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COMPUTER DESIGN AND OPTIMIZATION OF CRYOGENIC REFRIGERATION SYSTEMS

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This paper describes the design and optimization of a cryogenic refrigerator using the LBL computer program GEOTHM. This computer program has been used extensively to design and optimize geothermal power plants and power cycles of other types. GEOTHM can calculate the thermodynamics of a wide variety of cycles including refrigeration cycles. GEOTHM has an optimizer which permits single step multiparameter optimization of a thermodynamic cycle. A refrigeration cycle may be optimized for minimum cost refrigeration, or maximum thermodynamic performance using up to 55 optimizable parameters. The use of the optimizer in GEOTHM is demonstrated on two liquid nitrogen refrigeration cycles which are optimized for minimum refrigeration cost and minimum input power.

INTRODUCTION

The Lawrence Berkeley Laboratory has been developing a thermodynamic cycle program called GEOTHM since 1973. The GEOTHM program was first applied to water dominated geothermal energy cycles. (The program name comes from this application.) [1,2,3,4] More recently the GEOTHM code has been applied to simulate and optimize other types of electric and non-electric energy producing cycles. [5] This paper demonstrates the use of GEOTHM in designing and optimizing cryogenic refrigeration cycles using nitrogen as a working fluid.

The GEOTHM code is an extremely versatile thermodynamic cycle simulator which can design and optimize a variety of thermodynamic cycles. The primary features of the GEOTHM code are:

1. The thermodynamic processes are modularized into fundamental building blocks. The blocks can be arranged into any type of thermodynamic system.
2. The calculation of thermodynamic properties and transport properties is separated from the thermodynamic process elements.
3. The program is fast due to efficient programming in all of its iterative convergence routines. (For example, simple binary geothermal cycles require less than 100 ms of CDC 7600 computer time.)
4. The thermodynamic cycle generator in GEOTHM can be used as a function generator which is steered by a mathematical optimizer. The optimization routine can design and optimize a thermodynamic cycle with respect to any user specified criterion.
5. The inputting of data into the program is done interactively. A preprocessor detects many errors which the user makes in inputting data into GEOTHM. The program, which can be run from a remote terminal, is user oriented. [6]

GEOTHM has subroutines which model various thermodynamic processes which include: turbines, pumps, fans, compressors, isenthalpic expanders, flash tanks, direct

contact condensers, surface heat exchangers, surface condensers, pipe, geothermal wells, fossil fuel burners, wet cooling towers, and dry cooling towers.

Fluid properties are calculated using several equations of state. The program includes equations of state for pure water and sodium chloride brines with a concentration of up to 300,000 ppm. [7] The Starling BWR equation of state is used with nitrogen, carbon dioxide and nonchlorinated light hydrocarbons. [8] Two forms of the Martin equation of state can be used for the Freons and ammonia. [9,10] Air, noncondensable gasses and products of combustion are represented by a modified ideal gas equation of state. Thermal transport properties currently coded were obtained from the National Bureau of Standards. [11]

GEOTHM designs a thermodynamic cycle from user specified input data which includes: resource conditions (i.e., source and sink conditions), the configuration of the thermodynamic cycle and the thermodynamic efficiency factors for various components in the cycle. GEOTHM also requires a set of thermodynamic cycle or *state parameters* which provide the minimum information necessary in order to calculate the thermodynamics of the plant. The user selection of the kind and number of the parameters in this set is flexible in so far as the set adheres to the conventions by which the thermodynamic process routines within GEOTHM can compute the thermodynamics of the entire cycle. For example, the liquid nitrogen refrigeration cycle shown in Figure 1 is completely specified by the following *state parameters*: (1) The compressor exit pressure, (2) the exit temperature (or quality from the refrigeration load), (3) the exit pressure from the load (this often determines the refrigeration temperature), (4) the temperature of the high pressure inlet stream to the cryogenic heat exchanger, (5) the exit temperature of the low pressure stream from the cryogenic heat exchanger, (6) the pinch point temperature difference across the nitrogen-water heat exchanger, and (7) the temperature of the water entering the cooling tower.

Given the system *state parameters* and other required input data (i.e., for the cycle shown in Figure 1, the efficiency factors and the cooling tower air wet and dry bulb temperatures), the program then designs the cycle to satisfy a particular constraint such as net refrigeration at the load. The program also calculates a set of design performance factors such as input (or output) power, cycle efficiency, fluid mass flows, heat exchanger areas, etc. The cost of the plant is a function of these various design factors.

