A GENERAL COMPUTER MODEL FOR PREDICTING THE PERFORMANCE OF GAS SORPTION REFRIGERATORS*

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ABSTRACT

Projected performance requirements for cryogenic spacecraft sensor cooling systems demand higher reliability and longer lifetimes than the state-of-the-art provides. The gas/solid sorption refrigerator is viewed as a potential solution to these cryogenic cooling needs. A generalized analytical software model of an entire gas sorption refrigerator system has been developed. The numerical model, generated from a systems point of view, is flexible enough to evaluate almost any combination and order of refrigerator components and any sorbent-sorbate pair for which the sorption isotherm data are available. Parametric curves for predicting system performance were generated for two types of refrigerators, a $LaNi_5$-H_2 absorption cooler and a Charcoal-N_2 adsorption cooler. It was found that precooling temperature and heat exchanger effectiveness affect the refrigerator performance significantly. Examination of the results indicates that gas sorption refrigerators are feasible for a number of space applications.

INTRODUCTION

Projected performance requirements for cryogenic sensor cooling systems demand higher reliability and longer lifetimes than the state-of-the-art can provide. A Joule-Thomson cryostat driven by a gas/solid sorption compressor, the "gas sorption refrigerator", is a potential solution. In support of the sorption refrigerator research program at JPL, a software analytical model of a general gas sorption refrigerator system has been developed. The numerical model was used to examine the relationships between heat exchanger effectiveness, cycle times, precooling temperature, total cooling power, system mass, and input power requirements. Two cases, a 30 K $LaNi_5$-H_2 absorption cooler and a 91 K Charcoal-N_2 adsorption cooler, with maximum compressor pressures of 40 and 60 bar, respectively, are presented here.

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NOMENCLATURE

c	gas to sorbent concentration ratio
COP	coefficient of performance
C_p	specific heat
C_{p_c}	specific heat, compressor case
C_{p_m}	specific heat, metal foam
C_{p_s}	specific heat, sorbent
d	inside diameter of compressor
d_{JT}	J-T valve diameter
f_o	sorbent packing factor
f_{micro}	volume fraction of solid sorbent per particle
f_{solid}	volume fraction of micropores per particle
h	heat transfer coefficient
h_{co}	enthalpy, heat exchanger cold side outlet
h_{hi}	gas enthalpy, heat exchanger hot side inlet
H^o	isosteric heat of adsorption/absorption
k	thermal conductivity
K	ratio of gas specific heats
L	length
$\dot{m}$	gas mass flow rate
m	mass
m_c	mass, compressor case
m_g	mass, gas
m_m	mass, metal foam
m_s	mass, sorbent
M	gas molecular weight
NTU	number of transfer units, heat exchanger
P	maximum pressure in compressor
P_{ho}	pressure, heat exchanger hot side outlet
Pr	Prandtl Number
Q_{ij}	heat input from point i to j in compressor
$\dot{Q}_L$	cooling power
$\dot{Q}_{heat}$	time averaged input power, total compressor
R	universal gas constant
Re	Reynolds Number
t_{ij}	time from point i to j in compressor cycle
Δt_{sorp}	desorption time in compressor cycle
T_{ci}	temperature, heat exchanger cold side inlet
T_{co}	temperature, heat exchanger cold side outlet
T_{hi}	temperature, heat exchanger hot side inlet
V	volume
x	volume percent of metal foam in compressor
η	heat exchanger effectiveness
ρ_g	mass density of gas
ρ_s	mass density of sorbent
σ_y	maximum yield strength of compressor metal

GAS SORPTION REFRIGERATOR SYSTEMS

The basic components of a gas sorption refrigerator are the gas sorption compressor, a precooling radiator, a counterflow heat exchanger, and a J-T expansion valve. A typical block diagram is shown in Figure 1. The gas sorption compressor is a non-mechanical compressor which utilizes the phenomenon of gas absorption or adsorption to pressurize the gas. The sorbent material in the compressor absorbs/adsorbs the gas in large quantities when cooled at low pressure and desorbs the gas at higher temperatures and pressures when heated. The flow through the compressor is controlled with self-operating check valves. The high pressure gas from the compressor is precooled below the inversion temperature upon passing through the radiator and is further cooled in the counter-flow heat exchanger in the J-T cryostat before being isenthalpically expanded through the J-T valve. The heat load is absorbed upon evaporation of the condensate.

