BNL--45271

BNL-45271
Informal Report

DE91 004788

INNOVATIVE THERMAL COOLING CYCLES FOR USE IN COGENERATION

Angelo Skalafuris

August 1990

Work Supported by the
Utility Systems Division
Office of Utility Technologies
UNITED STATES DEPARTMENT OF ENERGY

ENERGY EFFICIENCY AND
CONSERVATION DIVISION
DEPARTMENT OF APPLIED SCIENCE
BROOKHAVEN NATIONAL LABORATORY
ASSOCIATED UNIVERSITIES, INC.

This work was performed under the auspices of the U.S. Department of Energy,
Washington, D.C. Under Contract No. DE-AC02-76CH00016

REGISTER
DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED
js

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, nor any of their contractors, subcontractors, or their employees makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency, contractor, or subcontractor thereof.

INNOVATIVE THERMAL COOLING CYCLES FOR USE IN COGENERATION

by
Dr. Angelo James Skalafuris, P.E.

1	CONCEPTS	1
1.1	Environmental & Sociometrical Considerations	1
1.2	Environmentally Benign Organic Fluids	3
2	THEORETICAL WORKING CYCLES	4
2.1	The Carnot Cycle	4
2.2	The Clausius Cycle	5
2.3	The Rankine Cycle	7
2.4	Limitations on Rankine Cycle Efficiency	8
3	WORKING FLUIDS	9
3.1	Comparative Criteria	9
3.2	Previous Researches	10
4	LIMITATIONS OF ENGINES	12
4.1	Technological Limitations	12
4.2	Chemical Limitations	14
5	SECTION SUMMARY ON DESIGN PHILOSOPHY	15
6	INNOVATIVE THERMAL REFRIGERATION CYCLES	16
6.1	Jet Refrigeration	16
6.1.1	Circuit Configuration	16
6.1.2	General Considerations	18
6.1.3	Aerodynamic Considerations	18
6.1.4	Supersonic Computations	20
6.1.5	Thermodynamic Considerations	24
6.1.6	Mechanical Considerations	25
6.1.7	Chemical Considerations	26
6.1.7.1	Toxicity	26
6.1.7.2	General Properties of Ammonia	27
6.1.7.3	Ammonia-Water Mixtures	30
6.1.7.4	Transport Properties	31
6.1.7.5	Jet Nozzle Implementation	31
6.1.7.6	Jet Cycle Implementation	33
6.1.8	Jet General Conclusions	34
6.2	Bottom Cycle Turbine-Compressor	35
6.2.1	Performance with Freon 114	35
6.2.2	Performance with Fluoronol 85	37
6.2.3	Turbine-Cogenerator Integration	38
6.2.4	Turbo-Compressor: General Conclusions	39
7	Cogenerator Heat Exchanger Configuration	40
8	Cogenerator Cycle Consolidation	41
9	Comparative Economics	41
9.1	Cost Calculation Assumptions: Table 7.	41
9.2	Electric Vapor Compression	42
9.3	Jet Refrigeration	42
9.4	Turbo-compressor	42
9.5	Electric Vapor Compression vs. Thermal Refrigeration	43
9.6	Environmental Impacts	43
10	Acknowledgements.	43
11	BIBLIOGRAPHY	44

1 CONCEPTS

1.1 Environmental & Sociometrical Considerations

The recent confirmation (1988) by scientists from many nations, of an ozone hole in the arctic region has affirmed that life on earth may now be threatened by an extra terrestrial recessive cycle due to accumulations of freons, i.e., chlorinated fluorocarbons (CFC's), which migrate to the upper atmosphere. Even to halt the present depletion of ozone due to the catalytic action of free chlorine atoms liberated from CFC's, some 80% of the manufacture of chlorine compounds which include freons must cease. Although too many, the fatal consequences of this discovery is hard to grasp, the reasons for it are quite clear. Freon was invented to replace carbon tetrachloride (CCl_4), which was found to be an excellent refrigerant; but in air, released a poisonous chlorine gas. Its neutralization was accomplished chemically by replacing two of the four chlorine atoms with fluorine which made the substance more stable on earth; hence it was originally called a chlorinated fluorocarbon and this led to a large class of CFC's of similar chemical structure. Freon is a partially fluorinated, partially chlorinated hydrocarbon since chlorine remains in the molecular structure and continues to remain in all species of chemicals which are sisters to CCl_4 . Unknown at the time of discovery, chlorinated hydrocarbons when released on earth, diffuse into the upper atmosphere where the chlorine is removed from the freon photoelectrically by the sun's radiation. The free chlorine atom then attacks an ozone molecule removing an oxygen atom from it, thus neutralizing the protective screen ozone offers against harmful solar radiations, especially ultra-violet. Later, the chlorine disposes of its captured oxygen in another reaction and is made free to repeat the cycle again. The life time in the upper atmosphere of the freed chlorine atoms is over 100 years, during which time it endlessly can play it's repetitive role as a catalyst in the destruction of the insular layer of ozone. When chlorine is ultimately removed from the atmosphere, it descends as acid rain inducing more terrestrial destruction.

It is a fact that living forms make absolutely no constructive use of halogen compounds and they are foreign substances. Practically all pesticides (DDT) and herbicides are made of chlorine compounds, the most electronegative substance found in nature. It is used in swimming pools, drinking water and a halogen fluoride is used in tooth paste, all applications for destruction of bacteria by which it inhibits the metabolism of certain of their enzymes. Iodine, used in the treatment of over active goiter problems acts as a purely inhibitory agent and not rejuvenatory. In order to thwart the regression of the earth's ozone layer it would be necessary to employ biodegradable organic

compounds as refrigerants and in building materials. The materials which are benign to life on earth are composed of the basic atomic building blocks of living systems, i.e., carbon, hydrogen, oxygen and nitrogen which do not induce extra-terrestrial chemical changes. The objection to carbon dioxide is its action as a screen to the earth's infra-red reradiation; but, causing no chemical changes and so would have a solution to its insular effect in principle, which lowers the earth's temperature. One has to distinguish between terrestrial and extra terrestrial contamination as well as reversible and irreversible pollution. Most of the terrestrial contamination is water pollution and comes from industries as well as the invention of the flush toilet. In prior times, the biochemical oxygen in the earth promoted the growth of aerobic bacteria which decomposed most complex organic compounds in sewage and waste into simpler and unobjectional chemical species. There are no such bacterial colonies in outer space by which we could hope to neutralize our extra-terrestrial waste.

Water pollutants can be classified into the following types:

1. Oxygen Consuming; Excreta and decay products of humans, animals and vegetation.
2. Infectious Agents; Viruses and bacteria.
3. Organic Substances; Detergents, oil spills, insecticides.
4. Inorganic Chemicals; Mercury, Cadmium, Lead.
5. Plant fertilizers; Nitrates, phosphates.
6. Colloidal Suspensions; Silt from land erosion.
7. Thermal Pollution; Electric utilities.
8. Radioactive Pollution; Fallout products and wastes of nuclear origin.

The most dangerous of the group are No. 5 because many of these substances are fatal when injected in minute traces (Arsenic, Beryllium, Cadmium, Lead and Mercury). However, all of these problems can be handled by proper management -- the extra terrestrial pollution is not managed so easily .

The major air pollutants can be classified into the following types:

1. Suspended particles; Aerosols, Soot and dust.
2. Sulfur Oxides; SO_2 , SO_3 (acid rain).
3. Nitrogen Oxides; NO , NO_2 .
4. Hydrocarbons; CFC's, gasoline fumes.
5. Carbon Monoxide (CO) and Carbon Dioxide (CO_2).

By far the largest contributor of the atmospheric pollutants, by over 50%, are motor vehicles by unventing gasoline, burning fossil fuels and the charging and discharging automotive air conditioners, most of the pollution occurring during transient demand. i.e., acceleration and deceleration.

1.2 Environmentally Benign Organic Fluid

Any fluids which contain only hydrogen, oxygen, nitrogen and carbon, the basic building blocks of living systems, should be environmentally benign. All Halogen compounds especially those which contain chlorine in the gas phase, such as freons (CFC's) should be considered detrimental to the environment. Some of the biodegradable compounds which have been employed as refrigerants in the past include, carbon dioxide, ammonia, potassium and ammonium hydroxide and water. Although many biodegradable refrigerants, such as ammonia (NH_3) are irritants, their prohibition by regulatory bodies has been brought on more by political rather than scientific pressures. Therefore, this report, in its search for more efficient methods of energy conservation, will consider only such reference biodegradable organic fluids. The only exception will be the analysis of carbon tetrachloride which is the prototype chlorinated hydrocarbon, and this will be done for purposes of comparison with biodegradable fluids.

A brief review of some basic and inexpensive biodegradable fluids and their thermodynamic properties follows:

Chloroform C_2HCl_3 . Once used as an anesthetic, now believed to be a carcinogen. Boils at 140F. Specific Gravity 1.48. Is a solvent of ammonia.

Ethyl Alcohol $\text{C}_2\text{H}_5\text{O}$. Light colorless liquid with an agreeable odor. The fluid addicts approximately 9% of the world's population. Used as a topical antiseptic. Inflammable, colorless, immiscible in water. During combustion or metabolism, emits an absolutely benign exhaust composed of water, carbon dioxide and an aldehyde, (the organic basis of perfumes). Boils at 172F. Specific Gravity 0.79

Acetone $\text{C}_3\text{H}_6\text{O}$. Is a ketone, produced in the body as a toxic substance after ingestion of alcohol. Also released with the metabolism of fats. Common to diabetic circulatory systems. It has a strong unpleasant odor. Is used as a commercial solvent and is considered toxic although not poisonous. It boils at 135F. Specific gravity 0.80.

Carbon Disulphide CS_2 . Is a pungent clear poisonous liquid due to the presence of sulphur. Boils at 115F. Specific gravity 1.3

Ether $\text{C}_4\text{H}_{10}\text{O}$. There are many chemical variations of the basic diethyl ether, which is the most common, being used as an anesthetic. It is comparatively inert and non reacting; is highly flammable, slightly immiscible in water and used as a solvent.

Carbon Dioxide CO_2 . Commonly known as dry ice and as a refrigerant to obtain very low temperatures. Also used in liquid form under pressure of some 58 atmospheres as a fire extinguisher. It is the natural consumptive product of plant life and the excretory product of animal life constituting the basic life cycle of exchange between them. Combustion results in warm carbon dioxide rising into the atmosphere which prevents the escape of infrared rays from the earth, thus creating an insular blanket which heats the earth. The depletion of ozone causes transparency of the ultra violet rays which have the same effect. The gradual extinguishment of plant life in large forests and jungle is believed to be destroying the natural consumptive sink for carbon dioxide as well as restricting the earth's only source for oxygen. Presumably, the problem can be solved by turning the planet green again with sufficient flora. No such theoretical solution exists for the escaping chlorine from CFC's. However, since 1950 the average temperature of the world has dropped somewhat when compared to records from 1850 onward. This has been attributed to the particulate matter and aerosols that have entered the atmosphere from industrial combustive processes. They form a dispersion of tiny mirrors which reflect the sun's rays and thus limit the heating of the earth by the sun in the manner of the theoretical nuclear winter. Triple point at -135°F and 47 psi for liquid-gas-solid phase.

