THERMOECONOMIC OPTIMIZATION OF AMMONIA REFRIGERATION SYSTEMS USING ATOM BOMBS TO KILL ANTS?

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ABSTRACT
The chemical process and power production industries routinely use thermodynamic analysis tools in the design of facilities and equipment. The designers of energy intensive industrial systems - chemical process plants, oil refineries, etc. are familiar with second law analysis and thermoeconomic optimization. However, designers in less energy intensive industries, and especially the designers of industrial facilities, are likely to ask "*thermoecowhat*? I can't mess around with optimizing *system equations*! I have a facility to design." Industrial facilities are the "environment" for the process and are typically treated as a given when designing an industrial process facility. Support systems and services such as lights, space heat/cooling, compressed air, refrigeration etc. are the environment and, like the natural environment, as long as they "work" they are often ignored.

Take an ice cream production plant. The plant takes in milk and ships ice cream. The focus is on the product and process, not the ammonia refrigeration system that supplies the necessary `cold'. The refrigeration system capacity and first cost are the primary considerations - energy consumption/efficiency is secondary (if considered at all). Refrigeration designers do not develop and optimize systems equations. They assume components are designed to operate efficiently. They lookup manufacturer's data and match components to meet system design requirements. Essentially they combine performance data from each component to predict overall system operation at fixed design conditions. Often a successful system design is repeated at various locations with only slight adjustments for local conditions. Typical design practice may or may not result in the refrigeration system with the lowest life-cycle cost for a given location and process.

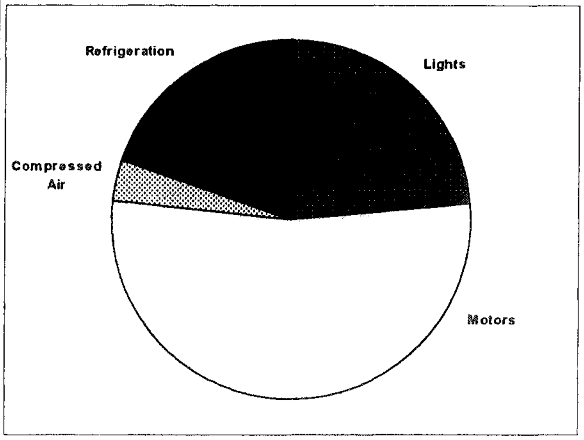
The manufacturer supplied data is empirical, usually presented in tabular form or supplied from the manufacturer for a given operating condition. Performance at off-design conditions and climate interactions are often not considered. Simulation or thermodynamic modeling, necessary for optimization, is very unusual for these types of systems. Thermoeconomic optimization of refrigeration systems is like using an atom bomb to kill ants. Or is it?

By using the available empirical data and the capabilities of modern desktop computers we have attempted to simplify the atom bomb into something resembling a very large hammer. Using the principles of thermoeconomic optimization and systems analysis we have developed a spreadsheet based computer model to help design life-cycle optimized industrial refrigeration systems. The model incorporates the effect of climate and load profile on system operation. It combines hourly outside air conditions, performance curves from component manufacturers and thermoeconomic analysis helps to define the most energy efficient and economical refrigeration system for a given climate and a defined system load profile. We have used this program to examine the effects of climate , load profile and condenser size on system performance and economics.

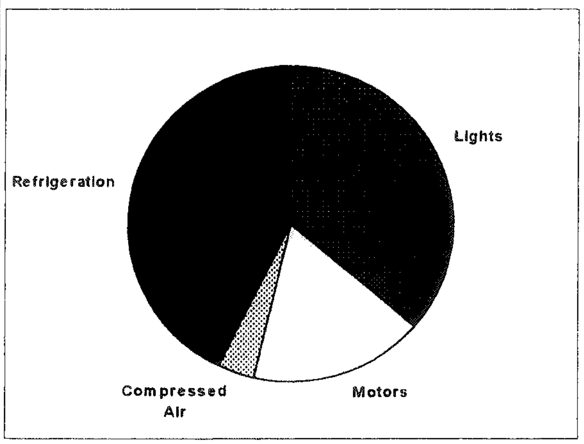
BACKGROUND
The work described in this paper was inspired by our experience with energy audits of industrial facilities and the discrepancies between design in the chemical process industry and that practiced in less energy intensive sectors, especially facilities for industrial process. This work is an effort to simplify inclusion of energy efficiency concerns into refrigeration system design decisions. We are trying to bridge the gap between thermodynamic analysis and actual equipment selection; specifically to incorporate climate effects into the thermodynamic analysis and energy calculations