NUMERICAL MODEL APPROACH

A previous numerical model (ref. 1) evaluated one refrigerator system design with a $LaNi_5$-H_2 absorption compressor. The current, modified version has the capability to evaluate a wide range of combinations and orders of refrigerator components and any sorbent-sorbate pair, for which the data are available, in the compressor. Generated from a systems point of view, the model is a steady-state program that evaluates the characteristics of each refrigerator component from the state properties of the working fluid at the corresponding nodes. The program evaluates the component masses and pressure drops; the required fluid mass flow rate; the required J-T valve diameter; and the gas to sorbent concentration ratios, temperatures, and pressures in the compressor based upon the input refrigerator performance requirements and constraints. The program starts by evaluating the J-T cryostat and then proceeds to evaluate the remaining components in an order opposite to the gas flow. The $LaNi_5$-H_2 cooler schematic, Figure 1, illustrates the node numbering scheme and component order. The Charcoal-N_2 cooler is similar but with only one radiator and no intermediate heat exchanger. All the temperatures and pressures throughout the system are found and any required gas properties such as enthalpy and thermal conductivity are evaluated through a gas properties look-up code (ref. 2). Currently the capability of the model is restricted to one fluid loop with one J-T cryostat and only data for Charcoal-N_2 adsorption (ref. 3) and $LaNi_5$-H_2 absorption (ref. 4) have been entered into the program.

J-T CRYOSTAT

The J-T cryostat is a combination of a counterflow heat exchanger, a J-T expansion valve, and an evaporator. The required fluid mass flow rate through the system is found from the given cooling heat load and the change in enthalpy through the J-T cryostat. The change in enthalpy is a function of the precooling temperature, T_{hi}, and the heat exchanger effectiveness, η. The heat exchanger cold side outlet temperature is found from the precooling temperature, the cooling temperature, and the effectiveness

$$T_{co} = (T_{hi} - T_{ci})\eta + T_{ci} \tag{1}$$

$$\dot{m} = \dot{Q}_L / (h_{co} - h_{hi}) \tag{2}$$

The maximum JT valve diameter is found as a function of the mass flow rate and the gas properties, assuming choked isentropic expansion:

$$d_{JT} = 2\left[\frac{\dot{m}}{\pi P_{ho}}\right]^{.5}\left[\frac{T_{ho}R}{MK}\left(\frac{K+1}{2}\right)^{\left(\frac{K+1}{K-1}\right)}\right]^{.25} \tag{3}$$

The mass of the J-T cryostat is assumed to be the mass of the counter-flow heat exchanger because the mass of the J-T valve is negligible in comparison. The masses of the refrigerator components tend to increase linearly with fluid mass flow rate, therefore the system mass increases nearly linearly with the cooling load. Consequently, minimizing the fluid mass flow rate will tend to optimize the coefficient of performance, COP, and minimize the system mass.

COUNTERFLOW HEAT EXCHANGER

The hot and cold inlet temperatures of the heat exchanger are determined by the adjacent components in the system. The hot and cold outlet temperatures are determined from the heat exchanger effectiveness and an energy balance across the heat exchanger. The required length of the heat exchanger is determined as a function of the number of transfer units, NTU, and the heat transfer coefficient, h,

$$L = \frac{NTU\ \dot{m}\ Cp}{\pi d h} \tag{4}$$

The mass of the exchanger is determined from the passage size, the required wall thickness, and the required length. As the effectiveness increases, the change in enthalpy through the cryostat increases, decreasing the fluid mass flow rate. As the masses of the system components are all dependent upon this mass flow rate, the total system mass is reduced as the heat exchanger effectiveness increases, as shown in Figure 2. Also the COP is improved as the effectiveness is increased due to the reduce mass of the compressor.