Carbon Tetrachloride CCl_4 . Once used for dry cleaning and fire extinguishers; later discovered to be a carcinogen, forms phosgene in air at high temperature, a poisonous gas. Boils at 168°F . Specific gravity 1.6. The prototype fluid for the development of Freons.

2 THEORETICAL WORKING CYCLES

2.1 The Carnot Cycle

Water is generally the preferred working fluid in most heat engines, due to its inexpensiveness, good heat transfer properties and high latent heat during phase changes. The question then arises if there are other fluids in nature which can compete with water in these properties and yet be environmentally benign. First, one must have a criterion for evaluating fluids and in many cases this has been taken to be the Carnot efficiency. The Carnot efficiency is the limiting theoretical efficiency for any engine and is a solely a function of the ratio of absolute temperatures within which the engine operates; $(1 - t/T)$. The Carnot cycle forms a rectangle in the TS plane, as illustrated in Figure 1., and the enclosed area, bounded by isotherms and adiabats represents the work which is achieved.

FIGURE 1. The Carnot-Clausius Cycles

2.2 The Clausius Cycle

However, the Carnot cycle is an impractical criterion for a vapor, because there must be a change of phase between liquid and gas. In one sense, the success of a working engine depends upon the physical fact that pumping a liquid to high pressure is more efficient than pumping a vapor, while on the other hand, expanding a vapor is more efficient than expanding a liquid. Thus, in order to have a working cycle, the working fluid must be converted to liquid in order to be pumped to a the high pressure efficiently and then converted back to a vapor in order to realize the maximum work upon expansion. The necessity for phase changes requires the cycle to cross the saturation curve of the working substance. There are many cycles which can transgress the saturation curve, from liquid to gas and back again, the most elementary being the Clausius cycle. The Clausius cycle also operates between two temperature limits, represented by the same horizontal lines in Figure 1 which determine the Carnot efficiency. The cycle begins on the liquid line at the lower temperature (1), is then pumped up to the second horizontal line to a higher pressure (2) and preheated to match the pressure of the boiler. Arriving at the boiler, it crosses the saturation curve at constant temperature during a vaporization process, converting the liquid to a gas (3). Having arrived to opposite side of the triple point, the gas is now expanded to extract work from it. This expansion cycle is represented by a descent down the saturation curve to the lower horizontal line or direct adiabatic descent into the saturation region (4). The gas is then

condensed, and its state point traverses back through the saturation curve in the opposite direction to close the cycle.

The Clausius cycle takes into consideration the basic chemistry of liquid-gas conversion. It can be seen to be less efficient than the Carnot cycle because of the sloping shoulders of the saturation curve which makes the cycle deviate from rectangularity. The necessity for rectangularity can be easily seen by dividing the area of the work cycle into vertical differential rectangles, all of which have coincident expansion and pumping paths, ascending and descending, which cancel one another, all, except the first and last. Then one can see that the differential cycles which are bounded by the saturation line enclose comparatively smaller areas. One can also see that the wider is the width of the saturation curve crossings, the more insignificant become the contributions from the shoulders of the saturation curve. For this reason, a fluid with high latent heat inherently delivers greater efficiency to a cycle.

The efficiency of a Clausius cycle can be written as,

$$\text{Clausius Efficiency} = [H_2 - H_1 + L_{23} - T(S_4 - S_1)] / [H_2 - H_1 + L_{23}] \quad (1)$$

(H is enthalpy, L is latent heat, S is entropy and t is the sink temperature).

The shoulders of the trapezoidal area of work can be made smaller by choosing a vapor which has a low specific heat capacity, since the entropy along a saturation curve can be shown to be directly proportional to the specific heat and the natural log of the temperature ratio. The latent heat is proportional to the boiling point and so a good working fluid is one that has a high ratio of latent heat to specific heat or high boiling point to specific heat capacity. An alternative definition of this ratio can be derived by using the Clausius-Clapyron equation, which states that the slope of the saturation curve is proportional to the latent heat divided by the temperature and divided by the difference in specific volumes of gas and liquid. Therefore, one should choose a working fluid that expands to a large specific volume when vaporized and has a low specific heat.

We apply this simple rule to a number of elementary compounds to test its accuracy. The table which follows shows the method of this computation.

Table 1. RELATIVE EFFICIENCIES OF VAPORS (300F & 60F)

Vapor	Sp. Heat	Latent Heat	Ratio
Water	1.0	980	980
Carbon Bisulphide	0.25	140	560
Ethyl Alcohol	0.80	370	463
Chloroform	0.24	110	458
Carbon Tetrachloride	0.21	80	381
Acetone	0.58	212	365

Table 1. has been ordered in accordance with the best fluids on top.

We can now actually calculate the efficiencies according to the formula for a Clausius cycle to see if the order is preserved. The results are as follows. First we calculate the Carnot cycle efficiency as a check upon what follows.

$$\text{Carnot Efficiency} = [300-60]/[300+460] = 0.47$$

Table 2. Clausius Cycle Efficiencies

Water	0.31
Carbon Bisulphide	0.28
Methyl Alcohol	0.29
Chloroform	0.26
Acetone	0.25
Carbon Tetrachloride	0.24

One can see that Table 2. corresponds essential with the order of Table 1. confirming the validity of this method of choosing promising organic fluids for Clausius cycle machines.

2.3 The Rankine Cycle

The Rankine cycle is inherently more efficient than the Clausius because it extends the heat preparation process beyond the vapor saturation line into the superheated region along a line of constant pressure, as seen in Figure 3. There, the extension of the cycle into the super heat region, up and above the saturation curve adds differential Carnot cycles whose work contributions are seen to be superior to those of the Clausius cycle. Vapors with low triple points lend themselves to superheating and therefore would produce more efficient Rankine engines. However, the expansion in the superheat region, generally does not extend down to exhaust pressure, but at some higher value from which a line of constant volume is followed into the saturated region where it intersects the condensation line at atmospheric pressure. Hence, the work is given by the enthalpy difference along the vertical adiabat plus the work of constant volume expansion into the saturation curve. The efficiency of a Rankine cycle can then be written as,

$$\text{Rankine Efficiency} = [H_4 - H_5 + V_4(P_5 - P_6)] / [H_4 - H_1], \quad (2)$$

which assumes complete adiabatic expansion so that the expansion curve is vertical in Figure 2, not following the saturated vapor line.

FIGURE 2 The Rankine Cycle

2.4 Limitations on Rankine Cycle Efficiency

Generally the upper pressure of any cycle is limited by structural costs of heat exchangers or boiler. Off the shelf, this has been about 150 psi, but can easily be extended to 250 psi by custom fabrication. For steam, this would correspond to a temperature of about 400F. However, the highest temperature of the cycle is determined by the heat source, not the fluid, and in waste heat recovery systems as well as cogeneration. A practical temperature would be about 600F. This would mean that steam can be usefully superheated in most waste heat recovery systems.

Although the ambient temperature of air or cooling water determines the lowest possible exhaust temperature, there is another more important consideration at this lower limit and this is the necessity to maintain the vapor state of the working fluid throughout the working expansion cycle in order to maximize efficiency as well as avoid erosion of cylinder or turbine blades. Thus one must consider the practical limits for maintaining a vacuum in a closed cycle. In an internal combustion engine which exhausts into the atmosphere, the limit is one atmosphere (bar) of pressure, which for water means 212F. Thus, water expansion cycles must be closed cycles in order to reach lower temperature limits and so realize their full Carnot efficiency. A vacuum of 28 inches of mercury is a steam temperature of 100F closely akin to a good dry cooling

temperature (i.e., air cooling). The steam is then at less than 1 psig.

Accepting these as practical limits on a cycle, we can compute 3 different Carnot efficiencies for water,

$$\text{Non condensing steam efficiency} = (400-212)/(400+460) = 0.22 \quad (3)$$

$$\text{Condensing steam efficiency} = (400-100)/(400+460) = 0.35$$

$$\text{Condensed superheated steam} = (600-100)/(400+460) = 0.47$$

This illustrates that a closed system with superheated vapor in which expansion is permitted to the practical limits of vacuum, may be more than twice as efficient as a simple non condensing open system.

3 WORKING FLUIDS

3.1 Comparative Criteria

If we assume that our parameters for the model will be 600F or 250 psi on the high side and 28 inches of vacuum on the low side, then we can examine organic fluids and compare them with water on the following basis.

1. A fluid may have a higher temperature at 250 psi without superheat. This implies that more entropy in the heat source can be transferred to the fluid because boiling, a change of phase, is a more efficient process of heat transfer than transfer to sensible heat, requiring less heat transfer surface. But more important, the Carnot efficiency increases with higher boiler temperature.

2. A temperature of 600F may lead superheat, or more superheat, thus facilitating the use of a Rankine cycle and realizing the higher differential efficiency of the superheat region. In this case, expansion must be complete so as to leave no superheat for condensation because regenerators for superheat are expensive and difficult to design. However, a low boiling point, may imply a low latent heat, which by the criterium of page 6. would decrease the rectangularity of the cycle in the TS plane and lead to inefficiency.

3. A lower temperature might be achieved at 28 inches vacuum, in which case the Carnot efficiency would be increased accordingly. However, there is a limiting temperature on the low side determined by the cooling water or air. In the case of water, this is about 60F. In the case of dry cooling, which usually is less expensive and environmentally preferred, this

would be about 100F. If we were to use cooling water in a closed cycle, then the Carnot efficiency would rise to the following;

$$\text{Closed Condensed Superheated Efficiency } [600-60]/[600+460] = .51 \quad (4)$$

There are many fluids which meet condition No.1 but all of these fluids violate condition No.3. and give a higher condensing temperature. The question is now more than just the relative merits of a higher boiler temperature relative to a higher condenser temperature. The preferred gas is one that boils at both ends, at the high side temperature and pressure as well as the low side condenser temperature and pressure. Also the pressure would have to be no more than a few psi at the condenser if the fluid were to be transported to a cooling tower. However, this restriction is not present in a cogenerator which is compact and transfers its heat to incoming domestic water. In such a case domestic hot water would require a temperature of 120F. In that case, the Carnot efficiency for a closed condensable system would be,

$$\text{Cogenerator Carnot Efficiency } [600-120]/[600+460] = 0.45 \quad (5)$$

If we now apply some of these restrictions to Table 2. we would see that water and ethyl alcohol, fluids 1 and 3 in importance would be eliminated because at the temperature of 60F each would reach limits of vacuum pressure that extend below the practical limit we have imposed of 28 inches of mercury. This means that we have found the Achilles heel of water. Organic fluids can make better fluids for working cycles if their pressure remains much higher than that of water at the condenser stage.

3.2 Previous Researches

The most exhaustive analysis of working fluids for use in Rankine cycles was carried out by D.R. Miller (1974) with the intended use in solar powered cycles with supplementary heating through the burning of some hydrocarbon fuel in an external combustor. He examined the following fluids; water, isobutane, toluene, chlorobenzene, benzene, hexafluorobenzene, thiophene, freon 113, pyridine, flutec PP3, Fluoronol 100. The most efficient fluids turned out to be, chlorobenzene, pyridine, toluene, thiophene and benzene, all of which were within a few percentage points of one another. However, Miller permitted high top side temperatures of 1100F because of supplementary heating. He assumed that there was 80% generator efficiency and 75% expansion efficiency (including pump work). Because water pressure is unacceptably high at 1100F, it was disqualified as a candidate for the Rankine cycle operating at this temperature.