The design of moderately sized industrial and manufacturing facilities is typically performed by an architecture and engineering (A/E) team. The AE team is responsible for the facility design while the owner and or equipment supplier handles the manufacturing and process equipment The owner wants a "home" for his process, with shelter, light, ventilation, heat and cooling - as well as auxiliary services such as compressed air and refrigeration. The facility and auxiliary systems are almost always secondary to the process and process equipment Second place inevitably receives less attention and effort. Energy using systems are designed and specified so that they will work - out of sight, out of mind. Seldom is the lifetime operating cost of the facility considered.

Figure 1 a Baseline Energy Consumption Characteristics



b Estimated Energy Savings Opportunity



We performed an energy analysis for the design of a recently built food processing facility. The facility had an estimated construction cost of approximately $12 million including approximately $5 million in specialized process equipment. The facility design included cold storage, process refrigeration and compressed air systems. Figure 1a below summarizes the energy consumption calculated for the facility. At local energy costs (which are low, $0.0285/kwh and $8.45/kw demand) this translates to an annual operating cost of $ 200,000 or $3 million over a typical 15 year life span. Forty percent of the energy cost is directly associated with the facility - lights, compressed air, and refrigeration.

We estimated available energy savings potential for the facility, including opportunities associated with lighting, compressed air, process motors and the refrigeration system. Approximately forty percent of the savings potential was associated with the refrigeration system. Energy savings modifications to the refrigeration system were mostly involved with the evaporative condenser and included over sizing the condenser and allowing the condenser temperature to float - change with the outside wet-bulb temperature and system load.

In this fairly typical situation, the energy savings potential associated with industrial refrigeration systems is significant. However, savings are seldom estimated, in part because they are so complex to calculate. Energy efficiency is a low priority design concern.

.REFRIGERATION SYSTEM

Figure 2a below is one thermodynamic representation of a refrigeration cycle showing the relationship between the pressure and enthalpy of the refrigerant (ammonia) as it flows through the system. Heat is added to the ammonia in the evaporation step, increasing its enthalpy. Work is used to increase the pressure (compression), heat is removed condensing the refrigerant and reducing its enthalpy, then the pressure is reduced (expansion) back to the starting pressure.

The pressure enthalpy (p-h) diagram in Figure 2 is simple representation of a refrigeration system but is not particularly useful for a system designer.. The designer must select and specify equipment based on early decisions about the working fluid and design pressures dictated by process and facility requirements.

The typical design begins with estimating the maximum load on the refrigeration system. The load includes the refrigerated space volume, losses from the space, product load, internal electrical load, load, plus other loads process peculiar to the design problem. Performance data from the component manufacturer's data are combined with other system data to yield information about how the system will operate and to assure will meet the design specifications. A safety factor of 1.2 is often used to ensure that the system will fill the need. Then - maybe - the designer will look at ways to improve the energy efficiency of the system.

Figure 2. Refrigeration Cycle Pressure Enthalpy Diagram (Ammonia)

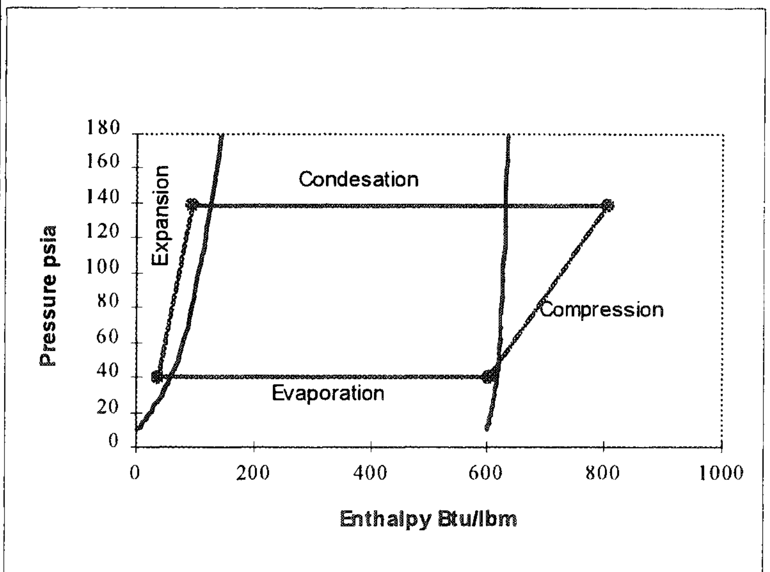


Figure 3 shows a schematic of a two stage refrigeration system with the basic components a designer must size, match and specify. In a two stage system there are four major energy intensive components two - compressors and two heat exchangers. The four components interact as an open system. The two heat exchangers interact with the environment - the evaporator through envelope loads, and the condenser which is directly affected by the environment.