SPACE RADIATOR

The compressor heat rejection component and the gas precooling component are both passive space radiators. Assuming a constant heat load, an estimate of the radiator masses was made by based on the equations developed for JPL's advaced radiator (ref. 5,6). The COP is shown as a function of the precooling temperature in Figure 3 for a 30 K $LaNi_5$-H_2 system and a 91 K C-N_2 system. Reducing the precooling temperature increases the change in enthalpy across the JT valve thus reducing the required fluid mass flow rate to heat load ratio and improving the COP. As the mass of the entire system is nearly linearly dependent upon the fluid mass flow rate, the refrigerator system mass also decreases with the precooling temperature, Figure 4.

SORPTION COMPRESSOR

The gas sorption compressor is made up of a set of compressor sub-units cycled sequentially to supply an essentially continuous stream of high pressure gas to the J-T valve. The inlet and outlet pressures and temperatures are determined by the adjacent components in the refrigerator. The compressor cycle consists of a heating phase and a cooling phase. Each phase is assumed to be made up of a constant gas concentration pressure change and an isobaric gas concentration change, thus there are four states in the idealized compressor cycle. Consequently, the pressures, temperatures and concentration ratios can be found at all points in the idealized cycle from the sorption isotherms and the compressor inlet and outlet conditions. The required mass of sorbent is directly related to the fluid mass flow rate, the desorption time, the gas concentration change in the compressor, and the void volume in the lines

$$m_s = (\dot{m}/\Delta c)\Delta t_{sorp} \tag{5}$$

The void volume, or dead volume, increases the required sorbent mass as additional sorbent is required to pressurize this volume as well as the gas that is being passed through the compressor. At higher pressures and temperatures, this effect becomes predominant and the refrigerator efficiency degrades.

The heat transfer within each compressor unit is enhanced with a copper foam. Thus the components of the compressor unit are the sorbent, the metal foam, and the pressure case. The total volume is a function of the sorbent packing factor, the volume percent of metal foam, and the case thickness. The heat input between any two points i and j in the compressor cycle is the sensible heat transfer to the sorbent, metal foam, compressor case, and gas, and the isosteric heat of adsorption/absorption

$$Q_{ij} = (mCp)(T_j - T_i) + m_s(c_j - c_i)H^o + m_g(h_j - h_i)_g \tag{6}$$

$$mCp = Cp_m m_m + Cp_c m_c + Cp_s m_s \tag{7}$$

The coefficient of performance is determined from the calculated total heat rejection requirement and the cooling heat load

$$COP = \dot{Q}_L / \dot{Q}_{heat} \tag{8}$$

The coefficient of performance is a strong function of precooling temperature, as shown in Figure 3. This is expected as the mass of the compressor decreases nearly linearly with fluid mass flow rate which is reduced by lowering the precooling temperature. It was found that by staging the compressor for the Charcoal-N_2 system, illustrated in Figure 5, the COP can be improved somewhat at intermediate to high precooling temperatures, Figure 3. However, at low precooling temperatures a cross-over occurs and the single stage C-N_2 system becomes superior to the two-stage C-N_2 system, Figure 4.

The system masses shown include the estimate masses of the supports, insulation, and the compressor heat addition-rejection component. Using Table 1, the masses of each system without these extras can be deduced.

CONCLUSIONS

A numerical model has been developed and is available to size and evaluate gas sorption refrigeration systems for spacecraft applications. Two refrigerator types, $LaNi_5$-H_2 and Charcoal-N_2, have been studied to demonstrate the versatility of the model. It was found that the heat exchanger effectiveness and the precooling temperature have significant effects on the predicted refrigerator performance. It was also determined that staging the compressor with an intermediate radiator improves the performance of the Charcoal-N_2 adsorption refrigerator at higher precooling temperatures. Further studies need to be completed with different adsorption systems in order to characterize the effect of staging the compressor.

Although the COP's of the systems presented here are somewhat non-competitive with conventional mechanical refrigerators, the non-mechanical aspect of the sorption refrigerator makes the sorption refrigerator more attractive for long-life applications when excess waste heat is available.

