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THE APPLICABILITY OF CONSTANT PROPERTY ANALYSES IN CRYOGENIC HELIUM HEAT EXCHANGERS

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ABSTRACT

Cryogenic helium heat exchangers are commonly modeled with simplified techniques that assume constant fluid properties throughout the exchanger. The large variations in helium fluid properties over the temperature range experienced in cryogenic exchangers raises the question of how applicable these constant property exchanger analyses are. Two constant property techniques, the effectiveness—NTU method and the log mean temperature difference method, are compared to a finite difference heat transfer model. Exchanger model accuracy is judged by determining its influence on the performance of different refrigeration cycles. Failure to account for fluid property variations in cryogenic helium exchangers can lead to noticeable error in cycle performance predictions. The accuracy of the constant property methods can be increased by breaking the exchanger into regions where properties can indeed be treated as constant. This technique, however, works better for the log mean temperature difference method than for the effectiveness—NTU method.

INTRODUCTION

The performance of cryogenic helium refrigeration cycles highly depends on how effectively heat exchangers cool the high pressure inlet helium stream with low pressure return vapor. Accurate heat exchanger models are required when analyzing such systems. This paper is based on work done to model the heat exchangers in Fermilab's Tevatron Satellite Refrigerators, which provide liquid helium to cool superconducting magnets in the Tevatron particle accelerator.

The large helium property variations that occur in the cryogenic temperature range raises the concern of whether or not commonly used constant property heat exchanger analyses are applicable. Some earlier work on this topic[1] indicates that the effectiveness—NTU method using mean property conditions may give results that reasonably approximate the true exchanger effectiveness obtained from a variable property, finite difference solution. This paper will investigate applying two constant property methods, the effectiveness—NTU and log mean temperature difference methods, to an unbalanced turbulent flow shell and tube exchanger. Temperature dependent fluid properties will be accounted for with a finite difference analysis, as well as with multiple subdivision refinements to the constant property techniques. Furthermore, exchanger model accuracy will be based not only on the exchanger effectiveness, but also, and most importantly, on the effectiveness' influence on a refrigeration cycle.

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METHODS FOR SOLVING HEAT EXCHANGER PERFORMANCE

In general, at any point within a heat exchanger one can calculate a local convection heat transfer rate between two fluid streams:

$$Q = UA\left(T_{fluid1} - T_{fluid2}\right) \tag{1}$$

Where U is the overall heat transfer coefficient that is determined by the local heat transfer coefficients of both fluid streams, the total surface area of both fluid sides, the fin efficiency, and the conduction resistance of the wall material. The exchanger is a shell and tube configuration with the high pressure refrigerator inlet flow on the tube side and the low pressure return flow on the shell side. The inlet and outlet temperatures, pressures, and mass flow rates will be designated for both fluid sides.

The finite difference method (FD) breaks an exchanger into differential area elements and directly computes heat transfer based on the temperature difference between elements. Total exchanger heat transfer is found by numerical integration of the differential heat transfer values.

The log mean temperature difference method (LMTD) is based on an analytical integration of the finite difference model rate equation. Basing the integration on a counter flow exchanger:

$$Q = UA\Delta T_{lm,cf} \tag{2}$$

where $\Delta T_{\ell m,cf}$ is the log mean temperature difference for counter flow. The log mean temperature difference method assumes that U is constant throughout the exchanger. Thus, variations in specific heat, thermal conductivity, and viscosity which affect the heat transfer coefficients are not accounted for.

The effectiveness–NTU method (ε–NTU) defines a heat transfer effectiveness:

$$\varepsilon = Q / Q_{max} \tag{3}$$

Where Q_{max} is the maximum possible heat transfer as limited by the second law, which establishes the maximum allowable temperature change for a flow stream. Q_{max} is the minimum of:

$$Q_{max} = \dot{m}_{tu}(h(P_{tu,in}, T_{tu,in}) - h(P_{tu,out}, T_{sh,in})) \tag{4A}$$

$$Q_{max} = \dot{m}_{sh}(h(P_{sh,in}, T_{tu,in}) - h(P_{sh,out}, T_{sh,in})) \tag{4B}$$

Effectiveness is a function of the "number of transfer units" (NTU) and the "heat capacity rate ratio" (C_r) where $NTU = UA/C_{min}$ and $C_r = C_{min}/C_{max} = (\dot{m}c_p)_{min}/(\dot{m}c_p)_{max}$. When computing NTU, U is assumed constant throughout the exchanger. Effectiveness formulas are available from Kays and London[2] for various heat exchanger configurations. For a counter flow exchanger:

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C_r)}{1 - C_r \exp[-NTU(1 - C_r)]} \tag{5}$$

Note that as C_r approaches 1, use $\varepsilon = NTU/(NTU+1)$.