The evaporator is the heat exchanger on the cold side of the system. The cooling requirements - load and temperature - dictate evaporator size. Saturated or slightly sub-cooled liquid refrigerant enters the evaporator coils. Air from the refrigerated space (or another coolant) is forced over the coils and loses thermal energy to the refrigerant. The liquid refrigerant evaporates as it absorbs the thermal energy, leaving the evaporator and entering the booster as a slightly superheated vapor.

The booster is a compressor which increases the pressure of the ammonia vapor. The booster adds both work and thermal energy to the refrigerant as it is compressed. As a result, the refrigerant is superheated at the exit of the booster. The ammonia vapor enters the intercooler where it condenses, causing some of the liquid refrigerant in the intercooler to boil. The equilibrium between this condensing and boiling determines the intercooler pressure.

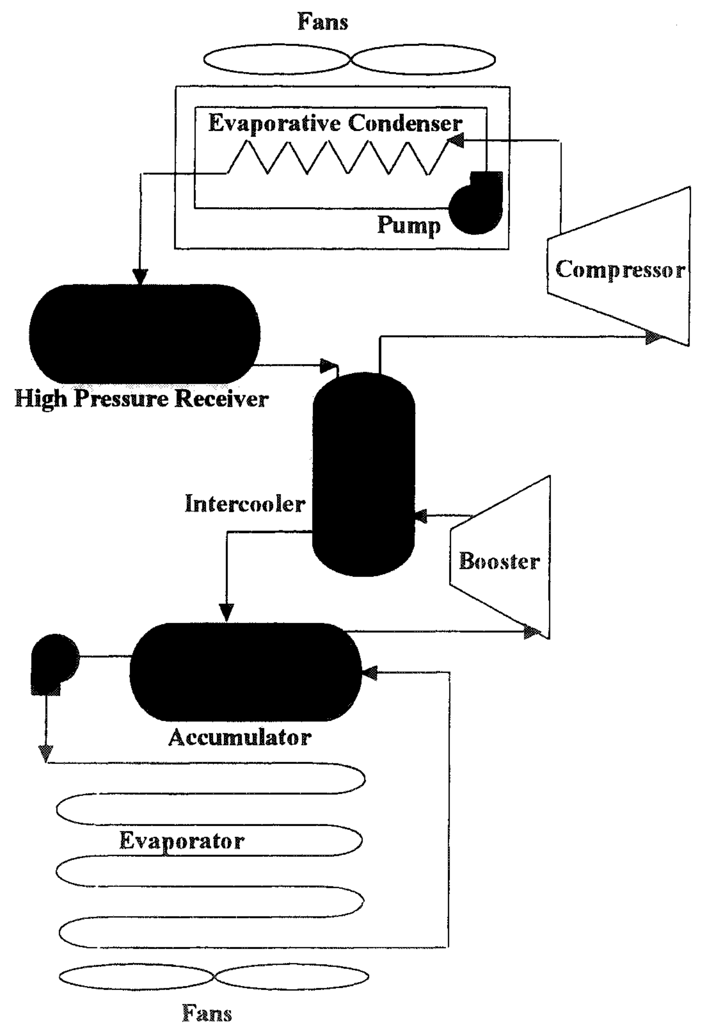
The vapor leaving the intercooler is compressed by the compressor, again resulting in a superheated vapor. The vapor enters the condenser - the heat exchanger on the hot side of the system. The vapor flows through the condenser coils, over which passes a coolant - usually water or air. The vapor condenses to as the heat from the refrigerant is absorbed by the coolant.

The high pressure liquid refrigerant leaves the condenser and is stored the high pressure receiver. The high pressure liquid is fed to the intercooler where it condenses the superheated vapor from the booster.

