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Project Summary

A Modeling and Design Study Using HFC-236ea as an Alternative Refrigerant in a Centrifugal Compressor

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A centrifugal compressor—part of a chlorofluorocarbon (CFC)-114 chiller installation—was investigated operating with the new refrigerant hydrofluorocarbon (HFC)-236ea, a proposed alternative to CFC-114. A large set of HFC-236ea operating data, as well as a limited number of CFC-114 data, were available for this study. It was determined that the compressor performance can be successfully described by a relatively simple analytical compressor model. Two compressor models, the first of which was obtained from the literature, were developed on the basis of thermodynamic analysis and by utilizing the database. Two empirical relations were required to predict mass flow rate and the refrigerant state at the compressor exit for each model. The second model is based on empirical relations derived directly from the data base rather than the general empirical relations used in the first model. The literature model had to be optimized for two parameters and corrected for the influence of the inlet guide vanes to yield results comparable to the newly developed model. Both models were based on the HFC-236ea data, and they exhibited systematic errors when used with CFC-114, which indicated the models' dependence on the refrigerant. Both models predicted refrigerant state at the compressor outlet excellently (±2.8°C), while the mass flow rate was predicted with larger differences to the data (±20%). In addition to being quantitatively superior, the new model has more physical relevance; therefore, it was used for the design analysis. In the design analysis, the compressor geometric parameters were varied for constant operating conditions seeking trends in compressor performance. It appeared that the compressor geometric parameters were appropriately chosen in the compressor design. Also, the model indicated a valid physical behavior since all of the trends in the compressor performance were readily explainable. The project was sponsored by the Strategic Environmental Research and Development Program (SERDP).

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Introduction

U.S. Navy surface ships and submarines are equipped with air-conditioning installations that have centrifugal chillers operating with CFC-114 refrigerant. The Environmental Protective Agency (EPA) in cooperation with the Navy has been seeking a CFC-114 drop-in replacement. Therefore, they sponsored a number of research projects related to the CFC-114 replacement in the mentioned chillers. One alternative refrigerant that satisfies many physical and chemical characteristics for use in the Navy fleet was found to be HFC-236ea refrigerant. Since HFC-236ea has very similar thermodynamic characteristics to CFC-114, it was presumed that the performance of a Navy chiller operating with this refrigerant would not change significantly. Hence, this project represents a part of the investigation directed to evaluate this CFC-114 alternative refrigerant as a possible drop-in replacement in a Navy chiller.

The U.S. Navy has an experimental chiller test facility in Annapolis at the Naval Surface Warfare Research Center. This chiller installation, which is typical for the Navy fleet, is well instrumented and can generate experimental data corresponding to different chiller operating conditions. The data have been gathered over several years with several different refrigerants. From this large database, the entire data set of HFC-236ea and several CFC-114 operating points were accessible to be investigated in this project. The data were already taken before this project started; therefore, the authors neither have any influence in setting up the experimental installation, nor any insight in the quality of the data recorded.

The objective of this study was to conduct a thorough literature review regarding centrifugal compressors and then, on the basis of the information gathered, build an accurate but simple compressor model using the available compressor experimental data. Further, the developed compressor model would be used to suggest eventual design adjustments to enhance compressor performance with the alternative HFC-236ea refrigerant.

Chiller (Compressor) Performance

The centrifugal compressor investigated in this study is an open, single stage compressor with a vaneless diffuser. The compressor performance was controlled by:

- Using the variable gear box to vary the compressor shaft speed. The compressor Mach number for the generated data was between 1.5 and 1.8.
- Using the inlet guide vanes to fine control the refrigerant flow rate (capacity). Also, the guide vanes can extend the stable compressor operating range, which is the useful range of compressor operation between the surging and choking limits.
- Bringing a certain amount of the refrigerant leaving the compressor to the compressor inlet through a by-pass loop where it is mixed with refrigerant leaving the evaporator. This crudely controls the refrigerant flow rate through the installation (the chiller capacity).

Studying the chiller operating points, it was concluded that the available data are a broad base for compressor model development. As an example, the chiller capacity was varied between 20% and 140% of the design capacity, while the inlet guide vane settings were altered between 10 and 90 degrees.