Several similarities exist between the LMTD and ε–NTU methods. Both give theoretically equivalent results because both provide an integration of the elemental area convective heat transfer rate equation for a given heat transfer configuration as the temperature profile changes along the length of the exchanger. Also, both methods rely on assuming a constant U throughout the length of the exchanger. These methods are, however, capable of handling variations in U (due to property variations) by breaking an exchanger into multiple smaller exchangers. Then, in these subdivided areas of the entire exchanger, the assumption of a constant U is more valid, allowing these analysis methods to be accurately used.

SYSTEM DESCRIPTION

Fermilab's Tevatron Satellite Refrigerators have 35 ft. long heat exchanger columns that are used in providing liquid helium to superconducting magnets in the Tevatron particle accelerator. Four exchangers, consisting of helically wound finned tubing encased in a shell, make up the exchanger column. In this report, the exchanger at the cold end of the column will be modeled to study the effects of various exchanger solution techniques. Specific geometric details for the cold end exchanger are given in Table 1. The exchangers normally operate with approximately 8% more shell side return flow than tube inlet flow since the Tevatron refrigerators are usually boosted with additional refrigeration by taking liquid helium from a central liquefier.

The problem to be solved is the prediction of heat exchanger performance for refrigeration cycle simulation. Assume that inlet temperatures, pressures and flow rates are known. The most versatile approach, that allows the use of any of the three heat exchanger solution methods, is to guess the tube outlet temperature. Then a heat exchanger solution technique can be used to predict the warm end temperatures, iterating until the predicted warm tube inlet temperature agrees with the known value.

CALCULATION DETAILS AND ASSUMPTIONS

The exchanger will be considered a counter flow exchanger. A tube and shell exchanger with many passes will approach counter flow performance. Assume that pressures are constant throughout the heat exchanger. Ignore pressure drop effects. Take the stream pressure to everywhere equal the cold end pressure since its in the cryogenic temperature range where properties show the most sensitivity to pressure.

The shell side fin effectiveness is estimated to be 80%. Since fin surface area accounts for almost all of the total shell area, we can therefore approximate the overall shell surface effectiveness be 80%.

Table 1. Heat Exchanger Geometric Details

Helix Dia.	9.625	Fin Dia.	1.0 in.
No. of Tube Passes	3	Fin Thickness	0.01 in
Tube Length/Pass	40 ft.	Fin Frequency	20 fins/in
Tubing Outer Dia.	0.50 in		
Tubing Inner Dia.	0.43 in.		

The convective heat transfer coefficient for turbulent flow inside a helically wound tube as given by Colburn[3], is:

$$h_{tu} = 0.023 \frac{k}{D_{tu,in}} \mathrm{Re}_{tu}^{0.8} \mathrm{Pr}_{tu}^{1/3} \left[1 + 3.5 \frac{D_{tu,in}}{D_{helix}} \right] \qquad \text{(6A)}$$

For the shell side, use a correlation for the flow normal to a bank of staggered finned tubes[4]:

$$h_{sh} = 0.45 \frac{k}{D_{tu,out}} \mathrm{Re}_{d}^{0.625} \left(\frac{A_t}{A_o} \right)^{-0.375} \mathrm{Pr}^{1/3} \qquad \text{(6B)}$$

where A_t is the complete outer heat transfer surface area including fins and A_o is the surface area of a finless tube of the same diameter and length.

Calculations will be performed using double precision. This helps iteration convergence and finite difference computations. Helium properties are evaluated from work by Arp and McCarty[5]. The heat transfer coefficients are based on properties at the mean fluid temperature, the average between the inlet and outlet temperatures. Average specific heats are obtained from the stream's inlet and outlet conditions by dividing the change in enthalpy by the change in temperature.

Heat leak into the exchanger and axial conduction effects will be ignored. Resistance to conduction in the walls will be neglected.