It's little wonder that energy efficiency analysis of such systems is neglected. The data available do not make it easy to calculate system energy consumption or to estimate the effect of design/equipment options on energy consumption. The task of accurately calculating energy consumption of such a system, changing one or more components and recalculating - i.e. optimization - is daunting and time consuming - especially since the calculation must account for the interaction of the system with the environment. In the real world the outside air dry bulb and wet bulb temperatures change and these temperatures can affect both the system load and component performance.

EXERGOECCONOMIC OPTIMIZATION

Figure 3. Two Stage Refrigeration System Schematic.



Exergoeconomics is a tool used to help improve overall system efficiency and lowering life cycle costs of a thermodynamic system. The exergy balance will show the system irreversabilities and their relative cost. Exergoeconomics has been successfully used in the design of complex thermal systems, however it generally uses a fixed reference environmental temperature, thereby disregarding local climate effects on the system.

For the system presented in Figure 3 we applied exergoecconomic optimization methods to one component of the system, the condenser. The goal was to optimize the condenser size while incorporating the effects of climate on condenser performance and, compressor energy consumption. Actual manufacture's data was used for the component performance characteristics. We quickly discovered that the gap between thermodynamic data in a text or reference book (i.e. the pressure enthalpy diagram in Figure 2) and the stack of tables, figures, and specifications supplied by equipment manufactures is intimidating and frustrating.

The maximum useful work in a system occurs when a process is completely reversible, that is, when there is no entropy generated by the process. The exergy of a process can be defined by the equation:

$$E = (W_c)_{MAX} = (U + KE + PE - U_o) + P_o(V - V_o) - T_o(S - S_o) \quad \text{(EQ1)}$$

where:

E =	exergy rate	U =	internal energy
W_c =	maximum work	KE=	kinetic energy
PE =	potential energy	U_o =	internal energy at environmental conditions
P_o =	pressure at environmental conditions	T_o =	temperature at environmental conditions
S =	entropy	S_o =	entropy at environmental conditions
V=	volume	V_o =	initial volume at environment conditions

The term exergy describes the quality of energy and is defined by Moran [1] as *"The maximum useful work attainable from an energy carrier under the conditions imposed by a given environment."* where the given environment is defined as "*Some portion of the surroundings for which the intensive properties do not change significantly as a result of any the processes under consideration.*"

Exergy is a measure of the work available from a system as it is brought into equilibrium with its environment. Once an environment is defined, a unique value can be assigned to the exergy in terms of the property values of the system. Therefore, exergy may be considered as an extensive thermodynamic property of a system once the state of the system and environment have been established.

$T_o(S-S_o)$ is the destruction of exergy by irreversibility's within the closed system. It can be called "exergy destruction", E_D. This term represents the destruction of the potential to perform useful work. Since the second law requires that entropy generated by a system always be equal to or greater than zero, the exergy destruction will also be equal to or greater than zero.

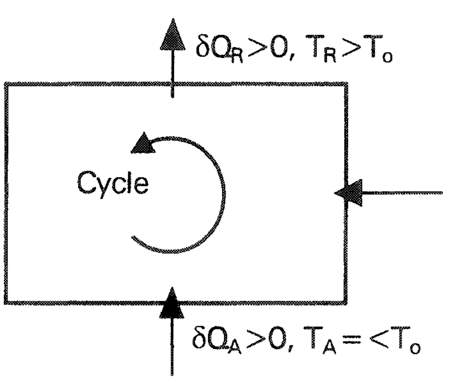
An exergy balance for a process in a steady-state flow may be written as:

exergy in = exergy out + exergy losses + exergy destroyed (EQ2)

Exergy losses are those associated with leaks and extraneous heat transfers. A gauge of how effectively the input is converted to the product is the exergetic efficiency ratio, ε, defined as

ε = exergy out /exergy in (EQ3)

Figure 4. Reversed Cycle Schematic.



Efficiency is commonly used as a gauge of the performance of a particular device or process. Energy-based efficiency defines quantities of energy. Improving energy based efficiencies emphasizes reducing of exergy loses. Exergy-based efficiencies account for second-law limitations. Improving exergy-based efficiencies emphasizes reduction of both exergy losses and exergy destruction - system irreversibility's.

Refrigeration is an example of a simple reversed cycle. Figure 4 is a schematic of such a cycle. A refrigeration system uses energy, in the form of work, to transfer energy from a low temperature to a high temperature. For a simple reversed cycle the coefficient of performance, β, is defined as,.

