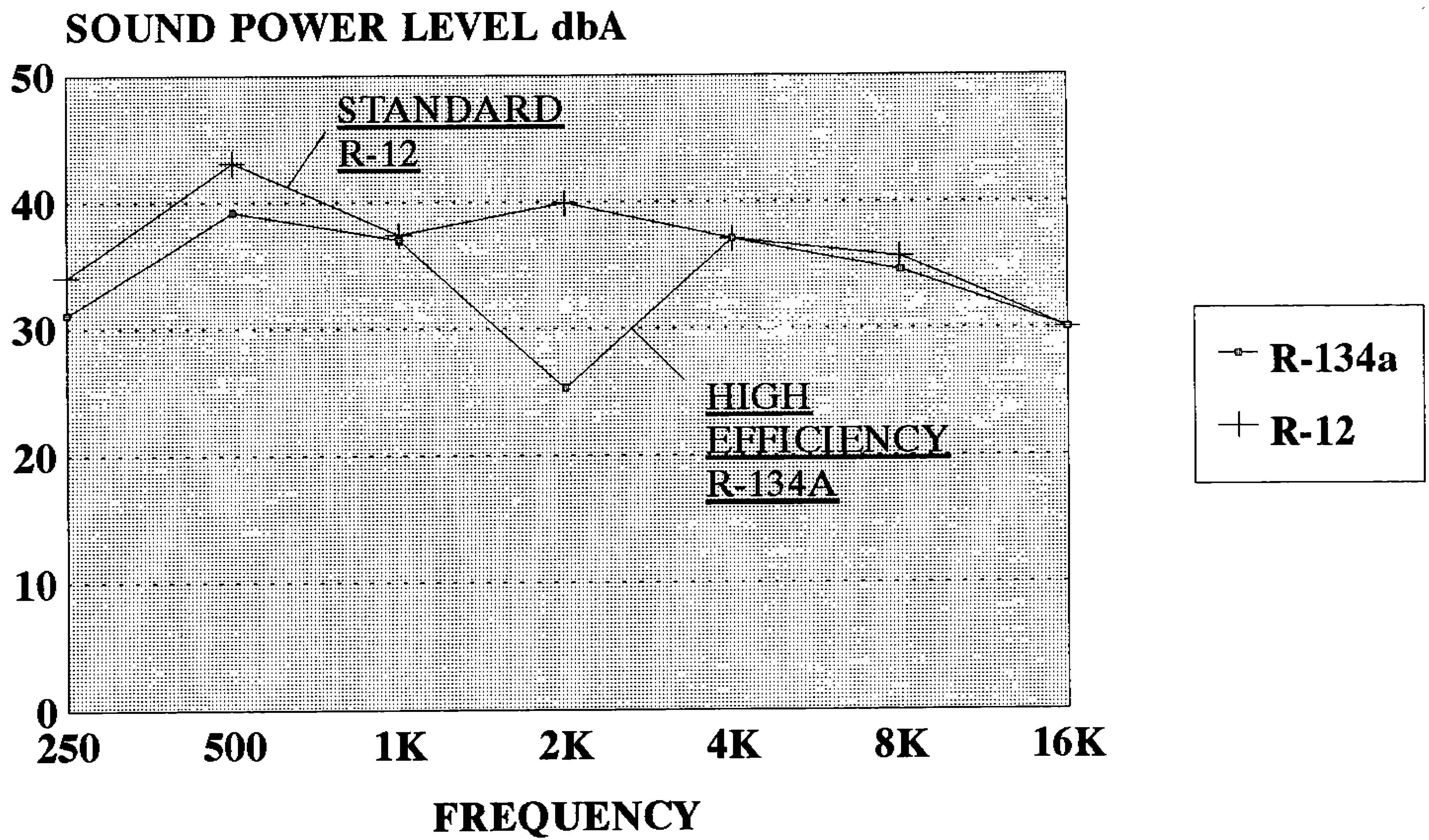
PRESSURE-ENTHALPY

FIGURE 2. R-134a



SOUND LEVEL COMPARISON

FIGURE 3.



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