APPROACH TO ANALYZING EXCHANGER PERFORMANCE MODELS

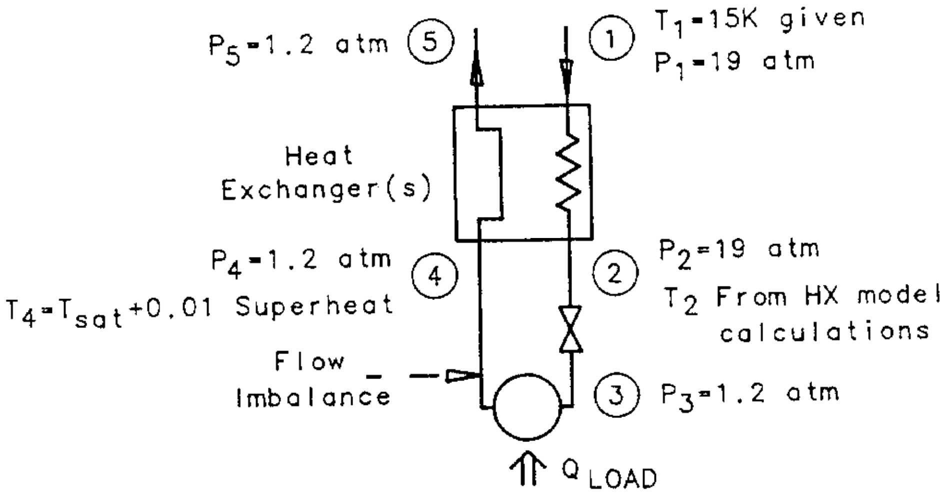
The heat exchanger will be modeled with the LMTD method, the ε–NTU method, and the FD method. The goal is to find a model that accurately handles the fluid property variations in the cryogenic exchanger while minimizing computations.

The FD method will provide a rigorous solution for heat transfer that accounts for property variations along the length of the heat exchanger. However, this method requires many computations to obtain an accurate elemental area approximation.

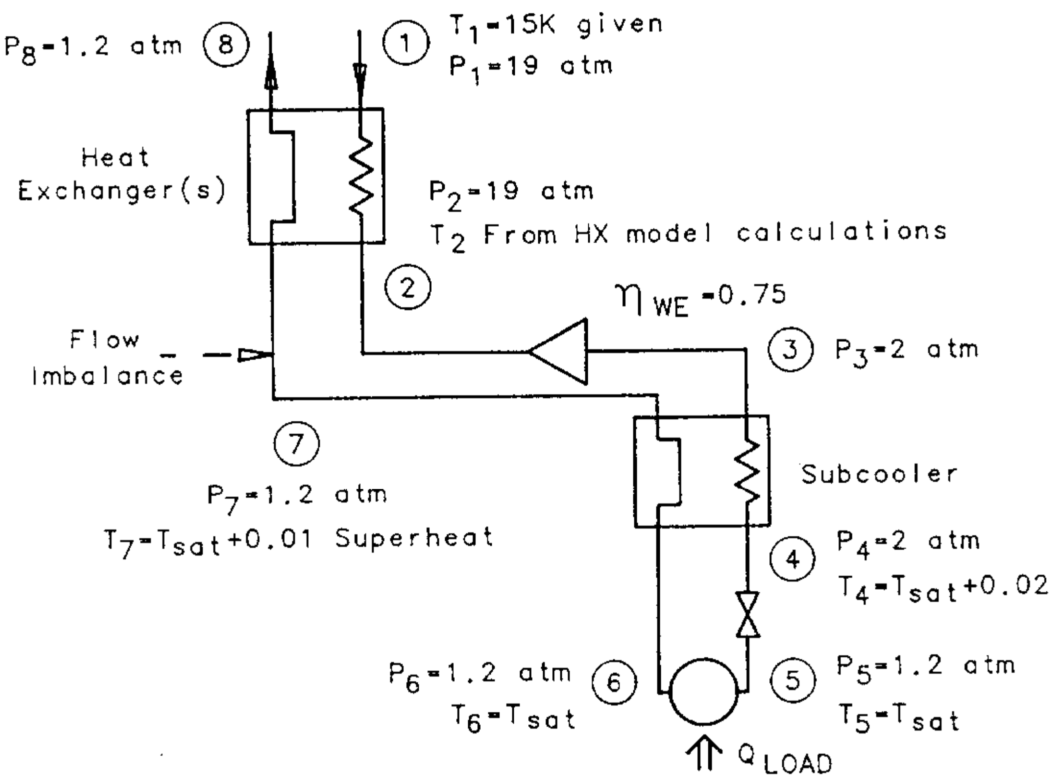
The LMTD method and the ε–NTU method require fewer calculations for a heat exchanger solution since they provide an analytical integration of elemental area steps. However, these methods rely on approximating fluids as having constant properties throughout the entire exchanger. Property variations can be handled with these methods by subdividing an exchanger into several parts. Then, the LMTD and ε–NTU methods can be used in these regions where a more valid constant property assumption can be made. Advancing through these multiple exchanger steps will give the results for the entire exchanger.