$\beta = Q_A / (Q_A - Q_R)$ (EQ4)

where Q_A is the energy added and Q_R is the energy rejected

The cooled fluid gains exergy and the environment loses exergy. The exergetic efficiency of a refrigeration system expressed in terms of coefficient of performance (COP) is:

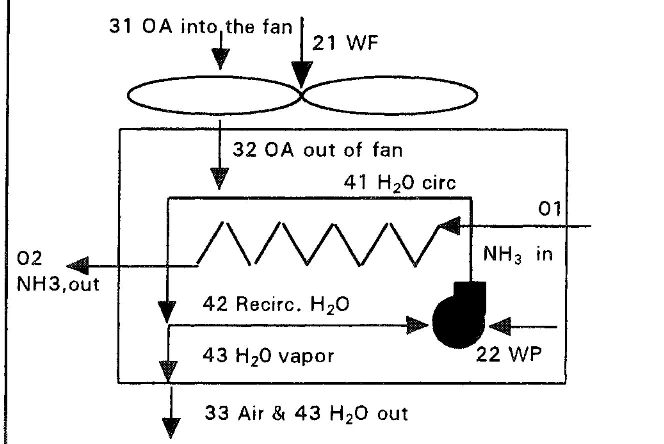
$\varepsilon = \beta\, E_A/Q_A$ EQ5)

where E_A is the exergy added to the system.

Thermodynamic Model of the Process.

For an exergetic evaluation of individual components or processes within a complex system, a control volume is drawn around each component, isolating it from the overall system.

Figure 5. A control volume around the evaporative condenser. *Energy streams of ammonia are identified with 0's, energy streams of electricity are identified with 2's, energy streams of air are identified with 3's and energy streams of water are identified with 4's.*



Energy streams in and out are identified in terms of EQ2. The desired product is defined by the process rendered by the component. It is assumed that the system has achieved a steady state. The addition of heat, Q_A, to a control volume is positive; removal of heat from the volume is negative. The exergy losses associated with individual components are minimal compared to the exergy destroyed and ignored in component optimization.

Consider Figure 5. The product of the evaporative condenser is the heat rejected which by definition has a negative value. As a result, the exergy destroyed during the heat rejection process will be negative.

The first law of thermodynamics (energy balance) and the second law of thermodynamics (entropy balance) for the evaporative condenser are given by:

$$W_{21} + W_{22} + (m_{01}h_{01} + m_{31}h_{31}) = (m_{02}h_{02} + m_{33}h_{33} + m_{43}h_{43}) \quad \text{(EQ6)}$$

$$W_{21} + W_{22} + S_{pr} + (m_0 s_{01} + m_{31}s_{31}) = (m_{02}s_{02} + m_{33}s_{33} + m_{43}s_{43}) \quad \text{(EQ7)}$$

h_{ii} represents the specific enthalpy of the corresponding mass flow stream. The kinetic and potential energies are considered to be negligible in the evaluation of the system components (although in most refrigeration designs the potential energy must be included in the evaluation when sizing the intercoolers and the accumulator). S_{pr} represents the entropy production within the control volume and s_{ii} is the specific entropy of the corresponding mass flow stream. S_{pr} is a measure of all irreversibilities due to heat exchange, friction, etc., associated with the evaporative condenser. S_{pr} can be calculated from equations 6 and 7.

The exergy balance of the flow stream through the evaporative condenser (where the subscripts F and P represent the fuel and product of the evaporative condenser) is:

$$E_{P,NH3} = m_{01}\,[(h_{02} - h_{01}) - To(s_{02} - s_1)] \quad \text{(EQ8)}$$

$$E_{F,NH3} = m_{43}\,[(h_{43} - h_{41}) - To(s_{43} - s_{41})] + m_{31}\,[(h_{33} - h_{31}) - To(s_{33} - s_{31})] + W_{21} + W_{22} \quad \text{(EQ9.}$$

Optimization

Typically more efficient components are more expensive. Increasing the efficiency to better utilize energy is limited by the increase in the initial cost. The tradeoff between energy savings and additional investment must be weighed carefully. Component cost typically dominates the final decision about the system components.

Calculation of the exergy destroyed by each component can give the owner/operator an idea of the thermodynamic performance of the system, but yields no information on where the major costs of the system occur. Exergy costing relates the thermodynamic value with the economic value of the energy carrier. Exergy costing is used to optimize the cost effectiveness of the system.

A monetary value is attached to each energy stream in the energy conversion process, *i.e.*, the electricity required to compress the ammonia. The value assigned is the total cost required to produce the energy stream.