Nevertheless, the data had several limitations, of which the most important concerned the refrigerant flow rate. The flow rate was not directly measured on the installation, but it was determined from the energy balances on the heat exchangers. Also, the total power consumption and the amount the by-pass valve was opened were not recorded from the entire data set. The CFC-114 data were scant, which prevented a complete comparison between the two refrigerants.

Compressor Energy Model

The losses occurring in the energy transfer from the compressor shaft to the refrigerant are approximately constant. Quantitatively, these losses are around 1% of the total power delivered to the compressor shaft. Since the energy losses are constant, the compressor shaft power is modeled as a linear function of the energy transferred to the refrigerant. Thus, the compressor shaft power can be very successfully estimated by knowing the compressor refrigerant mass flow rate and the enthalpy difference across the compressor.

The CFC-114 data indicated lower energy losses within the compressor than the HFC-236ea operating data. Consequently, the HFC-236ea compressor performance line, relating the compressor shaft power with the amount of energy transferred to the refrigerant in the impeller, is different from the CFC-114 performance line. This was the first indication that compressor modeling is not invariant to the operating refrigerant.

Compressor Performance Map

A standard form performance map for the investigated compressor was provided by the compressor manufacturer, although the map was constructed for the compressor operating with a refrigerant different than the refrigerants examined in this study. A feasible relation between the performance map and the actual compressor operating points was impossible to find. The experimental data, when plotted in the form of the compressor performance map, vaguely resembled a compressor map. Therefore, it was inferred that the modeling of the compressor performance should be sought in some other way.

Compressor Modeling

Based on a thorough literature review, it was decided to describe the compressor performance thermodynamically by regarding the compressor as a control volume. This modeling approach considers overall compressor performance characteristics rather than exact flow field information within the compressor. Further, a centrifugal compressor must be considered to be two entities, the impeller and the diffuser. Since these compressor parts are treated as separate control volumes, the refrigerant state between the impeller and the diffuser must be determined.

Basically, the model determines the refrigerant flow rate and the refrigerant state at the compressor exit when the following are entered: the refrigerant state at the compressor inlet, the inlet guide vane setting, the refrigerant pressure at the compressor exit, and the compressor shaft speed.

With the model output known, the compressor shaft power can be determined. The compressor models are based on empirical relations. For each model output magnitude, one empirical relation is required; thus in this model format two empirical relations were required for each investigated compressor model.

Existing Model

A centrifugal compressor model was extracted from a chiller simulation that was found in the literature. The model complied with the desired model input/output format, and was built on two generic empirical relations based on an extensive database for similar centrifugal compressors. However, these empirical relations were based on compressors without inlet guide vanes.

When initially solved, the model greatly overestimated the refrigerant flow rate; therefore, the model was subjected to an optimization procedure. The flow rate errors were minimized using available data for two model parameters. When the optimized parameters were incorporated in the model, the model output yielded reasonable output values. One optimized parameter, the blockage factor, was found to be physically unreasonable. In addition to the doubtful physical validity of the model, another shortcoming of the existing model is the requirement to input the compressor polytropic exponent.

Further, the compressor model flow rate prediction was corrected for the influence of the inlet guide vane angles using the data available. The flow rate was predicted within ±20% relative difference between the measured and the modeled flow rates, while the exit state temperature was estimated within ±2.8°C of the measured temperatures for most of the HFC-236ea data points. The CFC-114 data points indicated the presence of a systematic error, implying that the compressor model is dependent on the particular refrigerant.

The New Model

The main difference between the new model and the existing model is that the available experimental data were used to generate empirical relations rather than using the general empirical relations. In addition, the inlet guide vane settings were introduced in the compressor model, as well as, the slip factor.

It was determined that the soundest empirical relations were those matching the following model parameters:

- impeller isentropic efficiency as function of inlet guide vane position, and
- dimensionless enthalpy as a function of flow coefficient.

HFC-236ea data were used to generate these two empirical relations. The first empirical relation had a larger variance than the second relation, and also the data variations generated in deriving the empirical relation were proportional to the difference in model output and the measured data.

Since the by-pass compressor operation mode flow rate was successfully correlated to the measured flow rate for the known percentage of the by-pass valve opening, the new model is feasible for this operating mode.

A systematic error occurred in the model estimate for the CFC-114 data points; hence, the model is a function of the refrigerant type.

The blockage factor was optimized to improve the physical validity of the new model, and the optimized value has more physical relevance than the value determined for the existing model.