Initially heat exchanger calculation methods will be compared using computed heat transfer rates. Since inlet conditions are known, this comparison can be made by looking at calculated exchanger "effectiveness". Assume that the effectiveness predicted from the FD model is the most correct value.

However, merely comparing effectiveness and heat transfer predictions is not sufficient to completely assess an exchanger model. A better means of comparison will be to determine the exchanger effectiveness influence on a refrigeration cycle. This enables exchanger models to be compared by determining their influence on physical parameters that are of most concern such as refrigeration load or liquid yield. Very small changes in heat exchanger effectiveness can have a significant impact on cryogenic cycle performance, particularly in the liquid helium temperature range[6].



(a)



(b)

Fig. 1. Cycles for exchanger performance assessment
(a) Refrigerator with Joule-Thomson valve,
(b) Refrigerator with expander and subcooler.

The heat exchanger models will be integrated into two refrigeration cycles. The first cycle will employ a simple Joule–Thomson expansion valve (Figure 1a), while the second, more sophisticated cycle will use an expansion engine and a subcooler (Figure 1b). For the established exchanger inlet condition, each model will predict a refrigeration load. The predicted load will be compared to the load calculated when the cycle uses a FD heat exchanger model, which is assumed to give the most accurate cycle performance.

RESULTS

The shell inlet is taken to be saturated vapor at 1.2 atm with 0.01K of superheat (4.432K). The exchanger was assumed to have a high pressure tube inlet temperature of 15K. The high pressure inlet flow to the exchanger is taken to be 50 g/s. Flow imbalance cases of 4%, 8% and 12% will be studied.

Figure 2 shows results for various exchanger models. First, the inadequacy of simple handling the individual exchanger with one step of either the LMTD or ε–NTU method is demonstrated in this figure. When the exchanger is analyzed in this manner, overall effectiveness predictions can be close (within about 12% for ε–NTU, and 4% for LMTD) to the "true" effectiveness value as determined by using a finite difference solution. Yet these apparently small differences in effectiveness lead to a noticable difference in predicted cycle performance. When using a single step, constant property model in a refrigeration cycle with a Joule–Thomson valve, heat load predictions were off around 30% to nearly 100%. Heat load predictions were off around 10% to 35% for expander and subcooler refrigeration cycles. Therefore, to model the heat exchanger column such that refrigeration loads and temperature profiles profiles are predicted with accuracy, the exchanger must be broken into more than one step.

For this exchanger, the ε–NTU method was never able to satisfactorily approach the FD model prediction. This is due to the fact that specific heat in this region is changing too rapidly to obtain an accurate Q_{max} value. Even though the logic used what was intended to be the most realistic Q_{max} based directly on enthalpy change as shown in Eq. (4A) and (4B), the actual derivation of effectiveness relies on applying the minimum heat capacity $C_{min}=(\dot{m}c_p)_{min}$ over the entire maximum temperature range to compute Q_{max}. That is, the ε–NTU method derivation assumes that the minimum capacity side average constant c_p for a given node is valid over the entire maximum temperature range of:

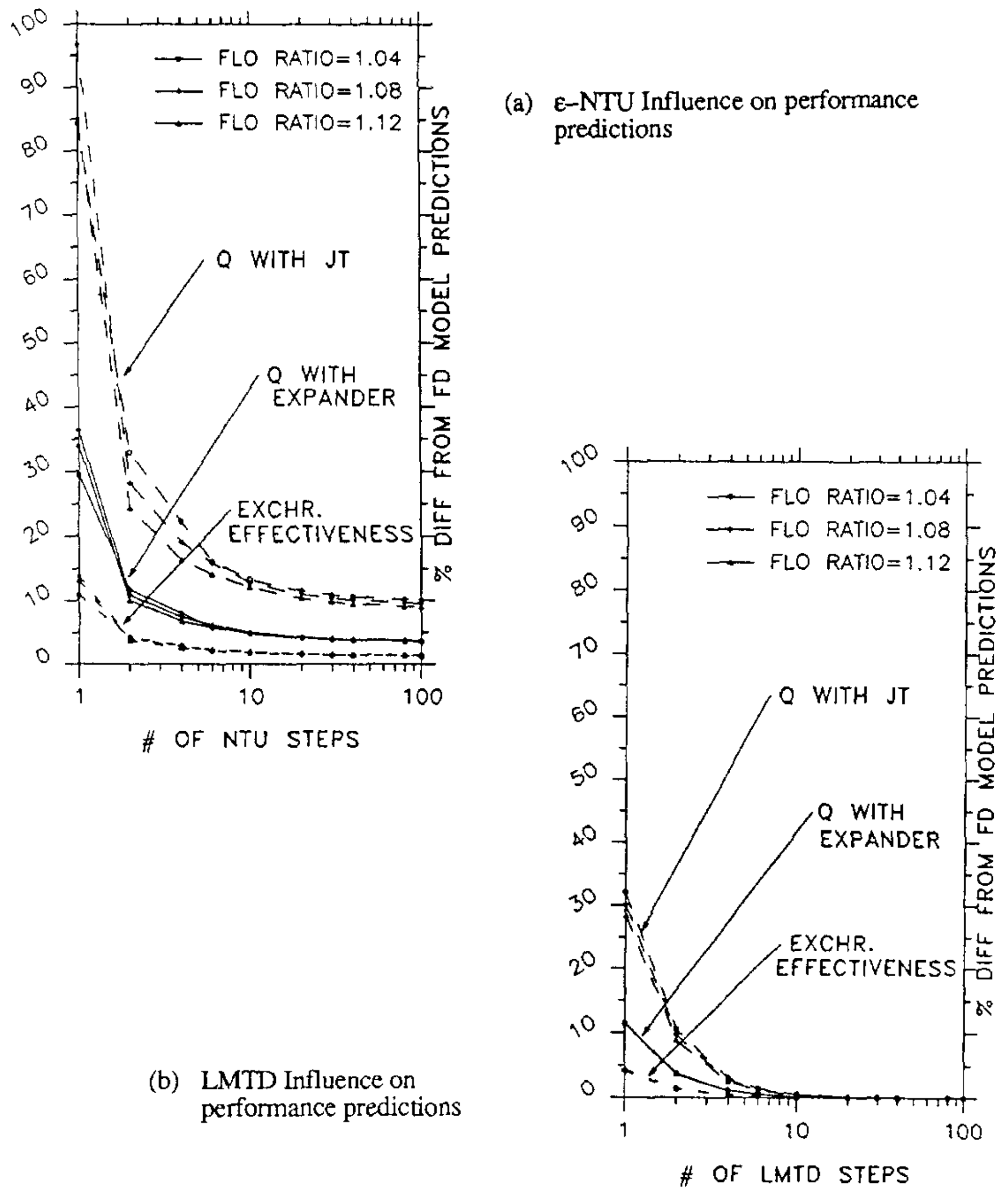
$$\Delta T_{max} = T_{tu,in} - T_{sh,in} \tag{7}$$

This c_p and ΔT_{max} are used to determine the Q_{max} upon which ε is defined in Eq. (3). However, at very low temperatures for helium, c_p changes so rapidly that the constant c_p assumption is not valid over the range of the inlet temperatures, no matter how small the step length. Thus, it is impossible to determine a realistic Q_{max} for this method. The LMTD method is indeed able to approach the FD solution. It has the advantage of only requiring that the constant property assumptions apply over each stream's step inlet and outlet temperatures and not over the entire maximum temperature range of the inlet temperatures. Figure 2b shows that after breaking the exchanger into about six LMTD steps, the FD solution is approached well enough to reduce the error in predicted refrigeration loads to less than 1%.

CONCLUSIONS

Failure to consider fluid properly variations when analyzing cryogenic helium exchangers can lead to noticeable error in cycle performance predictions. The effect of property variations is significantly felt in the exchanger region below about 15K. Cycles that rely solely on the Joule–Thomson expansion process are particularly sensitive to constant property exchanger solution errors.

Traditional constant property analysis methods can be applied to the variable property helium exchanger problem by subdividing exchangers into regions were properties can indeed to treated as being constant. In this manner, the large number of steps required for a finite difference solution, which would most precisely account for property variations, can be avoided. The log mean temperature difference method is the most robust technique when subdividing an exchanger in that it is always able to produce a solution that approaches the finite difference solution. The effectiveness—NTU method will not work in regions approaching liquid helium temperatures because of the very rapid specific heat changes. Therefore, the ε–NTU method is not recommended when subdividing exchangers



(a) ε-NTU Influence on performance predictions

(b) LMTD Influence on performance predictions

Fig. 2. Exchanger modeling results

with temperatures less that 15K. This method, however, is valid for temperatures above this range.

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