A cost balance on a component shows that the total cost of the output streams is equal to the cost of the input streams plus the capital investment of the component. The cost balance for the evaporative condenser shown in Figure 3 is as follows:

$$C_{21} + C_{22} + C_{33} + C_{43} + C_{02} = C_{31} + C_{01} + Z_{EC} \qquad \text{(EQ10)}$$

The term Z_{EC} is calculated by dividing the annual capital investment costs, maintenance costs and other costs related to owning and operating the evaporative condenser (except fuel costs) by the average number of hours of system operation per year. C_n the total cost rate to produce the nth stream is written as follows:

$$C_n = E_n c_n \qquad \text{(EQ11)}$$

where E_n is the exergy rate, MBTU/H, and c_n is the average cost per exergy unit, \$/MBTU.

In the exergy evaluation described above, a product and fuel were defined for each component. The following equation is used to find the real cost source of each component. (The subscripts P and F indicate the product and fuel, respectively.)

$$\Delta c = c_{P,} - c_{F,} = (Z + c_{F,} \cdot (E_{D,} + E_{L,}) - c_{L,} E_{L,})/E_{P,} \qquad \text{(EQ12)}$$

where:

$E_{D,}$ = exergy destroyed during each process
$c_{L,}$ = cost of exergy losses and will be zero for individual components
$E_{L,k}$= exergy lost during each process and is assumed negligible for each component.

Several methods of optimizing the system have been presented in the literature, most require an extensive knowledge of the costs throughout the system and of mathematics. A method derived by Tsatsaronis and Krane [2] expressly for optimizing the major components of a system introduces a relative cost difference, r, where:

$$r = c_P - c_F / c_F = Z / c_F E_P + 1 - \varepsilon / \varepsilon \qquad \text{(EQ13)}$$

The relative cost difference is the average cost difference between the fuel and the product of the component. Minimizing r results in an economical and energy efficient component or system.

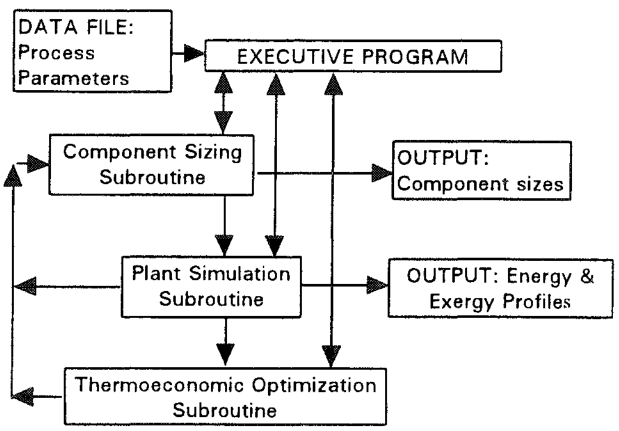
REFRIGERATION SYSTEM OPTIMIZATION

A computer program which uses empirical (supplied by equipment manufacturers) performance data and thermoecconomic optimization methods to size and optimize the evaporative condenser component of a two stage ammonia refrigeration system has been developed. The program simulates the operation, energy consumption and exergy destruction of a refrigeration system including the interaction of the system with the environment. Weather data obtained from NOAA (TMY data as is used for standard building simulation models) was used. The process parameters include an hourly load profile, weather data at the system location, temperature at which the refrigerated space is to be maintained, the type of booster and compressor used, and energy cost data.

Since the plant simulation is done for an entire year, the load profile should represent the operation over the year. The load profile is a function of the outside air dry bulb temperature (envelope load) and the time of day (process load). The wet bulb temperature at which the peak load occurs is initially used to size both the compressor and evaporative condenser. The size of the evaporative condenser is adjusted according to the simulation results.

Both the sizing and the simulation subroutines contain performance curves derived from component information found in manufacturer catalogs. These curves include the effects of part load, peak load operation and oil cooling for the booster and compressor.

Figure 6. Computer Program Schematic.



The ammonia mass flow rate is calculated from the peak load using the following equation:

$$m = Q/(h_g - h_f) \quad @(T_{RS} - 10°F) \qquad (EQ14)$$

where:

h_g = enthalpy of saturated ammonia vapor
h_f = enthalpy of saturated ammonia liquid
T_{RS}= temperature of the refrigerated space
Q = tons refrigeration (TR) * 12,000 BTUH/TR

Compressor/booster manufacturer catalogs define the maximum heat rate of each component in terms of tons refrigeration, TR, at a given saturated inlet and exit ammonia temperatures.

