United States Environmental Protection Agency
Research and Development

National Risk Management Research Laboratory
Research Triangle Park, NC 27711

EPA/600/SR-97/058 July 1997

EPA

Project Summary

Comparison of CFC-114 and HFC-236ea Performance in Shipboard Vapor Compression Systems

D. T. Ray, M. B. Pate, and H. N. Shapiro

In compliance with the Montreal Protocol and Department of Defense directives, alternatives to the refrigerant 1,2-dichloro-tetrafluoroethane (CFC-114) are being investigated by the U.S. Navy and the U.S. EPA for use in shipboard chillers. The refrigerant 1,1,1,2,3,3-hexafluoropropane (HFC-236ea) has emerged as a candidate for drop-in replacement.

A computer model was developed for comparing these two refrigerants in a simulated 440-kW centrifugal chiller system. Equations for modeling each system component were developed and solved using the Newton-Raphson method for multiple equations and unknowns. Correlations were developed for CFC-114 and HFC-236ea boiling and condensing coefficients taken at the Iowa State Heat Transfer Test Facility. The model was tested for a range of inlet condenser water temperatures and evaporator loads. The results are presented and compared with data provided by the Naval Surface Warfare Center (NSWC) in Annapolis, MD.

The experimental data provided by the NSWC sufficiently validate the model, and the simulation model predicts that HFC-236ea would perform favorably as a drop-in substitute for CFC-114.

Several recommendations are discussed which may further improve the performance of HFC-236ea in Navy chillers. Recommendations include adjusting the load of the evaporator to achieve positive gage pressure, use of a purge device, use of a variable speed compressor, further testing with azeotropic mixtures, and use of high performance tubes in the heat exchangers.

The work represented by the report was funded by the Department of Defense's Strategic Environmental Research and Development Program (SERDP).

This Project Summary was developed by EPA's National Risk Management Research Laboratory's Air Pollution Prevention and Control Division, Research Triangle Park, NC, to announce key findings of the research project that is fully documented in a separate report of the same title (see Project Report ordering information at back).

Introduction

Background

Fully halogenated chlorofluorocarbons (CFCs) are manufactured chemicals with properties that make them useful for such applications as aerosol propellants, foam blowing agents, solvents, and refrigerants for automotive, residential, and commercial applications. CFCs became popular in part because they were chemically stable, nonflammable, and nontoxic. Ironically, the chemical stability of CFCs is the cause for their present perceived threat to the environment. Scientific evidence suggests that the harmful alterations of the Earth' s atmosphere occurring from the use of CFCs are of regional and global proportions. As early as 1974, concerns about the potential harmful environmental effects associated with the use of CFCs were raised when it was suggested that the chlorine from these compounds could efficiently destroy stratospheric ozone. Additionally, there is a growing consensus among scientists that CFCs may contribute to global warming.

CFC-114 has been in use on Navy ships since 1969 and has demonstrated excellent reliability. However, design improvements have often lagged behind commercial advancements in compressor technology, advanced heat transfer surface technology, and intelligent control system technology. Trichlorofluoromethane (CFC-11), used extensively on commercial ships, was found to be unsatisfactory to the Navy because of problems unique to surface craft and submarine applications. For example, CFC-11 decomposes at high temperatures causing toxicity problems on submarines as the air is recycled in high-temperature air purification equipment. In contrast, CFC-114 remains stable at high temperatures.

Other requirements unique to the Navy include the need for small inventory and small components due to space constraints. Energy efficiency has been a low priority in the past, but with shrinking defense budgets, it has become more important. Additionally, surface craft and submarines need to operate silently in tactical situations, and recycle air in living spaces. Cooling systems must be able to operate at as low as 10% of maximum capacity during normal peacetime operations yet handle dramatic increases in load when firing weapons in combat or training situations. Fully halogenated refrigerants, such as CFC-114, generally exhibit the best compatibility and impose the least restriction in choice of materials; a suitable replacement must display similar material compatibility. Other requirements for a suitable replacement include meeting safety and environmental standards for toxicity, flammability, ozone depletion potential, and global warming potential.

A need exists for a suitable near-term replacement for existing equipment using CFC-114 . Because industry attention has been focused on CFC-11 and dichlorodifluoromethane (CFC-12) replacements, the Navy must devote substantial resources to address the CFC-114 problem. Current potential alternatives for CFC-114 are not well developed, and significant modifications to system equipment will likely be necessary in order to accommodate them.