GEOTHM can be operated in two modes, the *passive design mode* and the *dynamic design mode*. When the program operates in the *passive design mode*, the program designs the power plant using user specified *state parameters*. In the *dynamic design mode*, the optimizer iterates upon the set of *state parameters*, now called *optimizable parameters* to rapidly converge upon the optimum cycle design. Optimization of thermodynamic cycles with up to 55 *optimizable parameters* is possible because the program is extremely fast. Any reasonable objective function (the function to be optimized) can be optimized using the program. For example, we have optimized power cycles for: (1) minimum plant capital cost, (2) minimum bus bar energy cost, (3) maximum energy yield per unit fuel flow, and (4) maximum cycle thermodynamic efficiency.

THE WORKING FLUID EQUATION OF STATE

The refrigeration cycle shown in Figure 1 uses three different working fluids. Nitrogen, atmospheric air (in the cooling tower), and water. Atmospheric air thermodynamic properties are calculated using a form of the ideal gas equation. Water is modeled within the program using the Keenan and Keyes equation of state. [12]

The nitrogen fluid properties are modeled using the Starling Modified Benedict-Webb-Rubin (BWR) equation of state. [9,13] This equation of state was chosen over the more standard NBS equation of state [14] for the following reasons: (1) The Starling equation of state requires much less computer time to run than does the more complex NBS equation, (2) the Starling equation of state is valid in both the dense liquid and light gas regions, (3) the Starling BWR equation of state already existed

within the code and Starling had already fitted the coefficients. The Starling equation of state and the NBS equation of state agree to within a few percent when the temperature is above 80°K. Below 80°K the agreement is not very good. Table 1 shows the two-phase dome parameters given by the Starling equation of state from the triple point to the critical point.

TABLE 1 The Liquid-Vapor Dome Generated by the Starling BWR Equation of State for Nitrogen Between the Triple Point and the Critical Point

TEMPERATURE (K)	PRESSURE (bar)	ENTHALPY* (Jg ⁻¹)		ENTROPY* (Jg ⁻¹ K ⁻¹)		SPECIFIC VOLUME (cm ³ g ⁻¹)	
		Liquid	Gas	Liquid	Gas	Liquid	Gas
63.106+	0.061	-1.0	352.2	-0.017	5.580	1.125	3043.
65.	0.104	31.3	353.0	0.504	5.453	1.141	1843.
70.	0.311	90.8	355.1	1.426	5.202	1.182	659.4
75.	0.697	127.1	357.1	1.958	5.026	1.222	309.0
77.383++	1.013	138.3	358.0	2.116	4.962	1.241	218.0
80.	1.327	150.7	359.0	2.289	4.893	1.262	169.7
85.	2.267	167.4	360.7	2.512	4.785	1.304	103.1
90.	3.589	180.4	361.9	2.676	4.694	1.350	66.81
95.	5.377	191.5	362.7	2.810	4.612	1.401	43.37
100.	7.719	201.9	362.8	2.927	4.535	1.461	31.82
105.	10.711	212.4	362.0	3.037	4.461	1.533	22.80
110.	14.456	223.7	360.0	3.146	4.386	1.625	16.54
115.	19.063	236.5	356.3	3.262	4.305	1.751	12.01
120.	24.649	252.2	349.8	3.396	4.209	1.945	8.592
125.	31.335	274.8	336.9	3.575	4.072	2.342	5.757
126.161+++	33.922	301.0	301.0	3.782	3.782	3.207	3.207

* Zero for enthalpy and entropy is defined at 63.16°K on the saturated liquid line

+ Triple point

++ Normal boiling point at 1.0 atm

+++ Critical point

CHARACTERISTICS OF THE REFRIGERATION CYCLES

The refrigeration cycles shown in Figures 1 and 2 are designed to generate 100 kW of liquid nitrogen refrigeration at a pressure of 1.4 bar (80.4°K). These cycles consist of the following major components: turbines, isenthalpic expanders, compressors, heat exchangers, pumps, and wet cooling towers. Three of the *state parameters* are fixed. They are: (1) The exit pressure at the refrigeration load (this is 1.4 bar), (2) the exit quality from the load (the quality is set to 1.0 so that saturated vapor enters the cryogenic heat exchanger at 80.4°K and 1.4 bar), and (3) the temperature of the water entering the cooling tower from the compressor aftercooler (this temperature is 323°K). The other four *state parameters* become *optimizable parameters*. The inlet air to the cooling tower has a wet bulb temperature of 25°C and a dry bulb temperature of 40°C.

The assumed adiabatic efficiency of the turbine is 0.65. The assumed adiabatic efficiency of the compressor and the water pump is assumed to be 0.7. The cryogenic N₂ heat exchanger U factor is assumed to be 100 WK⁻¹m⁻² (bare tube area). The compressor cost is around \$190 per kW of input power (this cost factor varies with pump power). The turbine cost (including the brake) is around \$800 per kW (the cost factor used varies with turbine power). The heat exchanger cost is calculated using an equation which consists of a constant plus a pressure dependent term which is multiplied by a cost factor. (At peak pressures below 21 bar, the heat exchanger cost is \$20,000 plus \$36m⁻² times the cost factor. At 66 bar, the heat exchanger

cost is \$20,000 plus $54m^{-2}$ times the cost factor.)