When the fluids of D.R. Miller are ranked by the ratio of latent

to specific heat, there appears to be no correlation with the resulting efficiencies. The reason for this is that these cycles are Rankine cycles with a great deal of superheat. A cogenerator would not have that much superheat with normal organic fluids so that its Clausius efficiency would not differ from the Rankine. However, if a fluid is found which has a low triple point that would produce much superheat, then one might have to abandon the simple criterion of maximum volumetric expansion upon vaporization and actually work out the details with an extensive computer routine.

The Monsanto Research Corporation (1972) was contracted by ERDA to produce a number of desirable fluids for Rankine cycle application to motive vehicles. They came up with two candidates; 60% Pentafluorobenzene with 40% Hexafluorobenzene and 65% Water with 35% Methylpyridine. It is not clear if these materials were manufactured by Monsanto and to what degree the company could be considered a neutral authority. The principle author of the Monsanto report was D.R. Miller who later transferred to the government laboratory at Sandia where he undertook the extensive computer simulations cited above. The Monsanto choices were not repeated in the later Sandia report. Aerojet Corporation, working under an ERDA contract chose a proprietary mixture of Hexafluorobenzene and Perfluorotoluene which they patented. However, their cycle sustains a maximum pressure of 1000 psi and a temperature of 650F and these fluids are chlorinated. They obtained an efficiency of 35%. Thermo Electric Corporation chose a mixture of Trifluoroethanol and water called Fluorinol 85 (the 85 meaning %) for a working fluid. The choice was made on the basis of toxicity, chemical stability, flammability and low freezing point but not environmental sensitivity. Their ultimate model produced between 8.5 to 15% efficiency. The fact of the matter is that exterior circumstances has played a role in the overall efficiency and performance of these machines. Above all, is the temperature source for the boiler. The best fluid at 1000F boiler temperature may turn out to be the worse at a lower boiler temperature. Part of TECO's problem was the transient loading of their turbine when powering a motor vehicle. The choice of a gear coupled turbine for motive applications is always a poor one because the turbine is a constant speed machine and is not expected to perform well off the design speed. Thus, a different criterion is chosen for each application, which depends upon the heat source as well as the application, even a jet device or a turbine bottom cycle machine would require different loading criteria in such application. However, the cogenerator operates at constant load and therefore one can make a valid comparison.

Argonne National Laboratory (1986), after a review of the literature chose to examine the stability of Fluorinol 85. They found insignificant degradation at temperatures above 757F after some 500 hours of exposure. The most significant degrading product was hydrofluoric acid, among the most corrosive acids

which can be created in a laboratory. Because all Freons contain fluorine as well as chlorine, once the chlorine is released, the Freon still contains fluorine. What the consequent effects of the fluorine molecule are, has not been discussed in the literature. During photodisintegration of CFC's by the sun, the chlorine atom is set free, leaving behind the Fluorine. The terminal phase of the freed chlorine is the formation of hydrochloric acid whose effects are long term. However, no investigation has been seen in the literature of the terminal phase of fluorine in the atmosphere whose acid compound is much more corrosive. This is an especially important research now that short term heating of Fluorinol at Argonne N.L. has shown it to result in hydrofluoric acid.

4 LIMITATIONS OF ENGINES

4.1 Technological Limitations

An important practical consideration in the expansion turbine cycle is a limitation upon the expansion ratio, generally at about 4 to 1, and not necessarily a limitation by the Carnot temperature. This is because of the extreme detrimental effect over the long run of cylinder condensation with more complete expansions. Some vapors such as carbon tetrachloride and the freons remain dry even after long expansions because their saturation curves have a negative slope in the vapor region, so, such fluids in general, can be expanded to their thermal limits. This is one of the advantages of the Freons and one of the principal limitations of water, for use in expansion cycles. Thus as a general rule, one must find a fluid which will have a volume expansion of about 4 to 1 between the two temperature limits of boiler and condenser in order to avoid erosion of the cylinder and piston, or turbine blade, the exact ratio varying somewhat with the properties of the chosen fluid; the only exception to this rule being a fluid with a negative slope in the vapor saturation region.

If one were to impose this 4 to 1 condition upon the expansion, then the order of efficiency for the Rankine cycle fluids would be different. What this means pictorially in the TS diagram is that the constant volume expansion of the cylinder towards the terminal end of its stroke must intersect the condenser temperature outside the saturation curve and not penetrate it. This places a limit on the vacuum which can be utilized at the condenser end of the stroke. During the expansion into a lower pressure, the condensation temperature is lowered and this moves the state point (1) which exits from the condenser, down the saturated liquid line, increasing the area to the left of the TS diagram and broadening it out. The broadening represents added heat input and the designer must evaluate this additional heat input against the additional work obtained by the widening. It could very well turn out that the effect is negative and the

efficiency decreases by too great an expansion into a vacuum. The practical design condition now is, given an upper temperature and an expansion ratio of 4 to 1, what should be the condenser temperature in order to maximize the work. This is a unique temperature for each fluid and the work is not maximized with ever increasing condenser temperature, contrary to what may be expected from a blind application of the Carnot principle. However, the expansion pressure cannot fall below 28 inches of mercury vacuum. Therefore from a practical point of view, the application of the Carnot principle may be unreliable. The condenser temperature, under this restraining condition, will vary with the fluid chosen and so each fluid must be examined to see if its maximum lower temperature coincides with the ambient cooling temperature available, the ultimate exhaust pressure falling where it may, but no more than 28 inches of mercury.

As an example, suppose we were to start a expansion with a temperature of 600F and a pressure of 100 psia. If we imposed a maximum lower pressure of 1 psi at the condenser after expansion, then the lower limit for steam would be 102F, which would be very generous to the condenser heat exchange designer if the ambient temperature were 60F. If we were to employ ethyl alcohol, then the lower temperature limit would be 72F. If ether were employed as a working cycle, the lower temperature at this pressure would be 28F. Detailed cycle calculations of vapor cycles under the restraint of a 1 psi final pressure leads to a rearrangement of efficiencies by the various fluids cited in Tables 1 and 2. For steam, the final temperature is 102F and the resulting efficiency is 0.27. For ethyl alcohol, the condensing temperature is 72F and the resulting efficiency is 0.29, thus #3 in Table 2. has displaced water and has become #1 in efficiency.

However, these efficiencies violate the condition that the expansion ratio be kept at a minimum of 4. If we impose this additional condition, require that the terminal pressure not fall below 28 inches of mercury vacuum and let the condenser temperature fall where it may, provided it does not fall below the ambient of 60F, then the following table results from this complexity of conditions.

Table 3. Efficiencies with expansion ratio - 4

Vapor	Discharge Temp	Final Pressure	Efficiency
Carbon Bisulphide	67F	-18"Hg	0.19
Chloroform	95	-18	0.18
Carbon Tetrachloride	95	-23	0.16
Water (steam)	109	-27	0.16
Alcohol	131	-19	0.15

First of all we see that all the efficiencies are drastically reduced as a result of this very practical condition, and therefore all prior results which claim high efficiencies that

have not considered these conditions must be held with reservation. What is demonstrated is that water is not necessarily the best fluid for use in a bottoming Rankine cycle which has extreme thermodynamic limits imposed upon it by metallurgy and ultimately cost. Also noteworthy is that the best efficiencies are obtained with a moderate amount of vacuum. As the limitation upon the expansion ratio is relaxed, the efficiencies in Table 3. would approach those of the unrestrained Rankine cycle as well as the Carnot cycle. However, in the real world of cost economics for heat exchangers, the Carnot criterion could be totally unreliable.

4.2 Chemical Limitations

Below we tabulate the freezing points and partial pressures as well as the triple point temperature and pressure. One additional constraint not yet cited is that the fluid cannot be expanded to a point where it will freeze up.

Table 4. Freezing and Triple Boiling Points of Vapors.

Vapor	Freezing*		Triple Point	
	Temp. (F)	Pressure (psi)	Temp.	Pressure
Steam	32	.08	705	3991
Carbon TetraChloride	-72	.64	320	107
Ethyl Alcohol	-177	.25	470	1072
Chloroform	32	1.2	320	169
Acetone	32	1.2	284	137
Carbon Bisulphide	32	2.5	302	176
Ether	32	3.7	248	149
Sulphur Dioxide	-20	5.0	104	90
Ammonia	-180	1.0	125	308
Carbon Dioxide	-140	3.0	86	1040

* Freezing points have an ambiguity of meaning because they depend upon the pressure and hence run a continuum of values. If a triple point exists where liquid, solid and vapor are in equilibrium, then that pressure and temperature is taken as the lowest freezing point. Otherwise, some relevant values at low pressure are used, at far below the limits for a working cycle condenser or else the partial pressure of saturated liquid at 32F has been used.

There are other substances which are immiscible mixtures of liquids such as sodium and potassium hydroxide. The alcohols, such Fluoronol are all immiscible in water. Alcohol and ether are two other immiscible combinations, and there are many others in nature. These mixtures provide an endless variety in which to maximize thermodynamic efficiency within the mechanical constraints of pressure and temperature cited above.

5 SECTION SUMMARY ON DESIGN PHILOSOPHY

The above discussion affirms the dictum, that the efficiency of any thermodynamic cycle is primarily a question of chemistry. A general restraint upon the thermodynamic design is that a fluid must be found which boils at the upper temperature and pressure as well as the lower temperature and pressure. The cogenerator supplies two levels of comparatively low temperature heat and one of high. The high temperature heat of the exhaust gas can be used to superheat the fluid to the vapor state, as far as permissible. The low temperature of the water jacket plus the portion of the exhaust which falls into the same temperature region can be used to convert the liquid to vapor. Because cylinder head pressures are similarly in the region of 150 psig, as are standard designs for off the shelf heat exchangers, this pressure is taken for the upper limit. The cooling jacket of the internal combustion engine is known to be able to withstand a temperature of 250F. Therefore the upper level transition from liquid to vapor is taken to occur at this temperature. Attempts will be made to employ direct evaporative cooling of the jacket by means of the organic fluid, thus avoiding both the cost and the entropy drop of intermediate heat exchangers. The low end temperature will be set at 120F to either manufacture domestic hot water or employ dry cooling with air. A low end pressure of 28 inches of mercury (vacuum) will be taken as the limiting pressure on the low side of the cycle. The design of compact heat exchangers can also benefit from an increase in the density of the organic fluid over that of water. For example, the density of saturated ammonia vapor, at room temperature is over 100 times that of water. This improves the condenser design for ammonia considerably, to a point where an ammonia system can be dry cooled with air, while a steam system would require a wet evaporative cooling tower. Such condenser designs are economically crucial. As an example every pound of steam that is condensed on hot side, say at 60F, requires water on the cold side to absorb this heat by a sensible increase in temperature. Since the latent heat of steam is 1000 BTU/lb. a design dT of 10 degrees in the cool side would require 100 pounds of cooling water for every pound of steam to be condensed. If ammonia were being condensed with a latent heat of 500 BTU/lb. then a water side evaporative condenser would require only a fraction of a pound of water, and so it could be replaced by the equivalent weight of air. Thus density also plays its role in the design of a bottoming cycle, all of these factors entering in, and requiring ingenuity for a correct balance into an optimum configuration.