At one time, 1,1,1-trifluoro-2,2-dichloroethane (HCFC-123) and 1,1,1,2-tetrafluoro-2-chloroethane (HCFC-124) were leading alternatives for CFC-11 and CFC-114, respectively. When it became apparent that these HCFCs would also be phased out as environmentally unsuitable, the EPA began investigating “backup” alternatives. As a result, a series of propanes have emerged as candidate replacements for CFC-114.

HFC-236ea is a promising candidate for replacing CFC-114 for several reasons. First, a commercial production route is available for large quantities through the use of hexafluoropropylene. Second, initial modeling conditions appear favorable as a drop-in substitute, with modeled performance being within 1% of CFC-114 and operating capacities, pressures, and temperatures matching closely. Flammability tests, materials compatibility tests, and oil miscibility tests appear favorable. Acute inhalation test results indicate lower acute toxicity than CFC-114. In addition, estimates predict that HFC-236ea has a short atmospheric lifetime.

Objective

The objective of this work is to evaluate HFC-236ea as a potential near-term replacement for CFC-114 in shipboard chillers using a computer-simulated system with data from the Iowa State University Heat Transfer Test Facility and also using data provided by the NSWC. With the information gathered, appropriate design recommendations to accommodate the use of HFC-236ea in shipboard chillers are offered.

Scope

A computer program has been developed that simulates the performance of a 440-kW capacity, single-stage, centrifugal, chilled-water air-conditioning plant. The design conditions shown in Table 1 are based on the design of a typical air-conditioning plant in use on Navy surface craft and submarines.

Given the entering and leaving temperatures of the chilled water, the entering temperature of the condenser water, and the flowrates of the chilled water and condenser water, the model predicts the required compressor power and the refrigerant saturation temperatures in the heat exchangers. With knowledge of fluid properties and tube geometries, the performance of the system with different refrigerants and enhanced surface tubes can be compared under similar operating conditions. For this study, the model is used to compare refrigerants CFC-114 and HFC-236ea using 10.23 fins per centimeter (fpc) tubes in the condenser and evaporator. The results are presented with design recommendations.

Model and Experimental Results Comparison

The NSWC, in cooperation with the U.S. EPA, has tested CFC-114 and HFC-236ea in a 440-kW laboratory centrifugal chiller representative of those used in the U.S. Navy’s fleet of surface craft and submarines. The laboratory chiller is fully instrumented.

Modeled and measured performance values were compared for both refrigerants. The measured compressor power provided by the NSWC was calculated from measurements of torque and angular velocity of the compressor shaft. The modeled value of compressor power is the rate of work performed directly on the fluid and does not include the mechanical or heat losses as the power is transferred from the compressor shaft to the impeller and ultimately to the working fluid. A linear relationship was found to exist between the shaft power and the power transferred to the working fluid for the data provided by the NSWC. The correlation was applied to the results of the model as an assumed mechanical efficiency.

Even with an efficiency factor applied, the model underpredicts the amount of compressor power required to meet the specified load by 6 to 26%. This could be related to the use of inlet guide vanes to the compressor which are not modeled in this study.

Table 1. Simulated Design Conditions

Component	Design Condition	Value
Evaporator	Chilled Water Flowrate	28.4 L/s
Evaporator	Entering Chilled Water Temperature	10.7°C
Evaporator	Leaving Chilled Water Temperature	7.0°C
Condenser	Water Flowrate	31.5 L/s
Condenser	Entering Water Temperature	31.4°C

A comparison of the system coefficient of performance calculated using NSWC's measurements and that predicted using the model developed in this study shows that the model overpredicts the coefficient of performance for HFC-236ea by as much as 38%. The result is consistent with what one would expect when comparing modeled with measured results. Since models often make use of simplifying assumptions, the results tend to be idealized. One would expect to see the actual results to be less favorable than modeled results.

A comparison of modeled and measured refrigerant temperatures in the condenser shows that the model predictions compare well with CFC-114 data; however, it underpredicts the condenser temperature for most HFC-236ea data by as much as 9°C. This difference could be caused, in part, by poor heat transfer in the condenser with HFC-236ea. If this were the case, it would take a higher condenser temperature to overcome whatever thermal resistance is present. More compressor power would be required to provide this additional temperature lift, resulting in lower system performance. Because the refrigerant temperature in the condenser is closely tied to the condensing coefficient by use of the log mean temperature difference equation for heat transfer in the condenser, one would also expect to see a difference in a comparison of the measured and modeled condensing coefficients. As the condensing temperature increases while entering and leaving, cooling water temperatures remain constant and the log mean temperature difference increases. This would result in a modeled decrease in the condensing coefficient.