The method used to calculate the capital cost of the plant is as follows: (1) Each of the major components is sized then a cost for that component is calculated. (2) The cost of the plant is the sum of the cost of major components times a direct cost factor which accounts for installation, assembly, piping, instrumentation, foundations and thermal insulation. The direct cost factor used is 2.0. (3) The component cost times direct cost factor product is multiplied by an indirect cost factor to yield the plant capital cost. The indirect cost factor accounts for engineering, legal fees, contingency, overhead, escalation, and the cost of government regulation. An indirect cost factor of 1.7 was used. Therefore, the capital cost of the plant is 3.4 times the sum of the major component costs.

The annualized plant capital cost is calculated by multiplying the capital cost by a capital cost factor which includes interest on borrowed money, taxes, depreciation, and return on investment. For the purpose of this study, an annualized capital cost factor of 0.25 was used. The cost of operation and maintenance was assumed to be 7 percent of the capital cost. The annualized power cost is the product of the input power, plant availability factor, the number of hours in a year, and the cost of electrical energy. The assumed availability factor is 0.85 (the plant operates 85% of the time) and the assumed electrical energy cost is \$0.04 per kWh.

The cost of nitrogen refrigeration given in \$ per kWh at 80.4°K is simply the annualized plant cost divided by the refrigeration rate (given in kW) and the number of operating hours in a year. The cost of liquid nitrogen refrigeration is one of the objective functions which is optimized.

TEST CYCLE STUDIES

Two liquid nitrogen cycles are described in this report. Both are simple Hampton [15] cycles which generate 100 kW of refrigeration at 80.4°K. One cycle has a turbine expander with an assumed efficiency of 65 percent (see Figure 1); the other cycle has a J-T expander (see Figure 2). The authors made no claim that the cycles shown here are the best cycles to use. However, the cycles do demonstrate the optimization process in GEOTHM.

The turbine expander cycle shown in Figure 1 was optimized for minimum refrigeration cost (the cost is given in US\$ per kWh at 80.4°K) and minimum input energy (maximum cycle thermodynamic efficiency). The cycles shown in Figures 1 and 2 include various irreversibilities such as heat exchanger pinch conditions, pressure drops, and turbine, compressor, or pump efficiencies. The cycle shown in Figures 1 and 2 was optimized in five dimensional space (four optimizable parameters and the objective function - the function to be optimized). The four *optimizable parameters* are (1) compressor exit pressure, (2) the high pressure stream inlet temperature to the cryogenic heat exchanger, (3) the low pressure stream exit temperature from the cryogenic heat exchanger, and (4) the pinch point temperature difference across the compressor after cooler heat exchanger. The fixed *state parameters* are the refrigeration pressure, nitrogen exit quality from the load, and the water temperature entering the cooling tower from the aftercooler heat exchanger.

Table 2 shows the changes made in the *optimizable parameters* from the first trial plant design to the optimum plant design. The J-T valve expansion cycle shown in Figure 2 was optimized for minimum refrigeration cost. The optimization process, which is a single-step process that takes about 30 seconds of CDC 7600 computer time, reduced the objective function (the cost of refrigeration at 80.4°K) by nearly a factor of two. The compressor exit pressure was raised and the pinch point temperature differences were reduced. This yields more nitrogen refrigeration per unit of input power at the expense of plant capital investment. The computer quickly finds the trade off between power cost and capital cost.

TABLE 2 A COMPARISON OF A FIRST TRIAL PLANT DESIGN WITH AN OPTIMIZED PLANT DESIGN FOR THE CYCLE SHOWN IN FIGURE 2. THE PLANT IS OPTIMIZED FOR MINIMUM REFRIGERATION COST.

	First Trial Design	Optimum Design
Fixed state parameters		
1. Refrigeration pressure (bar)	1.4	1.4
2. Exit quality from the load at the refrigeration pressure	1.00	1.00
3. Cooling tower inlet temperature ($^{\circ}\text{K}$)	323.00	323.00
Optimizable parameters		
1. Compressor exit pressure (bar)	150.50	240.82
2. Cryogenic heat exchanger high pressure stream inlet temperature ($^{\circ}\text{K}$)	315.00	299.26
3. Cryogenic heat exchanger low pressure stream exit temperature ($^{\circ}\text{K}$)	305.00	298.45
4. Aftercooler heat exchanger pinch point ΔT ($^{\circ}\text{K}$)	2.72	0.47
Objective function refrigeration cost at 80.4°K (\$/kWh)	6.965	3.563