Table 1

Component Mass Fraction (in percent) at Constant Precooling Temperature
1 Watt Heat Load

	Precooling Temperature		
Component	80 K	100 K	120 K
$LaNi_5$-H_2 Refrigerator:			
Compressor	3.9	5.7	6.5
Compressor heat add/rej component	41.0	60.2	68.4
High temperature radiator	2.2	3.3	3.8
Heat exchanger	1.0	3.3	5.2
Precooling radiator	42.1	19.5	9.0
JT cryostat	0.1	0.1	0.5
Support, plumbing, insulation, etc.*	10.0	10.0	10.0
	150 K	200 K	250 K
C-N_2 Refrigerator (2-Stage Compressor):			
Compressors	0.2	0.8	2.8
Intermediate radiator	5.8	3.1	0.9
Compressor heat add/rej component	78.5	83.7	86.3
Precooling radiator	6.5	3.3	0.9
JT Cryostat	-	-	-
Support, plumbing, insulation, etc.*	10.0	10.0	10.0

* assumed constant mass fraction of 0.1

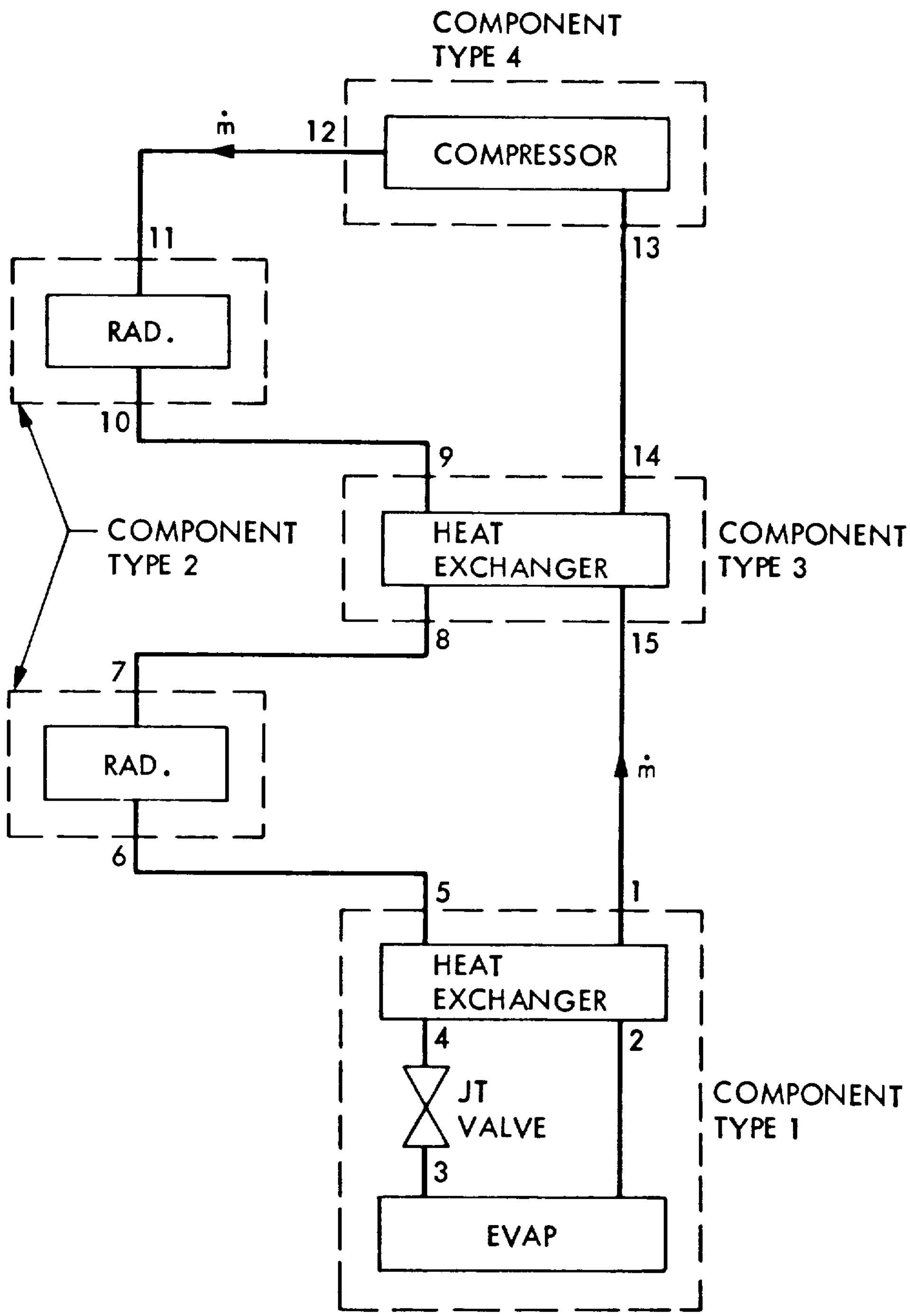