An example of when conditions may exist in the condenser that hamper heat transfer is when noncondensable gases, left unpurged, accumulate in the upper vapor space of the condenser. This is a plausible explanation for the differences in condenser saturation temperatures observed. For HFC-236ea, both the measured and modeled evaporator temperatures are near 2°C. The corresponding saturation pressures for these saturation temperatures are less than atmospheric pressure. This could cause noncondensable gases to leak into the evaporator due to negative gage pressure. These gases would migrate and collect in the condenser and could significantly degrade the performance of the condenser and the entire system. If air, in fact, was present in the condenser, it would drive the outside heat transfer coefficient down, resulting in a high condenser saturation temperature.

A comparison of modeled and measured cooling water temperatures leaving the condenser shows that modeled values are within 0.2°C of measured values. This is consistent with trends observed for condenser capacity. This is expected, since the rate of heat transfer and the temperature of the water leaving the condenser are the two variables in the water-side heat transfer equation for the condenser.

A comparison of modeled and measured evaporator temperature shows that modeled and measured boiling coefficients compare well despite some variance. One would therefore expect to see a variance in a comparison of boiling coefficients since these variables must balance in the log mean temperature difference equation for the heat transfer in the evaporator.

A comparison of modeled and measured evaporator capacity shows that the values match because the capacity is calculated based on three common inputs: entering and leaving evaporator water temperatures, and evaporator water flowrate. Thus, the evaporator capacity remains fixed as long as these three input values are held constant.

A comparison of modeled and measured refrigerant flowrate shows that the model consistently predicts the flowrate for both refrigerants within ±5%. This suggests that the enthalpy differences also compare favorably since the rate of heat transfer in the evaporator is constant and is equal to the refrigerant mass flowrate times the enthalpy difference across the evaporator.

Comparison of CFC-114 and HFC-236ea Performance

The model is used to predict the performance of both refrigerants, CFC-114 and HFC-236ea, through a range of operating conditions. This is done by using the fleet design point as the default and varying one parameter at a time over an appropriate range to see the effects on the system. The results yield additional insight as to the possible suitability of HFC-236ea as a drop-in substitute for CFC-114.

Entering Condenser Water Temperature

As the Navy operates its fleet around the world, ships encounter a wide range of condenser water temperatures because sea water is used directly in the heat exchanger to remove heat from the working fluid. Chillers for Navy ships are designed for a condenser water temperature of 31.4°C; however, temperatures encountered may range from -1.3 to 35.3°C depending on where the ship is operating. Since heat transfer in the condenser is driven by the temperature difference between heat transfer fluids, a condenser water temperature that is too high could hamper the performance of the condenser and subsequently the entire refrigeration cycle. Thus, the entering condenser water temperature is significant to the performance of the overall system.

The predicted power required to drive the compressor more than doubles for both refrigerants as the water temperature entering the condenser increases from 16 to 38°C. The trend is expected since better heat transfer occurs as the temperature of the cooling water entering the condenser decreases. The efficiency of the refrigeration cycle should thereby improve, resulting in less power input required to the compressor. Additionally, the model predicts that HFC-236ea used as a drop-in substitute for CFC-114 may result in energy savings. At the design point of operation, the predicted power required to drive the compressor using HFC-236ea is 91.4% of the power required using CFC-114. The model predicts that, for any cooling water temperature, the power required for a refrigeration cycle using HFC-236ea as a drop-in will be significantly less than the same cycle using CFC-114 as the working fluid. The predicted savings in power consumption by using HFC-236ea at the design point of operation is 550 kW.

The NSWC data for CFC-114 show nearly constant compressor power over the range of entering condenser water temperatures. The data for HFC-236ea show significant scatter. Both the CFC-114 and HFC-236ea measured values of required compressor power are above the predicted values for the range of entering condenser water temperatures.

The coefficient of performance is predicted to decrease as the inlet condenser water temperature increases. Additionally, at the design point of 31.4°C, the predicted coefficient of performance for HFC-236ea is 4.25 compared to 3.91 for CFC-114. The model predicts better performance using HFC-236ea over the range of condenser water temperatures simulated.