6 INNOVATIVE THERMAL REFRIGERATION CYCLES

6.1 Jet Refrigeration

6.1.1 Circuit Configuration

The use of the jet in refrigeration cycles differs from its use as a pump or evacuator of secondary fluids by the fact that the condenser in a refrigeration system places a restraint upon the relative mass flows of injected and aspirated fluids. This means that one is not unlimited in the rate at which fluid which can be aspirated up from the lower cycle. This can be seen by the ideal representation of the jet refrigeration cycle depicted in the TS plane (see Figure 3.)

Figure 3. Jet Cooling Cycle in Temperature-Entropy Plane.

One sees two independent work cycles superimposed upon one another. (Actually, the aspiration trajectory is one of constant enthalpy, not constant entropy). The ideal COP is a ratio of the Carnot driving efficiency to the Carnot cooling (driven) efficiency and is written as follows;

$$\text{COP} < (1 - T_c/T_b)/(T_c/T_e - 1). \quad (6)$$

The inequality affirms that you can abstract no more cooling power than work input. This COP is an entropy ratio of driving to driven temperature difference divided by the source temperature and cannot be greater than unity. By equating the COP equal to unity, one obtains the limiting condenser temperature, in terms of the evaporator and boiler temperatures by which cooling can be obtained.

$$T_c = 2T_bT_e/[T_b + T_e] \quad (7)$$

If the condenser temperature is above this limit, then the jet

will run backwards, abstracting heat from the condenser and causing the evaporator to heat the environment. Similarly, too low a fluctuation in boiler temperature can blow the condenser fluid back down the nozzle and so heat the evaporator. If the condenser temperature can be stabilized by ambient conditions, a higher boiler temperature should induce a lower evaporator temperature, but this is not true always. Thus, sub-freezing temperatures could possibly be induced in a jet system by raising the boiler temperature sufficiently high, even accounting for inefficiencies in the nozzle, the diffuser, pipe friction, pump work and other irreversibilities and parasitic losses, provided the condenser temperature remains below that of formula 7, and the second law of thermodynamics will permit the cycle to function. If the cooling tower malfunctions and the condenser temperature rises, then the jet system will surely turn into a thermal heat pump and revert to the heating mode.

The formula also predicts very close temperature differences between the components of the system. For example, if the condenser temperature were fixed at 100F = 560R, then the production of 0F chilled water would require a boiler temperature of only 256F for a Carnot machine. Chilled water at 40F would require a Carnot boiler of only 176F. Thus high quality chilled water service can still be realized even in a system with thermal efficiency 50% of Carnot.

A schematic layout of the jet refrigeration system is shown below in Figure 4.

Figure 4. Schematic of Jet Component Refrigeration System.

6.1.2 General Considerations

The most evasive problem to solve in the design of the nozzle system is the eduction chamber after the refrigerant has been aspirated. The problems concern the momentum transfer from ejected fluid to aspirated fluid inducing turbulence that may heat the fluid rather than compress it. In addition, the boundary layer of the nozzle causes vortex flow which results in turbulence, expending the energy of the fluid into thermal components. A designer must minimize this. The next most serious problem is starting the ejector and the effects of transient loading by which the thermodynamic throat condition is violated and the ejector refuses to work. This problem must also be addressed by the designer. Generally one would try and construct a free surface in the induction chamber to alleviate friction.

6.1.3 Aerodynamic Considerations

The construction of the free surface and the consequent supersonic flow which results within the eduction chamber requires the construction of fluid characteristic surfaces using the technique of Riemann Invariants (Courant & Friedrichs). The following technique was employed to obtain that surface. The free surface exists because the Rankine-Hugoniot relations limit the amount by which thermodynamic variables can change across a discontinuity. In a nozzle with two dimensional symmetry, there exists an areal-Mach number relation which in turn determines and limits all the other thermodynamic changes which can occur across the discontinuity as well as down the eduction tube. If we were to chose as an example, an expansion ratio of critical throat area to mixing chamber area equal to 2. Then the following (Table 5) records the limits to pressure, density and Mach number as they evolve down the tube with the march of educted fluid for a simple molecule $k=1.3$.

M	P/P ₀	ρ/ρ_0	A _t /A
1.0	.53	.64	1.00
1.1	.47	.58	.99
1.2	.41	.53	.97
1.3	.36	.48	.94
1.4	.31	.44	.90
1.5	.27	.40	.85
1.6	.24	.36	.80
1.7	.20	.32	.75
1.8	.17	.29	.70
1.9	.15	.26	.64
2.0	.13	.23	.59
2.1	.11	.21	.54
2.2	.09	.18	.50

Table 5. Thermodynamic Parameters on an Expanding Air Surface.

One sees that, for an areal change of two, the pressure drops by over a factor of ten for an air nozzle.

We assume that the velocity profile across the chamber, at the throat, is uniformly at Mach One. Thereafter, the free surface will evolve with the above relationship between thermodynamic parameters. All that is now left is to construct the free surface as new boundary, and then plot the characteristic surfaces within the envelope that results in order to fix the spatial distribution of the thermodynamic parameters tabulated in the above table. The spatial velocity distribution emerges from the following relation,

$$(M^2-1)du/u = dA/A. \quad (6)$$

This relation may be used to construct the free surface by numerical means which is a streamline of the velocity field.

$$u_{n+1} = u_n + [u/(M^2-1)/A]_n dA \quad (7)$$

An additional term may be added to the above formula by differentiating equation 7. Then the step size may be chosen by a ratio test between the linear and quadratic contributions of the Taylor expansion. Once the velocity field is constructed and the velocity integrated to find the free surface, the ejection chamber will be fully enclosed by a data surface and the Riemann invariants can be constructed to create the entire internal field. Upon doing this, it was found that a primary set of rarefactive characteristics emanate from the critical section of the ejector. A second set of secondary rarefactive waves also emanate from the free surface, determined by the compression waves from the opposite wall. However, one can assume that these

compression waves originate as specular reflections from the axis of symmetry. The two rarefactions compound themselves until secondary reflections create secondary compression waves. The pressure in the eduction chamber is found to first drop to a minimum and then rises again. Thus, the aspiration tube must be placed at this minimal point of ejection chamber pressure. Although difficult to establish, the possibility of a stagnation point along the axis of the ejection chamber may exist, by which the aspiration efficiency will be a maximum. Unfortunately, the point of minimum pressure can not be wrapped neatly into a simple formula and even the computer is taxed each time a calculation is made. Such a calculation depends upon the chemistry of the fluid being ejected as well as the ratio of critical areas.

Of special importance will be the deflection angle at ejection, which will set the limits on how well the corner of ejector entrance can be rounded in order to levitate any Prandl-Meyer corner expansion which dissipates energy. The areal expansion ratio, which arbitrarily was chosen to be equal to 2, will actually be a function of the chemistry of the ejector fluid chosen. I will use the same fluid for injection as for refrigerant in order to avoid the aerodynamic complexities of two component fluid flow. Even, two phase flow of the same component will be ignored in this analysis. Actually, better mixing occurs when droplets of condensation occur upon aspiration, because the cross section for momentum transfer increases and the ejected gas can more easily accelerate the aspirated droplets. This tends to pacify turbulence which occurs during the mixing of two gases. Thus, as the refrigerant is aspirated into a region of lower pressure and temperature, it may condense into droplets before reaching the condenser. This should assist with the compression process because condensation, which can absorb the heat generated by turbulence, will cool the ejected gas. Thus, the assumption of both fluids remaining in the gaseous state during mixing is a conservative one.

6.1.4 Supersonic Computations

The following procedure for finding the free surface will be understood to be an intuitive engineering approach and not one that will satisfy all the demands of scientific rigor. This is basically a very difficult problem in applied mathematics which cannot be fully developed here.

1. Choose a differential areal step size appropriate for convergence, dA .

$$A_{n+1} = A_n + dA, \text{ say } dA = 1/200 \text{ if } A_n < 1 \quad (8)$$

2. Calculate the new Mach number from the implicit equation,

$$[A_n/A_0]^2 = [2/(k+1) \{1 + (k-1)M_n^2 + 1\}/2]^{(k+1)/(k-1)} / M_{n+1}^2. \quad (9)$$

$A_0=1, k=1.32 \text{ for } \text{NH}_3.$

(Note to take the upper branch of the curve)

3. Obtain the angle of the free surface streamline with respect to the Z axis of propagation.

$$\cos \theta_{n+1} = [M_{n+1}^2 - 1]^{1/2} / M_{n+1}. \quad (10)$$

4. Calculate the position coordinates (R_{n+1}, Z_{n+1}) of this point.

$$dR_{n+1} = dA_{n+1} / (2 R_{n+1}) \quad (11)$$

and,

$$dZ_{n+1} = \cot \theta \, dR_{n+1} \quad (12)$$

5. We now continue this trajectory of isolated points until the area A equals to the desired expansion ratio which we have taken equal to 2. (if we wish a 2 to 1 expansion).

6. Now calculate the velocity profile of the free surface. The velocity vector is parallel to the free surface.

$$dU = U \, dA / [A(M^2 - 1)] + F_2 \, dA^2, \quad (13)$$

in which F_2 is the second term in the Taylor expansion of the velocity. One can choose a variable step size (in No. 1 above) for convergence by the following criterion:

$$dA = [dU/dA] / [d^2U/dA^2] / 100. \quad (14)$$

7. We now can expand the velocity out along the free surface.

8.. The above accelerating velocity field is the perpendicular component in the direction of expansion of the cross sectional area. The speed is the Euclidian sum of the radial and axial velocities, the axial being the speed of sound which is obtained from the Mach number. Thus the speed of the fluid particles along the free surface is given by,

$$V = U [1 + 1/M^2]^{1/2}. \quad (15)$$

9. Similarly, the differential density may be expanded in a Taylor series in terms of the differential velocity and differential area by means of the continuity equation.

10. Similarly, the pressure can be found by the adiabatic law.

This completes the construction of the free surface. We now must use the characteristic equations to construct an internal network

of flow until we reach the axis. We wish to determine the pressure distribution along the axis of the eduction chamber.

Interior Construction

Now that we have the velocity field along the free surface, we can determine the two Mach lines (+) which form the sonic cone at each point along the streamline. They will make the following angle with respect to the tangent of the free surface.

$$G^{\pm} = \pm \cotan^{-1}[M^2-1]^{1/2} \quad (16)$$

The complement of this angle, translated by the slope of the streamline, represents the direction of the Mach line with respect to radial direction:

$$H^{\pm} = \pi/2 - G^{\pm} + \theta. \quad (\text{similar for } -) \quad (17)$$

We now must find the intersection of a forward Mach line at each point with the backward Mach line of the point ahead. Let this point be (R,Z). (one must adapt some notation for this collection of points which will fill the interior of the free surface),

$$Z_n - Z = \tan H_n^+ [R_n - R] \quad (18)$$

is the forward Mach line equation.

$$Z_{n+1} - Z = \tan H_{n+1}^- [P_{n+1} - R] \quad (19)$$

is the backward Mach line equation through the point ahead.