Boosters are sized using the evaporator exit temperature and the intercooler pressure/temperature. A good estimation [3] for the intercooler pressure, P_{int}, is

$$P_{int} = (P_c * P_e)^{1/2} \qquad (EQ15)$$

where:

P_c = condensing pressure.
P_e= evaporating pressure.

The intercooler temperature is the saturation temperature corresponding to P_{int}. The intercooler temperature is defined in manufacturers catalogs as the intermediate temperature. This number is rounded up to meet the intermediate pressure called out in the manufacturer's catalogs. Higher intermediate pressures result in lower temperatures at the end of high-stage compression. Oil used for lubricating the booster/compressor will remove some heat that is added to the ammonia during compression. Therefore the oil must also be cooled by the system. Liquid injection cooling imposes a load on the compressor (and hence the condenser) while thermosyphoning imposes a load only on the condenser.

The compressor size is based on the evaporator load, the booster performance and the type of booster and compressor oil cooling. The heat rejected by the condenser will include the evaporator load, the energy added to both the ammonia and the oil through the booster and compressor. This value, along with the peak coincident wet bulb temperature, is used to initially size the condenser.

Once the system is sized, an hourly simulation of the plant is used to predict an annual energy use and exergy destruction. This is done for each component and for the entire system.

The initial cost of each component was derived from manufacturer supplied cost. The cost of the evaporative condenser is based on the heavy section (coil region) of the condenser, the program can estimate condenser price if the weight of the heavy section is known. The cost of the booster or compressor is a function of the horsepower of the motor. Booster initial costs are slightly higher than those of a compressor. Electricity costs are input by the user.

RESULTS

The computer program was used to size and optimize the evaporative condenser in a two stage ammonia refrigeration system in two climates - Salt Lake City, Utah and Atlanta Georgia, using TMY weather data for the two cities. The booster and compressor models are derived from data supplied by McCormick Manufacturing and are specific to their equipment. Since the code is modular it is possible to modify it such that different brands of boosters/compressors can be compared according to the manufacturer supplied data. The evaporative condenser model is derived from data supplied by Baltimore Air Coil. The model for the evaporative condenser is interchangeable with other manufacturer data, since most are constructed with similar materials and in similar configurations. The code does not account for the flywheel effects associated with the thermal mass of the stored product. The input profile in this program was generated by assuming a peak and a dependence on outside air temperatures. Many simplifying assumptions were made in order to complete this analysis.

Refrigeration systems are often designed with a fixed condensing temperature, typically 75-85 F. One method for improving system energy efficiency is to allow the condensing temperature to "float", following the performance of the evaporative condenser as it is affected by the climate. Figure 7 shows the calculated energy annual energy consumption for fixed and floating point condenser temperature for two loads in two climates.

Figure 7. Comparison Of Energy Consumption For Fixed And Floating Point Condenser Temperature. S= Salt Lake City, A = Atlanta, P= process dominated load, E= envelope dominated load.

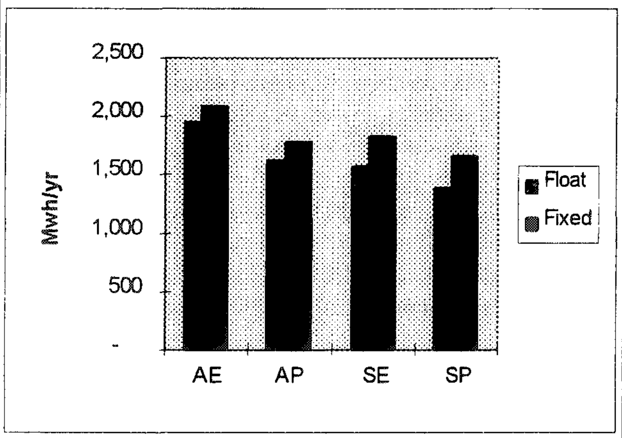


Figure 8. Energy Consumption and Relative Cost Difference for Various Evaporative Condensers Salt Lake City.