The measured values of coefficient of performance for both CFC-114 and HFC-236ea are less than predicted values. As the entering condenser water temperature increases, the measured and predicted values of the coefficient of performance move toward better agreement.

Modeled trends predict that both performance indicators—compressor power requirement and coefficient of performance—improve as the inlet condenser water temperature decreases from 38 to 16°C since the required power consumption decreases and the coefficient of performance increases. These are expected trends since lower condenser water temperatures provide a higher temperature difference between the heat transfer fluids, resulting in increased cooling potential in the condenser. HFC-236ea is also predicted to outperform CFC-114 over a range of inlet condenser water temperatures, partly because measured heat transfer coefficients for HFC-236ea were found to be greater than those of CFC-114.

The predicted values for CFC-114 and HFC-236ea saturation temperature in the condenser for a given temperature of the cooling water entering the condenser are nearly identical. The measured values of condenser saturation temperature for CFC-114 agree with the predicted values, while the HFC-236ea data show the same trend but are generally higher than predicted.

The predicted saturation temperature for HFC-236ea in the evaporator as a function of the entering condenser water temperature is higher than predicted for CFC-114 over the range of entering condenser water temperatures. The HFC-236ea data compare well with predicted values, while there appears to be less agreement between measured and modeled values for CFC-114.

The evaporator capacity is compared with the temperature of the water entering the condenser. Since the evaporator capacity is fixed by holding the water-side conditions constant, the predicted capacity values for HFC-236ea and CFC-114 are identical. Scatter is shown in the measured values of HFC-236ea, while the measured values of CFC-114 agree well with predicted values.

The condenser capacity is also compared with the entering condenser water temperature. Predicted values for CFC-114 and HFC-236ea are nearly equal. For both CFC-114 and HFC-236ea, measured capacity values are higher than predicted, with significant scatter observed in the HFC-236ea data.

The refrigerant mass flowrate is compared with the temperature of the water entering the condenser. The measured and predicted values for CFC-114 are in close agreement, while there is significant scatter in the data for the flowrate of HFC-236ea. As condensing water temperature increases, an increasing trend is predicted in refrigerant mass flowrate.

Entering Evaporator Water Temperature

In this situation, the evaporator load is defined by a constant chilled water flowrate of 28.4 L/s, a chilled water inlet temperature of 7°C, and a chilled water exit temperature ranging from 9.2 to 12.6°C. Additionally, the temperature of the water entering the condenser is held constant at 31.4°C, and the flowrate of the condenser water is held constant at 31.5 L/s. As the cooling load is systematically varied, the effect on various performance indicators may be observed.

As the temperature of the chilled water returning from the load and entering the evaporator increases while other design operating conditions remain constant, increasing power is required to drive the compressor. This is an expected trend because, as the water temperature entering the evaporator increases, the load is increased in the evaporator. In order to accommodate this increased load, either the refrigerant mass flowrate must increase or the enthalpy difference across the evaporator must increase in order to provide enough heat transfer to maintain a constant chilled water exit temperature. The result is the need for more power to drive the compressor. A comparison of HFC-236ea and CFC-114 shows that, for the compared range of chilled water temperatures entering the evaporator, HFC-236ea always required less compressor power than CFC-114 when modeled as the working fluid.

A comparison of the coefficient of performance as a function of chilled water temperature entering the evaporator shows that as the temperature increases the coefficient of performance decreases. This means that as temperature increases, the rate of increase of power required by the compressor is greater than the increase in cooling capacity. Additionally, the coefficient of performance for HFC-236ea is higher than that for CFC-114 for the range of temperatures modeled.

Design Recommendations

One way to improve the performance of the fleet's 440-kW chiller and allow the use of HFC-236ea as an alternative working fluid is to reduce the load on the evaporator by increasing the temperatures of the chilled water entering and leaving the evaporator by a few degrees. This would allow the refrigerant temperature in the evaporator to rise slightly, which, in turn, would result in an evaporator pressure that is above atmospheric pressure. With a positive gage pressure in the evaporator, there is less possibility of noncondensable gases and contaminants leaking into the system where they can accumulate in the condenser and reduce performance. The low evaporator temperatures and high condenser temperatures reported by the NSWC suggest the possibility of this occurrence. This solution avoids the cost of redesigning system components.