Figure 1. $LaNi_5$-H_2 System Block Diagram

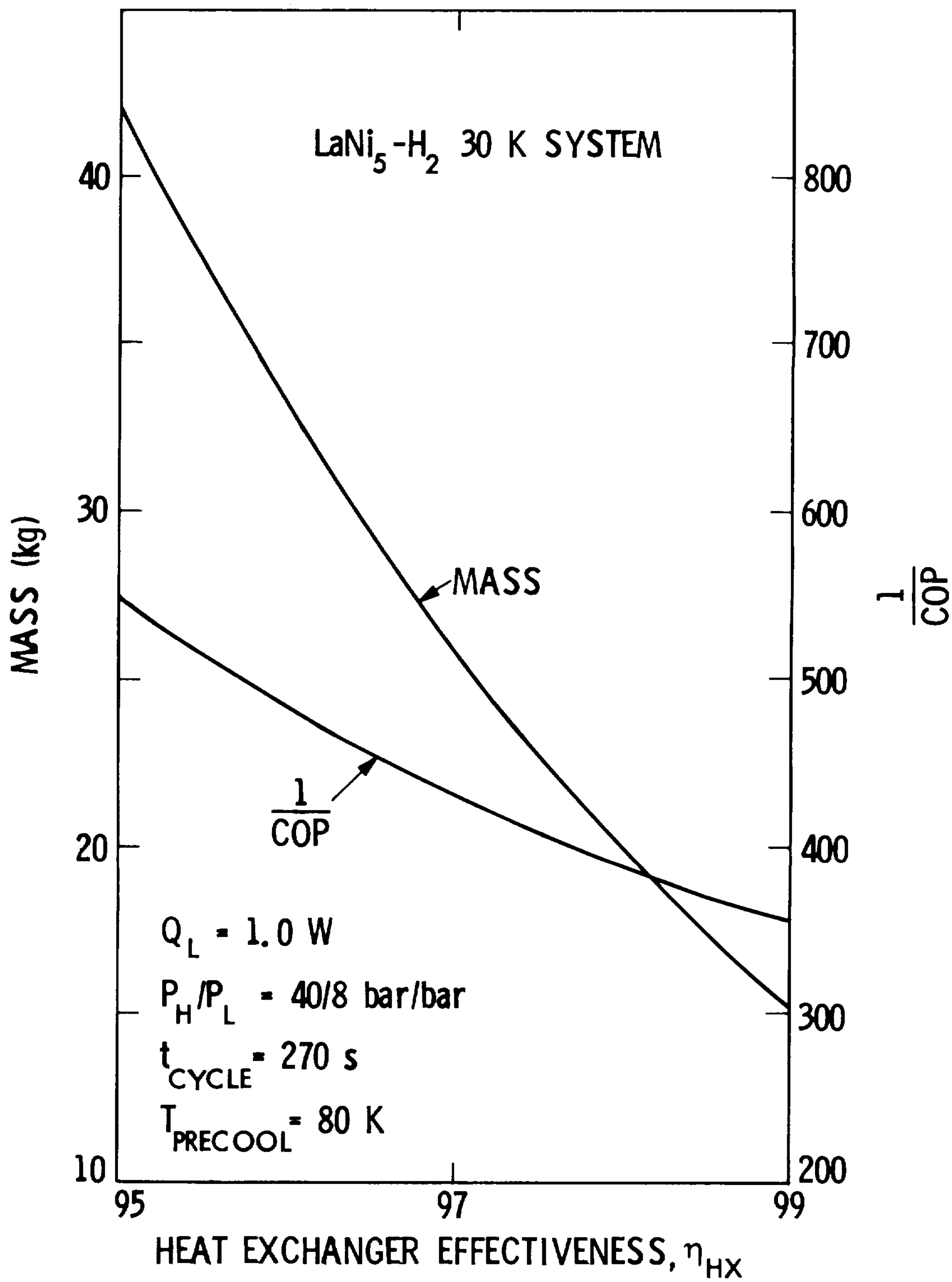

Figure 2. System Mass and COP versus Heat Exchanger Effectiveness

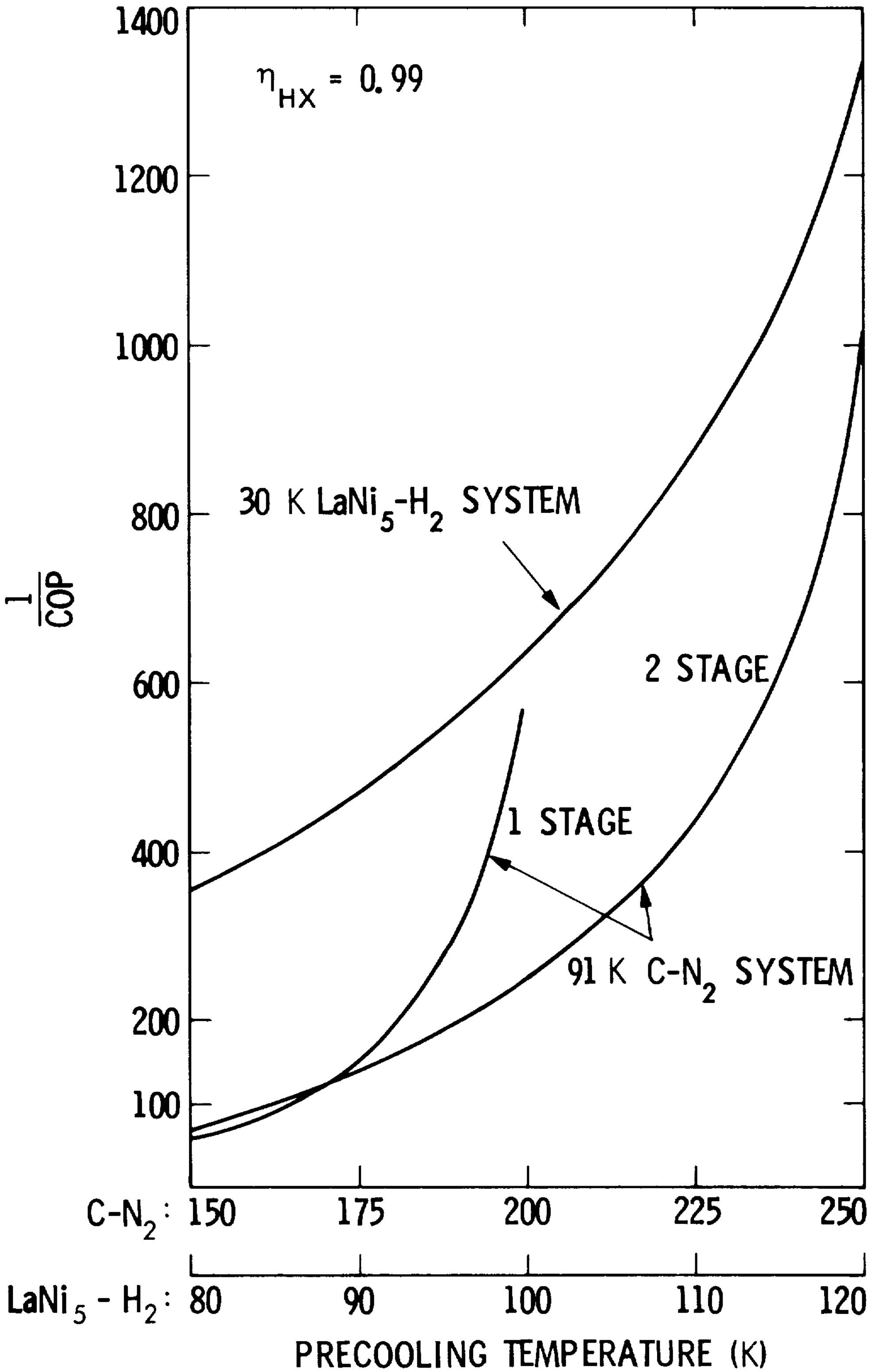

Figure 3. COP versus Precooling Temperature