If we subtract the two equations, we obtain the value of R. Substituting this value of R into the above equations then gives the value of Z, and so thus the pair (R,Z) are determined inside the free surface.

We now must calculate the incremental distance along each Mach line to the point just determined. If the step size is small enough we can ignore the curvature in the differential distance.

$$S^+ = [(Z_{n+1} - Z)^2 - (R_{n+1} - R)^2]^{1/2} \quad (20)$$

$$S^- = [(Z_n - Z)^2 - (R_n - R)^2]^{1/2} \quad (21)$$

Then the Riemann invariants which are the angle of the stream line and the Prandle-Meyer (P) function which measures the Mach number implicitly are conserved along each of these lines. At the intersection, the two Riemann invariants represent two equations in two unknowns and the new velocity vector angle and Mach number (via the Prandle-Meyer function) is determined. This gives the following resulting expressions.

$$O_{ZR} = 0.5[(P_n - P_{n+1}) + (\theta_n + \theta_{n+1}) + (\sin H_n^+ \sin \theta_n S^+ / R_n - \sin H_{n+1}^- \sin \theta_{n+1} S^- / R_{n+1})] \quad (22)$$

$$P_{ZR} = 0.5[(P_n + P_{n+1}) + (\theta_n - \theta_{n+1}) + (\sin H_n^+ \sin \theta_n S^+ / R_n + \sin H_{n+1}^- \sin \theta_{n+1} S^- / R_{n+1})] \quad (23)$$

in which the epicycloidal formula is used to abstract the Mach number from the Prandle-Meyer function.

We continue this procedure inward until we meet with the Z axis of symmetry. Here, we must determine the pressure which is obtained from the momentum equation and the isentropic equation of state.

$$P + \rho V^2 = P_t + \rho_t V_t^2 \quad (24)$$

$$P \rho^k = P_t \rho_t^k, \quad (25)$$

in which the constants are determined at the throat. These are two equations in two unknowns which can be solved for both.

Once having determined the parameters at the axis, we are confronted with the wave which emanates from the opposite side. Because of symmetry, this wave is the reflection of the impinging wave in which the Mach line changes from forward to backward and visa-versa. This means that the rarefactive wave which struck the axis now is reflected as a compression. This compression raises the pressure above that which would be obtained by simple aerodynamics and so we must consider the secondary rarefactive waves which are created when this compression wave is reflected from the free surface which pacifies the compressive stress exerted on the original rarefaction. The bouncing back and forth between axis and free surface with alternate compression and rarefactive waves continues indefinitely down the eduction chamber. As the rarefactive wave moves up and meets the preceding points, the pressure it creates must be added (subtracted) to the pressure created by the secondary and tertiary waves of compression and rarefaction.

Figure 5. illustrates the results of one run of calculations through an ejector. The characteristic curves described above have been drawn in to illustrate the discussion. The point of minimum pressure on the axis was found to be at about 1.5 critical throat radii out into the ejection chamber. The pressure ratio dropped to 0.06, below the 0.1 value implied by table 5.

Figure 5. Characteristic Surfaces Within Jet Eduction Chamber.

6.1.5 Thermodynamic Considerations

Because the three operations of boiling, condensation and evaporation involve heat transfer by latent means, the ratio of mass between injection and aspiration can be determined from an energy balance of these three systems for the Carnot case. The result is;

$$x = (dL_c - dL_e) / (dL_b - dL_e) \quad (26)$$

in which x represents the fraction of total mass flow to boiler. Then $(1-x)$ is the fraction of the mass flow to the aspiration circuit. Since x is less than unity, the specific heat requirements of the boiler must be greater than the heat requirements of the condenser which in turn must exceed those of the evaporator. This condition can be ameliorated somewhat by regenerative precooling of the liquid going to the evaporator and preheating of the liquid going to the boiler as Figure 4. illustrates. We can calculate the coefficient of performance with formula 6, of section 6.1.1.

$$COP = [(1-x)dL_e] / [xdL_b] \quad (27)$$

It is clear that a small x (boiler mass flow fraction) enhances the COP. Resubstituting for x gives,

$$COP = [dL_b - dL_c] dL_e / [(dL_c - dL_e) dL_b] \quad (28)$$

The COP is enhanced by a high evaporator latent heat. Formula 27 must be multiplied by the nozzle efficiency to get a correct result. Moreover, it states that a high COP is obtained with a high evaporator latent heat compared with low boiler heat. According to the above discussion, this requires both a small mass fraction through the boiler as well as a low latent heat in the boiling stage. The requirement for enhanced COP are

reasonable because heat absorption by the evaporator is enhanced if both its latent heat and its mass fraction are high. The analysis is not restricted to latent heat, but must be interpreted to mean enthalpy differences which encompass sensible heat as well, along the saturation curves leading to boiler and condenser.

The next most important quantity to calculate is the areal ratio of throat to aspiration chamber (as opposed to ejection chamber). If we assume that the aspirated fluid arrives from the cold reservoir to the ejection chamber and mixes with the ejected fluid at the same pressure, then we may assume that the areal ratio is proportional to the mass flow fraction, since both refrigerant and driver fluid are chemically the same. The divergence of compressed fluid as it emanates from the condenser is throttled by the internal pressure of the cold reservoir which in turn, is modulated by the aspiration pressure created in the ejection chamber by the driving fluid. This is a self stabilizing system in which the areal ratios need not be critical, but only adequate to contain the flow. The compressed-condensed fluid which is sent to the boiler must be pumped up to boiler pressure and this is a tax upon the efficiency of the system which must be assessed. The boiler pumping rate must be adjusted so that the refrigeration rate is maximized. Therefore ideally, the most efficient system is one with a low injected mass flow in circulation as compared with high aspirated mass flow. Clearly, this also leads to high rates of refrigerant cooling.

The extent of the aspiration tube into the ejection chamber, although calculable under ideal conditions, is bound to vary from theoretical. The areal ratios of throat to ejection chamber are also expected to have deviations. However, strategically placed sensors can arrive at the correct positioning. The aspiration tube can be adjusted by pitot tube sensing to always be at the point of minimum pressure. The throat area can be adjusted in order assure secondary stage acceleration which occurs if the fluid attains Mach One at the throat of the nozzle before evacuation into the ejection chamber. The boiler pump must also be adjusted so as to maintain a constant level of refrigerant fluid in the cold reservoir. Therefore, all that need be done is to calculate the position and configuration of all these parameters with sufficient accuracy so that the necessary excursions required by the actuators be minimal and lead to stable system. This will be true if the system is sufficiently close to equilibrium.

6.1.6 Mechanical Considerations

In actuality, the jet eductor replaces the compressor in a common refrigeration circuit. Therefore, the compression ratio which occurs at the end of the mixing chamber is an important parameter of the system and a measure of the coefficient of performance.

The compression ratio which seems to result from calculations falls in the range of about 50, which is about 3 times the compression ratios calculated for standard ejectors with aspirated fluids flowing outside of the ejector. The compression ratio is inhibited by the turbulence that occurs when the two fluids, ejected and aspirated, mix in the eduction chamber. Turbulence causes energy to leak into thermal modes rather than flow work that directly benefits the compression ratio. Such turbulence can be inhibited by this design for two reasons. First of all, the introduction of refrigerant in the center of the nozzle rather than along the sides, reduces the velocity change of the aspirated fluid which would be zero at the wall in the classical system. Instead, the aspirated fluid also arrives into the eduction chamber with essentially a free surface, whereupon it may expand into the injected fluid with much more ease. As the injected fluid flows over the aspiration tube, there naturally occurs a stagnation zone in front of the aspiration tube, just as it would occur behind any blunt body moving in an air stream. In addition, stagnation is assisted by the characteristic composition of primary and secondary rarefactions described above. These rarefactions, in turn, are strengthened by the existence of a free eductive surface in the region of sudden expansion, the combination of all these aerodynamic effects leading to a minimum value of the stagnation pressure along the axis of the eduction tube. Such a minimum may be surmised from symmetry principles which imply that, only along the axis of the tube could such a minimum occur, since this is a singular and preferred line. Once the two fluids are mixed, the flow is expected to become subsonic as it meets the resistance of the condenser at the end of mixing chamber. In order for the flow to decrease in an orderly fashion and hence convert the organized motion of the fluid to compressive flow work, the terminal diffuser must diverge until the gas is compressed and then gradually converge to a smaller value in order to be rammed through the condenser. This ram effect will also reduce the pump work necessary to recirculate the driving fluid through the boiler. These items are not easily calculable from basic theory without some empirical experimentation.

6.1.7 Chemical Considerations

6.1.7.1 Toxicity

In 1984, The University of Paris - Institute for Pharmacology published its research on the toxicity of ammonia upon animals. The following is a summary of their results.

Effects of short term exposures (one hour) upon mice and rodents with exposures of 6000 ppm (human olfactory threshold = 25 ppm).

Death for concentrations between than 5000 & 6000 ppm. and above.

Signs of poisoning for concentrations of and below 5000 ppm,

with full recovery at the end of exposure and no long term after effects.

Long term exposure of animals to low concentrations of 500 ppm
Effect upon immunological tolerance.

Concentration 250 ppm upon mice

500 ppm upon rodents.

For exposures of 8 days. No noticeable effect.

For exposures of 160 days, noticeable consequential
susceptibility to bacterial infections.

In addition, the industrial and medical literature of both the United States and Europe, as well as the regulations of these countries, was reviewed for accidents attributed to ammonia. The conclusion was made by the French that the risks associated with this substance were acceptable for its use as a refrigerant and as a power plant working fluid in bottoming cycles.

Regulations in the United States, for many of the substances classified as "toxic" rather than poisonous, have been due to the fear of pungent odors which emanate from leaks in large crowded buildings. There have been discussions of panic in ammonia air conditioned structures which have leaked the substance in trace amounts. However, the pungent smell of these substances is an advantage. Furthermore, well below human olfactory sensitivity, bees and flies congregate about sources of ammonia, such as urine and excreta, and thus form natural sensors to faults in ammonia circuits. Ammonia is a biodegradable substance that forms a natural part of our environmental cycle. Substances such as it, carbon dioxide, ether, alcohol and other organics which are simple hydrocarbon chains form ecologically benign substitutes for the freons and other halogenated fixing agents used in the building and plastics industries. So long as the fluid is not poisonous, as compared with designation toxic, nor contain any halogens, is not a carcinogen and is composed solely of hydrogen, carbon, oxygen and nitrogen, such organics would be considered benign vapors for working cycles. One may place a further restriction upon biodegradable vapors and that is that they not be anesthetic or compelling in smell such as ether or alcohol, because any substance, even oxygen, inhaled in gross amounts is detrimental to a living organism. In this regard, vapors should preferably have a pungent and immediately detectable odor whose threshold lies well below any imaginable toxic limit. Ultimately, ammonia is not detrimental to the upper atmosphere and may well prove to be beneficial.

6.1.7.2 General Properties of Ammonia

For the jet cycle, ammonium (NH_3) was found to have attractive properties. Its latent heat of evaporation at low temperature is

high. The existence of a low triple point at 270F means that it will be easy to vaporize in the boiler with most of the heat going to sensible elevation of vapor temperature. Thus the crossing of the saturation curve can be limited across the boiler and maximized across the evaporator as the conditions for ideal jet refrigeration require. Its relevant properties are enumerated in Figure 6. below.