A purge device at the high point of the condenser would allow purging of noncondensable gases that might accumulate there. If noncondensables are a persistent problem, the purge unit may be malfunctioning or the system may have an air leak larger than the purge unit can handle.

A variable-speed compressor would eliminate the need for hot gas by-pass or the extensive use of inlet guide vanes in the compressor to control the refrigerant flow. A variable-speed chiller would allow maximum system performance to be realized over a broad range of operating conditions, resulting in maximum energy savings.

Another possible improvement might be to mix HFC-236ea with other non-CFC refrigerants to form an azeotropic mixture with properties that allow the saturation point in the evaporator to stay above atmospheric pressure. The mixture could be chosen so as to maintain the desired properties of HFC-236ea.

Additionally, better performance in the Navy's fleet air-conditioning units could be realized by investing in commercially available high performance heat exchanger tubes. While not reported in this study, Turbo B tubes were simulated with CFC-114 and HFC-236ea under fleet design conditions and were predicted to perform significantly better than 10.23 fpc tubes in both the evaporator and the condenser.

Finally, the model predicts that HFC-236ea used as a drop-in substitute for CFC-114 without any design modifications may result in energy savings. The model predicts that, for any set of conditions, the power required for a refrigeration cycle using HFC-236ea as a drop-in will be significantly less than the same cycle using CFC-114 as the working fluid. The predicted savings in power consumption by using HFC-236ea at the design point of operation is 8.6%. If HFC-236ea is to be used only as a near-term replacement, it may be appropriate to use it without making any significant design changes to the system.

Conclusion

The Montreal Protocol began a worldwide drive to eliminate the production of CFCs and other chemicals which are thought to be harmful to the environment. As a result of the restrictive legislation that followed, there is an immediate need to replace CFC-114 which is used extensively in U.S. Navy surface craft and submarines. Preliminary research conducted by the EPA suggested that HFC-236ea might perform suitably as a near-term drop-in replacement for CFC-114. However, at the time of this study, heat transfer data for HFC-236ea were not available.

For this reason, a computer model was developed for comparing CFC-114 and HFC-236ea in a simulated 440-kW laboratory centrifugal chiller system representative of those found in the U.S. fleet. The model is semi-empirical, combining thermodynamic and heat transfer theory, as well as boiling and condensing heat transfer coefficient data measured at the Iowa State University Heat Transfer Test Facility.

The NSWC in Annapolis, MD, also provided data for this study. A 440-kW laboratory centrifugal air-conditioning plant and HFC-236ea were used for the data collection. The experimental data provided by the NSWC were compared with the modeled results.

The model was tested for a range of inlet condenser water temperatures, entering and leaving chilled water temperatures, and evaporator and condenser water flowrates. The simulation model predicts that HFC-236ea would perform favorably as a drop-in substitute for CFC-114 .

Additionally, several recommendations were provided for improved performance using HFC-236ea in centrifugal chiller systems. Design recommendations discussed in this study include manipulating the evaporator load to achieve positive gage pressure in the evaporator, ensuring the absence of noncondensable gases in the system, using a variable-speed compressor with a fixed inlet guide vane angle to the impeller, conducting further research using azeotropic mixtures with HFC-236ea as the major component, and installing high-performance enhanced surface tubes in both the evaporator and the condenser.

In summary, the simulation developed in this study provides results that are consistent with the expected behavior of a 440-kW refrigeration system. The results provided by the NSWC sufficiently validate the model. Finally, the results suggest that HFC-236ea would perform well in existing CFC-114 centrifugal chillers, although design modifications should be considered for optimal performance.

D. T. Ray, M. B. Pate, and H. N. Shapiro are with Iowa State University, Ames, IA 50011.
***Theodore G. Brna** is the EPA Project Officer (see below).*
The complete report, entitled "Comparison of CFC-114 and HFC-236ea Performance in Shipboard Vapor Compression Systems," (Order No. PB97-178735; Cost: $25.00, subject to change) will be available only from:
National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
Telephone: 703-487-4650
The EPA Project Officer can be contacted at:
Air Pollution Prevention and Control Division
National Risk Management Research Laboratory
U.S. Environmental Protection Agency
Research Triangle Park, NC 27711

United States
Environmental Protection Agency
Center for Environmental Research Information
Cincinnati, OH 45268

BULK RATE
POSTAGE & FEES PAID
EPA
PERMIT NO. G-35

Official Business
Penalty for Private Use $300

EPA/600/SR-97/058