Table 3 compares the various optimizations. A comparison of columns one and two shows the difference between a thermodynamic and refrigeration cost optimization. A comparison between columns two and three shows the effect of heat exchanger cost factors on the cost optimization of a refrigeration cycle with a turbine expander. A comparison of columns three and four shows the effect of the choice of expander on cost optimization (the heat exchanger cost factor is the same for the two cycles). The plant capital costs shown in columns two, three, and four are consistent with the Strobridge cost curves. [16]

The thermodynamic optimization resulted in low compressor exit pressure and zero pinch point temperature differences. As a result, the input power is minimized but the plant capital cost and the refrigeration cost become very large. The cost optimized cycles operate at higher pressures and the pinch point temperature differences open up. There is clearly a trade-off between plant capital cost and operating cost. Roughly half of the annualized cost of the cost optimized plants is the cost of input power to the compressors and pumps.

The primary effects of increasing the heat exchanger cost per unit area is increased compressor pressure and larger pinch point temperature differences. The result is reduced heat exchanger surface area at the expense of greater input power to the compressor and pumps. Increasing the heat exchanger cost affects nitrogen mass flow, turbine inlet temperature, and the quality of the nitrogen at the exit of the turbine. (It is interesting to note that the optimum cycles all expand into the two-phase dome on the saturated vapor side. Expansion into the dome on the liquid side requires excessive compressor exit pressure resulting in excessive power consumption.) The use of a multistage compressor with intercoolers is expected to raise the compressor exit pressure for minimum refrigeration cost cycles.

TABLE 3

A COMPARISON OF THERMODYNAMIC AND COST OPTIMIZATIONS OF 100 KW NITROGEN REFRIGERATORS USING A SIMPLE HAMPTON CYCLE WITH TURBINE EXPANDERS

	Thermodynamic Optimization Turbine Expander	Refrigeration Cost Optimization		
		Turbine Expander (see Fig. 1)		J-T Expander (see Fig. 2)
Heat Exchanger Cost Factor	DNA	10.0	3.0	3.0
Cryogenic Heat Exchanger Cost** (\$m ⁻²)	DNA	601.3	130.2	451.6
Optimizable Parameters				
1. compressor exit pressure (bar)	9.12	65.69	21.53	240.82
2. cryogenic heat exchanger high pressure stream inlet temperature (°K)	298.24	301.51	301.51	299.26
3. cryogenic heat exchanger pressure stream exit temperature (°K)	298.24	296.64	299.07	298.45
4. aftercooler heat exchanger pinch point ΔT (°K)	-0-	2.73	2.55	0.47
Other Parameters				
nitrogen mass flow (Kgs ⁻¹)	4.633	2.259	3.242	3.031
cryogenic heat exchanger area (m ²)	∞	310.8	1037.3	469.4
aftercooler heat exchanger area (m ²)	∞	97.4	126.7	182.1
expander inlet temperature (°K)	100.00	151.44	117.04	166.38
expander exit quality (%)	89.6	78.6	85.1	84.1
compressor motor power input (MW)*	1.683	2.197	1.914	4.927
plant capital cost (\$)	∞	2.483 x 10 ⁶	2.002 x 10 ⁶	3.706 x 10 ⁶
refrigeration cost (\$per kWh)	∞	1.945	1.625	3.563

* To produce 100 kW of refrigeration at a pressure of 1.4 bar (the temperature is 80.4°K).

** Heat exchanger basic cost factors vary by a factor of 3.3. The rest of the cost difference is due to pressure dependent terms in the heat exchanger cost equation.

The use of a single-stage turbine expander instead of a J-T valve expander reduces compressor exit pressure by one order of magnitude and it reduces refrigeration cost by over a factor of two. The minimum compressor exit pressure needed to produce any liquid at all in a J-T expansion Hampton cycle is around 140 bar (roughly four times the critical pressure for nitrogen). Turbine expander Hampton cycles will produce liquid even at very low compressor exit pressure. It is clear why J-T expansion was abandoned in commercial liquid nitrogen plants.

CONCLUSIONS

The GEOTHM computer program can be used to design and optimize refrigeration cycles as well as power plant cycles. Optimization of the cycles in 5 dimensional space took less than 30 seconds of CDC 7600 computer time. The optimization was a single-step process. While simple thermodynamic cycles were used in this example, the GEOTHM program can be extended to design and optimize complex cycles using a variety of working fluids including mixtures of gases (for example, liquified natural gas).

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Figure 1
SIMPLE HAMPTON REFRIGERATION CYCLE
SINGLE STAGE COMPRESSOR & TURBINE EXPANDER

Figure 2
SIMPLE HAMPTON REFRIGERATION CYCLE
SINGLE STAGE COMPRESSOR & J-T EXPANDER

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