Figure 6. Enthalpies & Saturation Curve for Pure Ammonia

If we choose to make chilled water at 40F using a condenser temperature of 100F, then according to formula (2.12), the Carnot boiler temperature will be,

$$T_b = (560)^2/500 = 627R = 167F \quad (29)$$

The temperature difference between boiler and condenser is 67F for this Carnot cycle (COP=1). If we can hope to obtain a COP of 0.5 then the condenser-boiler temperature difference would double to 134F and the boiler would have to rise to 234F.

Formula (26) for the bifurcation of fluid leaving the condenser is an exact formula, not limited to a Carnot engine. However, one must fully account for the heat loads of all components, especially radiation losses. We must assess the boiler with preheating the liquid along liquidus as it leaves the condenser to the boiler. This is an additional 189.25 BTU/# raising the boiler load to 440.94 BTU/#. We assume that the evaporator load which rises up the vapor curve requires an additional 19.05 BTU/hr, raising the evaporator load to 549.34 BTU/#. We assume that the condenser is limited to a dissipation rate of 482.55 BTU/# so that all transitory heat requirements are made up internally at the expense of system performance. (These can be ameliorated somewhat with recuperation, but this is ignored in the interest of conservatism). Using these figures, the boiler mass fraction becomes,

$$x = [544.85-482.55]/[544.85-440.94] = 0.60 \quad (30)$$

Then the fraction which goes to the evaporator becomes,

$$(1-x) = 0.40. \quad (31)$$

Fluid flow is almost evenly divided between boiler and evaporator in order to carry their heat loads, and this not only includes enthalpy across the trajectories, but up and down the saturation curve itself which normally represent heat losses in the absence of recuperation. However, the heat load of the evaporator is not the cooling load that it can address because the enthalpy losses up and down the saturation line which leads to the evaporative crossing of the curve is a heat loss that must be subtracted from the condenser latent heat which is expected to chill the load. This loss is some 68 BTU/#, and so the COP is assessed as the following,

$$\text{COP} = 0.4 \times (535.04 - 62.3) / (0.6 \times 440.98) = 0.71. \quad (32)$$

In obtaining the COP of 0.71 above, we assumed a boiler temperature based upon a COP of 0.5. This result represents the start of an iterative procedure which converges, each value along the way being less conservative than the former.

Any calculations such as the one above must be checked against the second law of thermodynamics to insure that the system will cool and not heat. This is determined by a condition on the mass fraction which utilizes entropies in a formula analogous to 26 which utilized enthalpies, i.e.,

$$X = [dS_e - dS_c] / [dS_e - dS_b] \quad (33)$$

The mass fraction obtained from formula 3.14 must be greater than the mass fraction obtained from formula 3.11., i.e. $X > x$. In the above example we have,

$$\begin{aligned} dS_e &= 544.85/505 = 1.078910891 \\ dS_c &= 482.55/560 = 0.861696429 \\ dS_b &= 440.94/694 = 0.635360231 \end{aligned} \quad (34)$$

$$X = 0.49$$

This value of 0.49 is smaller than our associated value of $x=0.6$ calculated by the enthalpy method and so we have violated the second law of thermodynamics with these state points. This means that we must lower the boiler temperature to prevent the system from running backwards in the heating mode. This brings us to the second stage of iterations. We then choose a mean x of 0.55. Placing this value into the formula for x and recalculating a value for the enthalpy of saturated vapor leaving the boiler, we come out with a value of 498.7 BTU/Lb. According to the Ammonia tables of Kalina, this implies a boiler temperature of 227.43F.

We now check this again against the second law of thermodynamics. We must revise the boiler change of entropy.

$$dS_b = 498.7/(227.43+460) = 0.725456733$$

Hence our new value of X becomes,

$$X = [1.078910891 - 0.861696429]/[1.078910891 - 0.725456733] = 0.61.$$

Now we have satisfied the entropy condition that $x < X$, and we can proceed with some confidence. The latitude we have to entropy loss, principally nozzle inefficiency is a $dx = 0.06$, which amounts to about 10%. Thus the mechanical efficiency of the nozzle must be greater than 90% in order for these parameters to work.

6.1.7.3 Ammonia-Water Mixtures

Because ammonia is immiscible in water, the blend can be used as a working fluid. To investigate the consequences of such a binary mixture I have chosen to examine a 50-50 mixture of NH3 and H2O. Figure 6. shows the significant enthalpies associated with such a mixture along the saturation curve.

Figure 6. Saturation Curve, Enthalpies of 50/50 mixture NH3/H2O

From the above curve one can calculate the bifurcation of mass flow and the COP.

$$x = [935.27 - 602.93]/[1044 - 602.93] = 0.75 \quad (37)$$

$$COP = [0.25 \times 602.93]/[0.75 \times 1044] = 0.19 \quad (38)$$

We can now see that any mixture of water in ammonia will depreciate the performance. This is obvious since steam is known to be an inferior fluid for jet refrigeration, its addition would hardly be expected to improve a system. Therefore, we are led to endorse the use of pure ammonia as a working fluid for jet refrigeration. The only difficulty it could present is in the high pressures it creates. Since the temperature of the boiler is moderate, and the jet structure has simple cylindrical symmetry,

the incorporation of structural integrity to withstand high pressures should not be an insurmountable problem.

6.1.7.4 Transport Properties

The pressures at the saturation crossings of Figure 6 were found to be as follows,

$$\begin{aligned} P_b &= 1073 \text{ psia at } 234\text{F} \\ P_c &= 203 \text{ psia at } 100\text{F} \\ P_e &= 56 \text{ psia at } 40\text{F} \\ \text{Ratio of specific heats} &= 1.3 \end{aligned} \quad (39)$$

The specific volumes of ammonia at 234F were found to be as follows,

$$\begin{aligned} v_1 &= 0.03749 \text{ ft}^3/\# \text{ saturated liquid} \\ v_v &= 2.736 \text{ ft}^3/\# \text{ saturated vapor} \end{aligned} \quad (40)$$

The specific volumes of ammonia at 40F were found to be as follows,

$$\begin{aligned} v_1 &= 0.02533 \text{ ft}^3/\# \text{ saturated liquid} \\ v_v &= 3.971 \text{ ft}^3/\# \text{ saturated vapor} \end{aligned} \quad (41)$$

6.1.7.5 Jet Nozzle Implementation

We are now in a position to calculate an actual operating system. We shall consider a 50kw cogenerator. This capacity can produce 300,000 BTU/hr at a temperature of 234F. The latent heat of boiling being 251 #/hr requires that an amount of $300,000/251.7 = 1195$ #/hr be circulated through the boiler. However, this is only a fraction of 0.55 of the fluid which circulates through the condenser. The total fluid in the system must be equal to $1195/0.55 = 2173$ #/hr. The component of mass which circulates through the evaporator must then be $2173 \times 0.45 = 978$ #/hr.

We can now calculate the critical throat area of the nozzle injection and aspiration chamber. If the boiler is at 687R and at 1145 psia, then choked flow at the throat will create a pressure of,

$$P_c = 1145 (2/2.3)^{1.3/3} = 624.85 \text{ psia} \quad (42)$$

The critical velocity will be,

$$V_c = 223.7 [c_n(560.75 - 512.23)]^{1/2} = 1543 \text{ ft/sec}, \quad (43)$$

using an injector nozzle efficiency of 98%. The injector throat area may be calculated from the continuity equation,

$$A_t = [1192 \times 1.82] / [1543 \times 3600] = .00039 \text{ ft}^2 = .0445 \text{ in}^2 \quad (44)$$

This area is equivalent to a circle of 0.119 inches in radius.

We must now calculate the pressure differential in the eductor which would be necessary to aspirate the correct amount of 899#/hr of refrigerant. This mass flow depends upon the differential pressure between the vapor partial pressure in the cooling reservoir (56 psia) and the pressure created in the eductor. The necessary pressure must come from the momentum equation. Since,

$$m = VA/v, \quad (45)$$

we obtain a pressure differential of,

$$dP = (m/A)^2 v/g, \quad (46)$$

in which g is the acceleration of gravity. If we chose a pressure differential of 10 psi, then with a specific volume of ammonia vapor cited above of 40F, we obtain an area of aspiration into the eduction chamber equal to 0.3286 in². This is equivalent to a circular area of approximately 5/8". Since the vapor pressure at the desired evaporator temperature of 40F is equal to 56 psia, we must obtain a pressure of 46 psia in the eduction chamber in order to aspirate the quantity of refrigerent sufficient to reach the design temperature under the design load conditions. Because the boiler pressure was found to be 1145 psia, a proportional drop of about 25 is required to reach 46 psia. We now must turn to the areal Mach number relation (Leipmann & Roshko) to give us the necessary areal expansion which would attain such a drop in pressure.

$$A_t/A = [1 - x^{n-1/n}]^{1/2} x^{1/2} / [((n-1)/n)^{1/2} (2^{(n-1)}(n+1)/(n-1)^{1/2})]^{1/2} \quad (47)$$

in which x , is the pressure reduction (.04).

Accordingly, we find that the correct areal increase is equal to 4.5. If the throat area is taken at the calculated value of 0.0445 in², then the eduction chamber must expand to 0.20 in², essentially by a factor of 4. This is equivalent to doubling the circular area of the injector from approximately 1/8" inch in diameter to 1/4" inches for the central aspirator.

6.1.7.6 Jet Cycle Implementation

If the above values are now used, then with a flow rate of 899#/hr thru the evaporator with a net available latent heat of (536-68) = 468 BTU/# would yield a chilling rate of 420,700 BTU/hr. which is contradictory to a COP of 0.6 and an input of only 300,000 BTU/hr. Such a contradiction arises because we have exaggerated the boiler heat requirements by taxing it with heat loads of approach in the amount of some 186.5 BTU/hr. Because the boiler load is coupled to the evaporator load by means of formula 26, this raises the flow rate thru the evaporator as well. This contradiction represents the start of a secondary iterative procedure. Initially we stated with the boiler's demands and arrived at the evaporator loading as a consequence. We now may start with the evaporator loading and calculate the boiler demands as a result.

If the COP is taken at 0.6 and the evaporator load is taken at 468 BTU/#, then an input of 300,000 BTU/# into the boiler results in a chilling load of 180,000 BTU/# or 15 tons of refrigeration. If the heat along the saturation lines is recuperated, then the evaporator load becomes 536 BTU/# leading to 17 tons of refrigeration. If we assume that half the heat can be recuperated from the evaporator, then we should obtain some 16 tons of refrigeration. This is an evaporator load of 502 BTU/#. Implies a circulation rate of fluid through the evaporator of $180,000/502 = 359$ #/hr. The flow rate through the condenser would be $359/.43 = 834$ #/hr. and the boiler flow rate would be 475 #/hr. This is less than half the amount calculated originally, under the most severe assumptions to the boiler mass flow. It now becomes a matter of design as to how much heat can be recuperated so as to minimize losses and so obtain a correct chilling rate.