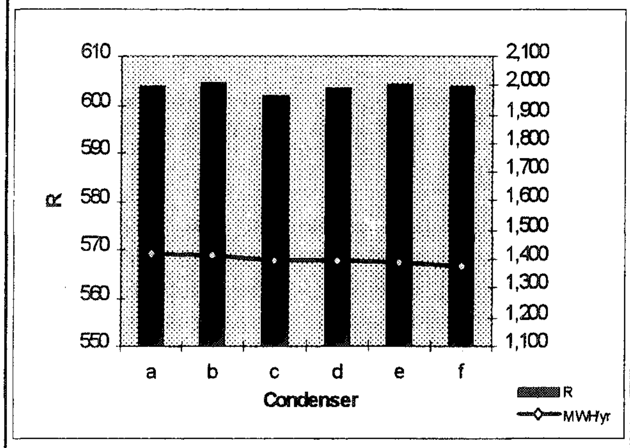
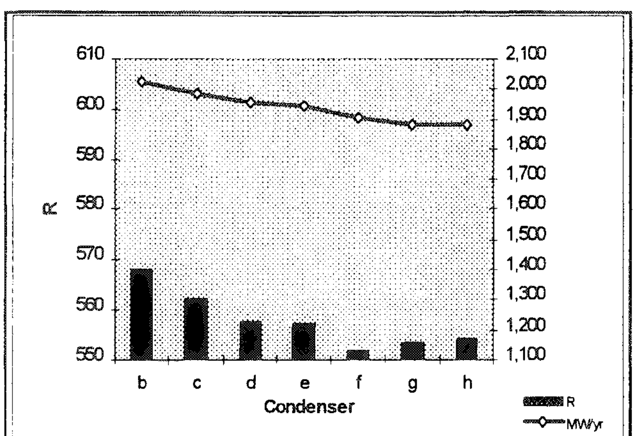


Figure 9. Energy Consumption and Relative Cost Difference for Various Evaporative Condensers, Atlanta.



Figures 8 and 9 show the results for the simulation of a two stage ammonia refrigeration system in two climates - Salt Lake City Utah and Atlanta Georgia. The model was used to simulate the system performance and optimize the evaporative condenser size given the same load characteristics and electricity costs in the two climates. (The condenser temperature was allowed to "float" for all cases.) As expected, the annual energy consumption was higher in Atlanta as was the optimum condenser size. (The condenser size increases as the letter increases - a is bigger than b etc.). The optimum condenser for each system has the lowest relative cost difference -R- indicated by the shaded column. The optimum condenser is not associated with the lowest system-energy consumption. With the optimum condenser, energy use was about 35% higher in Atlanta than in Salt Lake City. The relative cost difference for a given condenser was lower in Atlanta because the energy consumption was higher while the condenser cost was the same.

CONCLUSIONS

So, have we turned the atom bomb of thermoeccomomic optimization into a useful hammer? Not quite. The computer model as it exists is useful for evaluating the effect of the evaporative condenser performance on overall system energy consumption. It can be used to help select the optimum size condenser for a given load and climate. However, the model only calculates energy consumption of the system and relative cost factors for selected condenses leaving optimization is up to the user. This may or may not be a drawback since experienced designers can probably do a better job of "real" system optimization.

Other drawbacks include: The user must define the load characteristics. Only one manufactures compressor performance data is included. Calculations of the relative cost factor for multiple condensers requires several hours to run on a 486 computer. The program is currently in visual basic - a transparent but "piggy" language. Translating it to another language could improve the calculation time significantly.

It is also not clear that using the relative cost difference to determine the optimum condenser is more useful to a designer than calculating annual energy consumption and then performing individual life cycle cost analysis for various equipment configurations.

The simulated energy consumption correlates to manufacturer data, but no actual power, mass flow rates, temperature or pressure measurements have been done to verify the model. Such verifications should be performed.

The program is a first step toward a useful design tool. The model can be expanded. Additional system components can be added to the simulation and sizing subroutines. Much of the data required for the simulation - system loads and schedules and equipment performance data - are always required for a basic system design. The next step is to develop modules for various sized and types of other components i.e.- multiple compressor sizes and types, evaporator sizes and types. A simulation program which simplifies energy consumption and efficiency calculations will make it easier to include energy efficiency considerations in refrigeration system design.

In conclusion, thermoecconomic optimization of industrial refrigeration systems is probably using atom bombs to kill ants. But we have learned, in killing this particular ant, that combining a thermodynamic approach with standard equipment sizing and selection methods can be done and can be useful. We just need to modularize the bomb, giving the designer tools to perform the calculations needed to optimize both system energy efficiency and cost

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