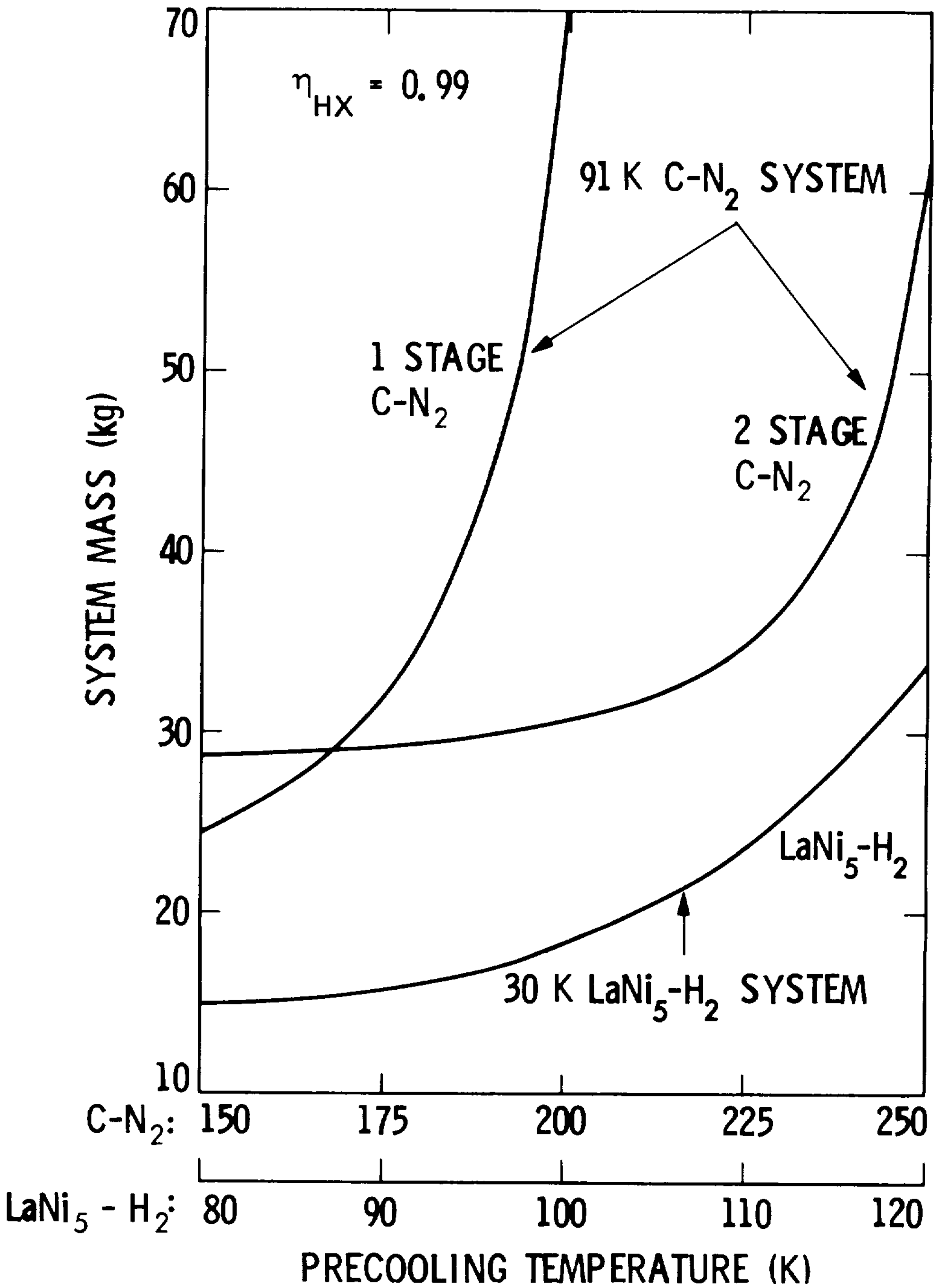

Figure 4. System Mass versus Precooling Temperature

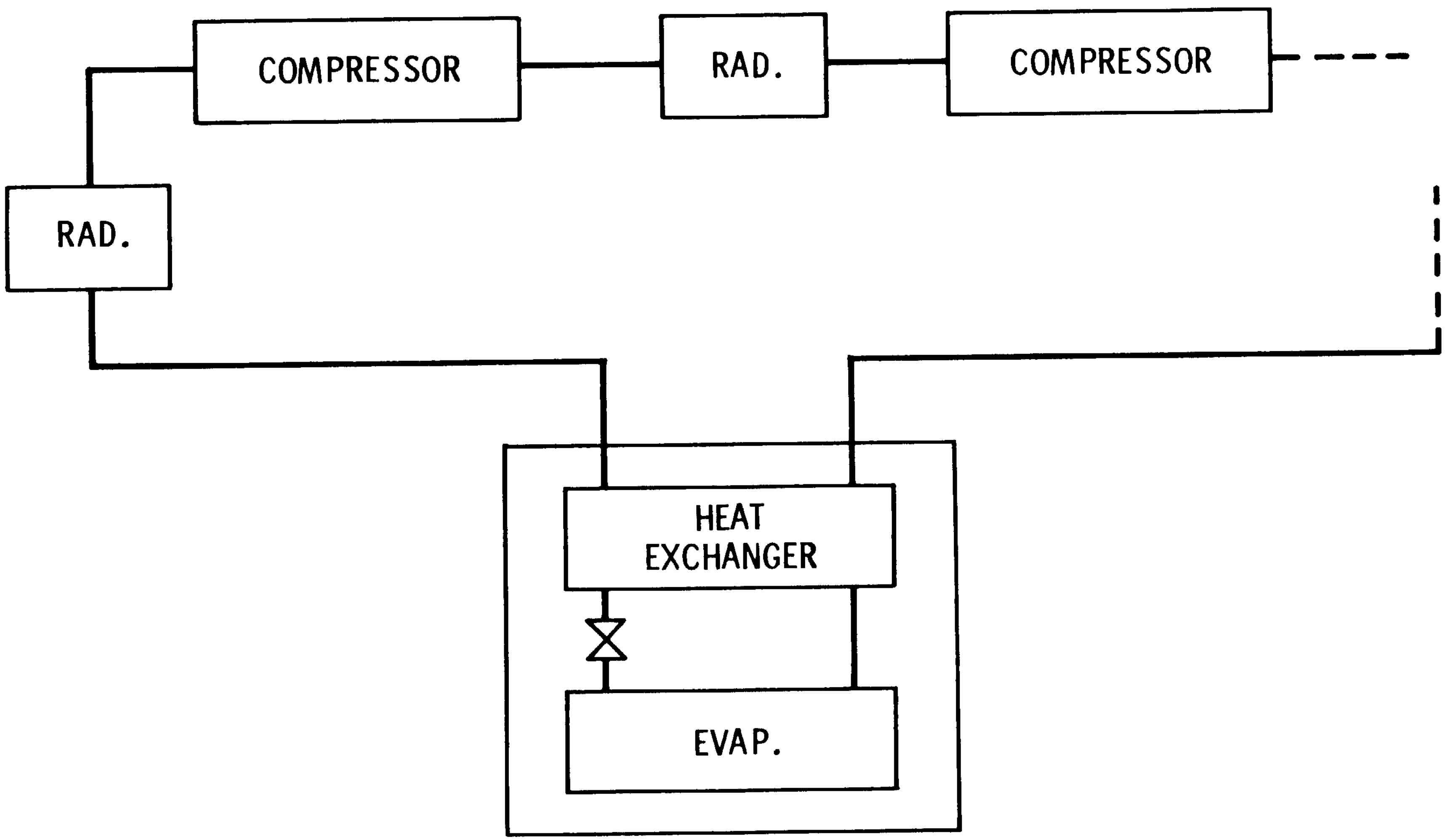

Figure 5. Schematic of Staged Compressors with Intermediate Cooling.

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