Paradoxically, the more we tax the boiler, the more the concomitant rate of flow through the evaporator is exaggerated, and so two iterative calculations appear in this procedure, one within the other. Ultimately, we will be reduced to empiricism when it comes time to design the heat exchangers and the refrigeration circuits. In the laboratory, we can build the machinery to almost any ideal configuration and so observe the accuracy of our method. At this stage of our illustration, the average values of all components in this system can be taken to yield the following specifications for a jet system driven by a 50 kw cogenerator. The actual values will depend upon the materials and equipment we have at hand for construction of the actual system.

The average specifications for the ammonia jet system are summarized in Table 6.

Table 6.
Parameters for Jet Refrigeration System via 50 kw Cogenerator

Working Fluid - NH_3 (desiccated)
Average Boiler Flow Rate - 833 #/hr.
Average Evaporator Flow Rate - 629 #/hr.
Average Condenser Flow Rate - 1462 #/hr.
Average Chilling rate of evaporator - 16 tons
Average COP of the chiller system - 0.6
Chiller temperature - 40F
Condensing Temperature - 100F
Boiler Temperature - 234F
Average Pump Work - 1 HP
Heat Input to Boiler - 300,000 BTU/hr.
Boiler Pressure - 1145 psia
Condenser Pressure - 206 psia
Evaporator Pressure - 56 psia

Please note that the above are design specifications in which certain parameters may be purposely exaggerated in order to permit a latitude of uncertainty for actual operating conditions. Thus the parameters could be theroretically inconsistent to this degree.

6.1.8 Jet General Conclusions

The attainment of a COP equal to 0.6 by a jet system would place it on equal footing with a single stage absorption system. The advantages of the jet would be low capital cost, estimated at about \$25/ton of refrigeration and the use of a more thermodynamically stable and inexpensive working fluid -- ammonia. However, because of the higher pressures involved in a jet ammonia system, perhaps a factor of 100, the associated piping and heat exchangers would have to be more substantial and therefore, more expensive by a decade. Counteracting this is the excellent heat transfer properties of ammonia which would lead to less heat transfer surface. In fact, the cooling of ammonia, many times could be done by dry air rather than a wet evaporative water tower. This would cut down costs of the cooling tower by more than a decade. One of the advantages (or disadvantages) of dry cooling is that its efficiency depends upon the outdoor temperature, more so than with a wet cooled tower. Ammonia, owing to its high latent heat, has a high conversion efficiency from thermal to mechanical. Because of its volatility, it can be converted from liquid to vapor at low temperatures and therefore a more full utilization of the entropy of a heat source is possible with the jet ammonia system. This implies, that for the same efficiency, ammonia can operate at a lower top temperature. Thus, a lesser amount of waste heat occurs in any thermodynamic system that can employ ammonia. The problem with its employment in other systems is that high pressure can only be contained in circuits with small volume to surface area. This is obvious from

the analysis of mechanical hoop stresses in a pipe or containment tank. Such small ratios are possible in the nozzle because of its simple cylindrical design which relies upon the length of the configuration and not the cross-section. In a turbine, a certain amount of blade area, or cross-section must exist to compress a particular quantity of gas. Not so with the jet, in which a variety of configurations are available. Given the prolific availability of low temperature waste heat in a cogenerator, every effort to employ ammonia for thermal mechanical recovery should be made. The stumbling block has been structural integrity against the high consequent pressures. It appears that only the jet can qualify for employment of such an advantageous fluid.

6.2 Bottom Cycle Turbine-Compressor

6.2.1 Performance with Freon 114

Although it is the intention of this project to use a benign working fluid, we will analyze Freon in order to be able to make a comparison of efficiency. Independently of their environmental detriments, Freons are not good candidates for turbines because they are operationally limited to temperatures below 350F, above which they decompose. We have chosen Freon 114 to illustrate the effectiveness of this technology and the manner in which the system may be integrated into the thermal characteristics of an internal combustion engine.

The upper temperature at which superheated Freon vapor may enter the turbine must therefore be limited to 350F. The condenser temperature will be held at above 100F in order to be able to manufacture tepid domestic hot water. Because the water jacket temperature of the internal combustion engine is limited to below 230F within very close tolerances, the water jacket heat must serve to evaporate the fluid at constant temperature into the gaseous state, leaving the exhaust gas to superheat the working fluid into expandable vapor. Therefore, a temperature of about 195F is chosen for the vaporization process. The pressure at which this will occur is determined by the expense of heat exchanger manufacture. Since current off-the-shelf components are tested at about 150 psi, we shall use this as a design pressure for the water jacket heat conversion. This means that we must find a fluid with chemistry that has a saturation crossing at less than 150 psi and about 195F. These temperature limits impose constraints upon turbine operations, limiting its efficiency seriously. Nonetheless, extrapolation can be made to more favorable conditions, once a more preferable fluid is found.

In Figure 8. below, I have drawn a schematic saturation curve for Freon 114, the only Freon that could be found which had the closest saturation crossing at 193F and 150 psi. The diagram itself, immediately gives the possible efficiency of the system

without any further considerations.

Figure 8. Saturation Curve for R114

The theoretical efficiency of the circuit is given by the enthalpy differences which measure the work of expansion over the heat input.

$$\text{Efficiency} = \frac{130.84 - 118.6}{130.84 - 34.8} = 13.4\% \quad (48)$$

The above efficiency does not include recuperation of heat from the final descent of the cycle in the superheated region to the saturation curve that marks the exit trajectory of the turbine. If we assume that we can capture the heat down this superheat curve until the temperature of the exiting gas reaches 137F, some 15 degrees above condenser temperature, then the amount of enthalpy which is recuperated will be 25.52 BTU/#. If in addition, we can then preheat the liquid entering the boiler from the condenser, in the amount of 80%, then the heat regenerated will be 23.68 BTU/#. With these amounts of heat recovery, the efficiency will improve to the following value;

$$\text{Efficiency} = [130.84 - 118.6] / [130.84 - 34.8 - 23.68] = 17\% \quad (49)$$

If the cycle efficiency is taken at 70%, then one may expect overall efficiencies between 9.4% & 12% under these very conservative assumptions concerning the chemistry of the fluid chosen.

6.2.2 Performance with Fluoronol 85

We have decided to recommend an alcohol based working fluid which can be mixed with water in any proportion to obtain desirable properties. This fluid has been utilized in the past in many other bottoming cycle projects (Thermo Electron Corp., Argonne National Laboratory) and it would form a prototype system for

direct comparison with other systems utilized in the past. In addition, this fluid appears to outperform F114 in cycle efficiency. In Figure 3.20, we have drawn a saturation curve for F-85 (15% H₂O & 85% Trifluoroethanol). One notices that the pressures are very modest with this fluid, suggesting that it could be boiled directly in the engine cooling jacket and thus eliminate the cost of one heat exchanger. Secondly, even though the exit temperature from the turbine is high, this fluid gives better efficiency than the expansion of F114.

Figure 9. Enthalpies of F-85 for Turbine Bottoming Cycle

The efficiency of the above cycle is,

$$\text{Eff} = [298.84 - 256.43] / [298.84 - 27.05] = 15.61\%. \quad (50)$$

Two percentage points are obtained with TFE, almost 10% improvement over the Freon turbine. If in addition, we can recuperate 80% of the heat in the descending super heat region which extends from turbine exit to condenser inlet, we obtain the following efficiency.

$$\text{Eff} = 42.44 / [298.84 - 27.05 - 0.8(256.43 - 235.63)] = 18\%. \quad (51)$$

To this we must include a turbine efficiency which we take at 80%, so that we are left with a systems efficiency between 12.5 & 14 %. Coupled to a screw compressor with COP = 6.5, we obtain an overall COP between 0.8 & 0.91. This will yield between 20 & 23 tons of refrigeration for a 50kw cogenerator.

6.2.3 Turbine-Cogenerator Integration

We take for an example, a 50kW cogeneration system. The total heat available from such an internal combustion engine will be 300,000 BTU/hr. The consequent output under the above cited conditions would be between 11 and 14 horsepower. Centrifugal compressors give one ton of refrigeration for each horsepower output and so this would imply 11 to 14 tons of thermal refrigeration from a 50 kw cogenerator. Such a rule of thumb assumes a COP of 4.7 for the centrifugal compressor. Current

screw compressors are quoted with COP's of 6.5. This would increase the refrigeration to between 15 and 19 tons of refrigeration. An absorption machine with a COP of unity would generate some 25 tons of refrigeration. However, an absorption machine may be much more expensive and occupy considerably more space. Furthermore, the bottom cycle turbine could reach lower temperatures, below those obtainable by an absorption machine. In addition, the compact size permits the turbine to be incorporated into the module of the cogenerator with considerably more ease.

However, the compromise of efficiency is not entirely so gross. The employment of a different working fluid which attains a higher temperature at lower pressure would improve the performance of the bottom cycle turbine considerably in its capacity to generate cooling power. On the other hand, the development of a triple effect absorption chiller with COP close to 2, would render marginal a bottom cycle turbine, even with proportionally higher efficiency obtained from a higher operating temperature, except that the triple effect machine can use only the top portion of the exhaust heat. A 50kw machine could supply only 132,000 BTU/hr. With a COP of 1.8, this would give 20 tons of refrigeration alone from the upper temperature heat of an exhaust in a triple effect machine. This would leave the lower temperature exhaust gas and the water jacket for use in a bottoming cycle. Such low temperatures would then be able to utilize ammonia for a working fluid because of the lower pressures associated with that fluid at these lower temperatures. This is an alternative that will be kept in mind.

6.2.4 Turbo-Compressor: General Conclusions

The advantages of the bottoming cycle turbine were principally that the thermal energy of a cogenerator could be used to drive a high COP vapor compressor thru the intermediate of an expansion turbine which utilized an organic Rankine fluid. In a US patent by (P. R. Trumpler, 1980) on a similar device, he takes notice of the compactness of a compressor that is at high RPM and the ability to use air directly as a refrigerant, thus obviating all environmental objections to polluting refrigerants by using the life sustaining fluid itself for the refrigerant -- air. The vapor compressor, having a high COP (as much as 6.5) would incorporate a known technology previously consumptive of electric energy, but now would consume thermal instead. Even if the efficiency of the turbine were low, this could be compensated by the high efficiency of the vapor compressor. Thus a turbine of 15% efficiency coupled to a vapor compressor with COP equal to 6.5 would yield a system with overall COP of unity. The compactness of the turbine driven vapor compressor would make it an attractive feature for packaging. The existence of a network of service and distribution centers thruout the world would ease implementation and maintenance. A machine with such

characteristics, when attached to a 50 kw cogenerator would yield 25 tons of refrigeration, a very attractive ratio of electricity production to cooling power. By comparison with the jet system, one would trade a rotating machine which would require attention with that of a static machine with no moving parts.