Quantitative Comparison Between Two Models

From the comparison results presented in Table 1 it can be inferred that the new model estimates the compressor performance parameters better than the existing model. In addition to the increased physical validity of the new model, this improvement in the parameter estimation definitely classifies the new model as the better.

The new model is dependent on the refrigerant type. The model represented here is developed for HFC-236ea data; hence, a different set of empirical relations should be developed for a different refrigerant operating in the same compressor. Not enough CFC-114 data points were available in this study to develop a CFC-114 model.

The new HFC-236ea model can estimate the flow rate within ±20% relative difference between measured and modeled flow rates, exit state temperature within ±1.7°C, and the compressor shaft power within ±14% relative difference.

In addition to being a quantitatively superior model, the new model also has more physical validity than the existing model for the following reasons:

- The blockage factor in the new model was derived to be 0.9, while the existing model blockage factor was optimized to the value of 0.24. The existing model blockage factor must have accounted for some other compressor performance effects, diminishing the model's physical relevance.
- The existing model required the polytropic exponent and the reference polytropic efficiency to be estimated as input.
- The existing model inlet guide vane setting is introduced in the model through the flow rate correction formula, while in the new model the inlet guide vane angle was a fundamental factor upon which the model was developed.

Table 1. Quantitative Comparison Between Two Compressor Models

Comparison Between Measured and Modeled Compressor Parameters	HFC-236ea Data Used to Build Empirical Relations		HFC-236ea Additional Data		CFC-114 Data	
	Average *	Max. †	Average	Max.	Average	Max.
The Existing Model						
Flow rate w/o IGV cor. [%] ‡	13.96	36.81	22.27	49.71	30.67	76.25
Flow rate IGV corrected [%] ‡	6.33	21.85	7.66	15.57	19.22	31.01
Exit state temperature [°F] §	1.22	3.43	3.49	9.07	13.97	47.31
Shaft power [%] ‡	15.67	21.04	20.27	39.40	33.04	52.99
The New Model						
Flow rate prediction [%] ‡	4.45	19.51	12.57	19.84	22.84	55.08
Exit state temperature [°F] §	0.89	2.21	2.13	6.11	6.21	17.81
Shaft power [%] ‡	3.13	13.38	9.26	17.22	20.54	42.61

* The mean value for a set of absolute differences between measured and modeled particular compressor parameters for the data set in question.
† The maximum absolute difference between measured and modeled particular compressor parameter for the data set in question.
‡ The comparison results are presented in terms of the absolute relative difference between measured and modeled particular compressor parameter, X, given as percentage; Difference [%]= $\left|(X_{meas} - X_{mod})/X_{meas}\right|$ 100. (IGV=inlet guide vane.)
§ The comparison between measured and modeled compressor exit state temperatures is given in terms of absolute difference between measured and modeled temperatures in Fahrenheit degrees for the particular data set; Difference [°F]= $\left|t_{3\,meas} - t_{3\,mod}\right|$ • (1°C = 5°F/9.)

Refined Model

A more detailed centrifugal compressor model was built based on the new model. This refined model needed two empirical relations, and the best relations appeared to correlate the same pairs of parameters as in the new model. The refined model contains the variable slip factor and a more detailed diffuser model, which are the improvements to the new model.

The set of equations of the refined model never yielded a reasonable solution. Errors generated in the model output solution were found to be functions of the sequence in which the equations were solved. However, the source of such spurious behavior of the system of equations was never identified. The authors still believe that the system might be solvable with good equation-solver software.

Design Analysis

The compressor design parameters included in the new model were investigated for the constant compressor operating input. These design parameters were the number of impeller blades, the blade inlet and exit tip angles, the impeller diameters, and the impeller exit axial width. The compressor performance was characterized with the compressor work coefficient which is proportional to the amount of energy transferred in the impeller and the magnitude of the pressure at the impeller exit. The other vital parameter in the compressor operation was the refrigerant flow rate.

The effects of the design parameters on the compressor model results include:

- The number of impeller blades is directly related to the slip factor. Impeller performance as a function of the slip factor has a positive gradient, while the flow rate as a function of the slip factor is negatively sloped. These trends were observed in the design analysis, and it appears that the number of blades was chosen appropriately. Increasing the number of blades by about 10% might improve compressor performance, but would result in a pressure drop of approximately 15%.
- The blade tip angle is inversely related to the slip factor, but still directly related to the compressor performance. The trends observed in the blade tip angle design analysis were very similar to the trends encountered in the number of impeller blades design analysis. The blade tip angle appears to be well chosen. Enlarging tip angle by about 5% might improve the work coefficient by roughly 10% and reduce the flow rate by about 20%, with an increase of 6.9-13.8 kPa in impeller exit pressure.
- Increasing the inlet impeller diameter improves compressor performance and reduces the mass flow rate. The flow rate variations are more significant than the variation in the work coefficient, which is the primary quantitative impeller performance measure. The pressure at the impeller exit and the compressor exit enthalpy are not affected by this parameter. The influence of inlet diameter on compressor performance rapidly diminishes as the flow is throttled with guide vanes. For these reasons, the inlet diameter should be excluded from the eventual impeller design modification.
- The impeller exit diameter greatly affects the magnitude of the flow rate, indicating a large gradient as the flow rate is varied with the size of the impeller. The compressor performance (work coefficient) is a negatively sloped function of the exit impeller diameter. The exit diameter changes should be very limited, since a 10% increase in the impeller exit diameter roughly doubles the mass flow rate.
- The impeller discharge area is directly related to the impeller axial width, so it affects the flow rate considerably. Impeller performance as a function of the impeller axial width has a small negative gradient, while no significant change is observed in the impeller exit state pressure. The impeller exit axial width may be altered to change flow rate if necessary for a relatively small change in compressor performance.

Although the compressor model indicated reasonable physical behavior, the design analysis should be taken very cautiously. The intent of the design analysis was to indicate trends in the compressor performance with the design input variations rather then to suggest specific changes in the compressor design. Since the compressor model was found to be sensitive to the refrigerant type, it is reasonable to assume that the model is sensitive to the compressor design modifications. Also, the assumption of constant operating input with design input variations should be regarded with some reservations because of the indicated sensitivity of the compressor model.

Recommendations

This study points to several recommendations:

1. The refrigerant flow rate was not measured on the chiller installation, but rather estimated from the condenser and the evaporator energy balances. Since the flow rate was found to be an extremely sensitive value in the compressor models, the flow rate should be verified on the experimental installation. As the vital value in any refrigeration compressor modeling procedure, the flow rate should be measured by at least several flowmeters on a single experimental refrigeration installation. At the very least, the flow rates through the chiller and the compressor by-pass loop should be measured.
2. The total compressor power consumption should always be measured and reported. The total compressor power consumption is an important parameter to include in the modeling in order to be able to predict compressor energy requirements for different compressor operating conditions.
3. The compressor in the Navy chiller has a vaneless diffuser. However, vaned diffusers are also widely used in centrifugal compressors. Their advantage over vaneless diffusers is in broadening the stable compressor operating range and reducing the fluid expansion losses at the diffuser inlet. However, this type of diffuser has greater friction losses than the vaneless diffuser. The use of a vaned diffuser should be considered.
4. Both models were dependent on the refrigerant type. The models presented here were developed based on HFC-236ea data, hence a different set of empirical relations should be developed for a different refrigerant operating in the same compressor. This should give insight into the dependence of the model on the refrigerant type.
5. A better compressor performance map than the one provided by the compressor manufacturer should be developed, which may require a wider range of the operating conditions, especially compressor Mach numbers. Such a performance map might be a better tool with which to model compressor performance.

6. Further attempts should be made to solve the refined model. It is anticipated that the refined model, if a solution to its set of the equations were found, would improve the accuracy of the new model.
7. Increasing the number of blades by about 10% might improve compressor performance, but would result in an increased pressure drop of about 15%.
8. Enlarging the tip angle by about 5% might improve the work coefficient by roughly 10% and reduce the flow rate by about 20%, with an increase of 6.9-13.8 kPa in the impeller exit pressure.
9. The inlet diameter should be excluded from the eventual impeller design modification.
10. One has to be very careful modifying the impeller diameter, since it strongly influences compressor performance.
11. The compressor exit axial width may be enlarged to increase flow rate, if necessary, for a relatively small deterioration in the compressor performance.

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The complete report, entitled "A Modeling and Design Study Using HFC-236ea as an Alternative Refrigerant in a Centrifugal Compressor," (Order No. PB97-156129; Cost: $35.00, subject to change) will be available only from:
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