The bottom cycle turbine can make use of a high efficiency, high velocity, vapor turbo-compressor with a possible COP of 7.0. Thus, if the bottom cycle turbine could be at least 10% efficient it would compete with a single stage absorption chiller in productive capacity. Turning at higher speed than normal compressors, it would more compact, and hence, the installation costs would be less; however, maintenance costs would be higher in such a high rotating mechanical system over the quiescent absorber. Unable to use ammonia at the high exhaust temperatures available, the bottom cycle turbine would employ, in this recommendation, the fluid trifluoroethanol (Fluoronol) which is an alcohol and currently environmentally acceptable. When integrated within the configuration of a cogenerator, the small equipment size of the turbine system is highly desirable because absorption systems are so large by comparison that their use is disqualified in most urban retrofits or even in high density suburban areas that place a premium upon real estate. Because startup and shutdown of absorption systems is a long and inefficient process, the output of an absorption chiller can not be modulated to daily and hourly variations in the weather patterns as easily as a vapor compressor. Thus, an absorption chiller for the most part, should operate at full capacity regardless of the heat load. This means that the absorption chiller operates with such inefficiency at reduced capacity that it is best left to run wide open, regardless of ambient conditions of the moment. The turbine, on the other hand, can be coupled to a screw compressor on one axis and a smaller generator on the other, delivering power to both or either system as the load demands; however, this would necessitate a slower RPM with a sacrifice in efficiency to the compressor. If this is satisfactory, then the turbine can be used to supplement the electrical output of the cogenerator raising its efficiency to above 40%, or else it can deliver its shaft power to a vapor compressor or divide its output among the two as the climatic conditions of the minute require.

7 Cogenerator Heat Exchanger Configuration

The problem of fully utilizing both the water jacket heat and the exhaust gas requires three stages of heat transfer. Ideally, the water jacket, being at essentially constant temperature, can only transfer its heat during a vaporizing process. That temperature is denoted by T_{vap} . The rise of the cycle in the superheat region, should ideally be carried out by the high temperature portion of the exhaust gas. Regeneration

from the lower return cycle in the superheat region may be carried out. This requires that the exhaust gas be divided up into three temperature regions; The high which creates the superheat, the intermediate which runs parallel with the regeneration and water jacket and the low temperature which takes the fluid up from the condenser to boiler temperature. Two isotherms are drawn in the superheat region, one emanating from the vaporization temperature and one emanating from the exhaust temperature of the turbine. The intersection of the isotherms represents the input and output conditions for a counter flow heat exchanger by which heat can be transferred from the lower superheat trajectory to the upper. The area enclosed by these isotherms and the vapor superheat trajectories is the amount of heat available to be regenerated. However, because there must be a temperature difference of generally 20F between the two fluids entering or leaving a heat exchanger, a smaller area, inscribed between these isotherms, is actually available for heat transfer. The temperature of approach between the upper and lower curves should differ by 20F. The partitioned area, which is thus lost, represents energy or entropy sacrificed to the transfer process.

8 Cogenerator Cycle Consolidation

The efficient use of heat exchanger circuits to fully utilize the thermal energy from an internal combustion engine, is not peculiar to the turbine cycle, but applies equally well to the jet refrigeration cycle. However, in the jet, an evaporator reservoir would have to be built specifically for that purpose and which would find no use in the turbine circuit. Except for the brief remark at the end of section 6.2.3, little has been said about the integration of either of these devices with an absorption chiller because that would be systems integration, a subject not within the present scope of this report. This report, at this stage, concerns itself with the feasibility of each of the two above described components.

9 Comparative Economics

9.1 Cost Calculation Assumptions: Table 7.

Cost of Electricity	\$0.10/Kwh
Cost of Natural Gas	\$0.35/Therm
Cooling Season	1500 hours
Period of Operation	20 years
Interest discount rate	10%/annum
Monthly discount	.013215074
Future/Present Value	2.707

Vapor Compressor Lifetime	5 years*
Vapor Capital Costs	\$350/Ton
Vapor Maintenance	\$70/Ton/yr.
One Ton - One Kwh	(COP=3.5)

Absorption Lifetime	20 years
Absorber Capital Costs	\$200/Ton
Absorber Maintenance	\$25/Ton/yr.
	(COP=0.7)

Jet Lifetime	20 years
Capital Costs	\$5/Ton
Maintenance Costs	\$25/Ton/yr.
	(COP=0.5)

Turbine Lifetime	5 years*
Capital Costs	\$700/Ton
Turbine Maintenance	\$70/ton
	(COP=7)

* Although centrifugal compressors can be run as long as 20 years, optimal performance requires periodic repairs over a 5 year period equal to the cost of the compressor. On the other hand, absorption systems can be run for much longer period of times, perhaps a half century. In these figures we include updating to keep up with changes in technology.

9.2 Electric Vapor Compression

Cost Summary over a 20 year period.

Vapor Capital expenditures	\$947/Ton
Maintenance	\$5297/Ton
Operating Costs	\$11351/Ton
Total Vapor Costs Summary	\$17,595/Ton/20 yr. period

9.3 Jet Refrigeration

Cost Summary over a 20 year period.

Jet Refrigeration Capital Expenditure	\$54/Ton
Maintenance	\$1892/Ton
Operating Costs	\$9535/Ton
Total Jet Refrigeration Costs	\$11,481/Ton/20 yr.

9.4 Turbo-compressor

Cost Summary Over 20 Year Period.

Bottom Turbine Capital Expenditure	\$128/Ton
Maintenance	\$10594/Ton
Operating Costs	\$1945/Ton
Total Costs for Thermal Turbine	\$12,668/Ton/20 yr. period

9.5 Electric Vapor Compression vs. Thermal Refrigeration

There are three economic factors which determine the life time costs of operating an air conditioning system. The first is the cost of fuel -- electricity or gas. The second is the installation costs and the third is the maintenance. Because the cost of summer electricity is higher when compared to natural gas, perhaps a factor of 10 on a caloric basis, electric driven vapor compression is 50% more expensive to operate than either of the thermal systems of jet refrigeration or bottom cycle turbine. The advantage becomes much higher if the cost of heat can be avoided by using waste heat or cogenerated heat. In this model we have paid for the heat at a rate of \$0.35/ therm, while electricity was assumed to sell for \$0.12/Kwh = \$3.52/therm. The reason electricity can hope to be competitive was that the vapor compressor has a higher coefficient of performance (COP) than the classical single stage absorption chiller, by a factor of 5. However, the bottom cycle turbine can attain a COP of 7 because it can operate at higher RPM (40,000 to 100,00 against 1800 RPM). This high rotational speed lowers the capital cost and operating cost of the bottom cycle turbine, by making it smaller, but raises the maintenance. The jet system is very simple, probably could be made of plastic and held in the palm of the hand, but it has a poor COP. Capital costs and maintenance costs of a jet system are non existent when compared with operating costs. Thus both thermal systems are preferable to electric vapor compression and have approximately equal costs over a 20 year lifetime. The jet has lower initial costs but higher operating costs. The bottom cycle turbine has higher initial costs but lower operating costs. Thus, the choice between the two thermal systems depends upon whether one wishes to invest up front in capital costs and so be free of operating costs or visa-versa. In the event that waste heat is available then the jet system is to be preferred, as it

has always been under these conditions, in spite of its low coefficient of performance. However, from an ecological point of view, if waste heat were not available and fuel had to be burned to operate a jet system, then bottom cycle turbine would be the environmental preferred machine.

9.6 Environmental Impacts

We have concluded that thermal chilling is economically preferable to electric vapor compression, especially if the thermal energy is waste heat. However, if fuel has to be consumed in order to operate the thermal chilling device, then from the point of view of ecology, the COP of the system would dominate the choice. Thus, the bottom cycle turbine is to be preferred over the jet because the COP is greater by a factor of 14.

Currently, there is being researched in Canada, a bottom cycle turbine of only 3% efficiency, which operates from very low temperature heat (195F inlet water & 60F condenser). Although the efficiency of this machine may appear to be low, the ratio of electricity usage compared with heat is also low in domestic and commercial applications which are the largest market segment for energy electrical consumption. A large family residence would demand less than a kilowatt of power, and so the return water to a boiler from heating an average building or the domestic hot water consumption of apartments containing a dozen units would have a substantial portion of the building's electricity bill canceled with the use of even such a small waste heat bottom cycle turbine. The Canadian turbine was designed to operate with a central urban nuclear reactor which, in the interest of safety, would generate low temperature hot water rather than steam. This hot water would be circulated in a district to each building which individually would have one of these small bottom cycle turbines. It is not clear that this eliminates the volatility of nuclear catastrophes, and so would justify placing them in the centers of urban population. However, cogenerated waste heat with the attainment of 15% from a bottom cycle turbine such as the one described, would be a substantial advance of the technology as well as obviating the risk associated with other options such as nuclear energy or central power plant generation.

10. ACKNOWLEDGEMENTS

The author is grateful to Dr. John Andrews, Dr. Thos. Butcher and Mr. Michael Piraino who spent much time in reviewing this work and upon whom the author will rely upon for increased participation in the experimental program which will follow this research. In addition, thanks go to Clark Industries of Springdale Arkansas, Mr. Mike Kramer, with whom we had consulted and who has provided us with the high pressure pump to be used in the forthcoming experiments.

11 BIBLIOGRAPHY

1988 May, "Ozone Depletion At The Poles: The Hole Story Emerges", Physics Today, p.17-21

1974 D.R. Miller, "Optimum Working Fluids for Automotive Rankine Engines". Sandia- ERDA, APTD Rep.#s. 1563/4/5/6. NTIS-SLA-74-0132

1977 S. Luchter and R.A. Renner. ERDA 77-54. UC-96 "An Assessment of the Technology of Rankine Engines for Automobiles"
Chemical Engineering Handbook

1972 D.R. Miller & Q.E. Thompson "Optimum Rankine-Cycle Fluids for Automotive Use", Monsanto Research Corp.

1976 Aerojet Corp. "Organic Rankine-Cycle Automotive Propulsion System Turbine Expander", Final Report, No. SAN 1096-1

1986 Sept, M.L. Jain, J.C. Demirgian and R.L. Cole. "Organic Rankine-Cycle Power Systems. Working Fluids Study: Topical Report No. 1 -Fluorinol 85. Argonne National Lab. ANL/CNSV-TN-182.

1986 February ,Kalina Cycle Power Systems as a Bottoming Cycle for Combined-Cycle Systems. Houston Texas, Section 7.
Thermodynamic Properties of an Ammonia-Water Mixture.

1978, L. Haar & J S Gallagher, J. Phys. Chem. Ref. Data, V.7, #3.

Halocarbon Products Corporation, 82 Burlews Court, Hackensack NJ, 07601. Thermodynamic Properties of Fluorinols.

ASHRAE Handbook & Product Directory, 1977 Fundamentals, ASHRAE 345 E. 47th. St. NYC, 10017.

Handbook of Thermodynamic Tables & Charts, by Kuzman Raznjevic. Mc Graw Hill

Journal of Physical and Chemical Reference Data, Volume 7, 1978. Ed: David R. Lide, Jr.

NBS bulletin No. 112. Tables of Thermodynamic Properties of Ammonia, Superintendent of Documents, U S Government Printing Office, Washington DC.

Chemical Engineers Handbook, Perry & Chilton, 5th. Ed., Mc Graw Hill, NY

Liepmann H W & A Roshko, Elements of Gas Dynamics, John Wiley, New York 1958.

Trumpler P R, US Patent No. 4,196,773 of April 8, 1980
"Heating and Air Cooling Employing a Gas Turbine".

END

DATE FILMED

01 / 08 / 91