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PRELIMINARY DESIGN OF AN EXPERIMENTAL CONTAINERIZED FREEZE DESA--ETC(U)

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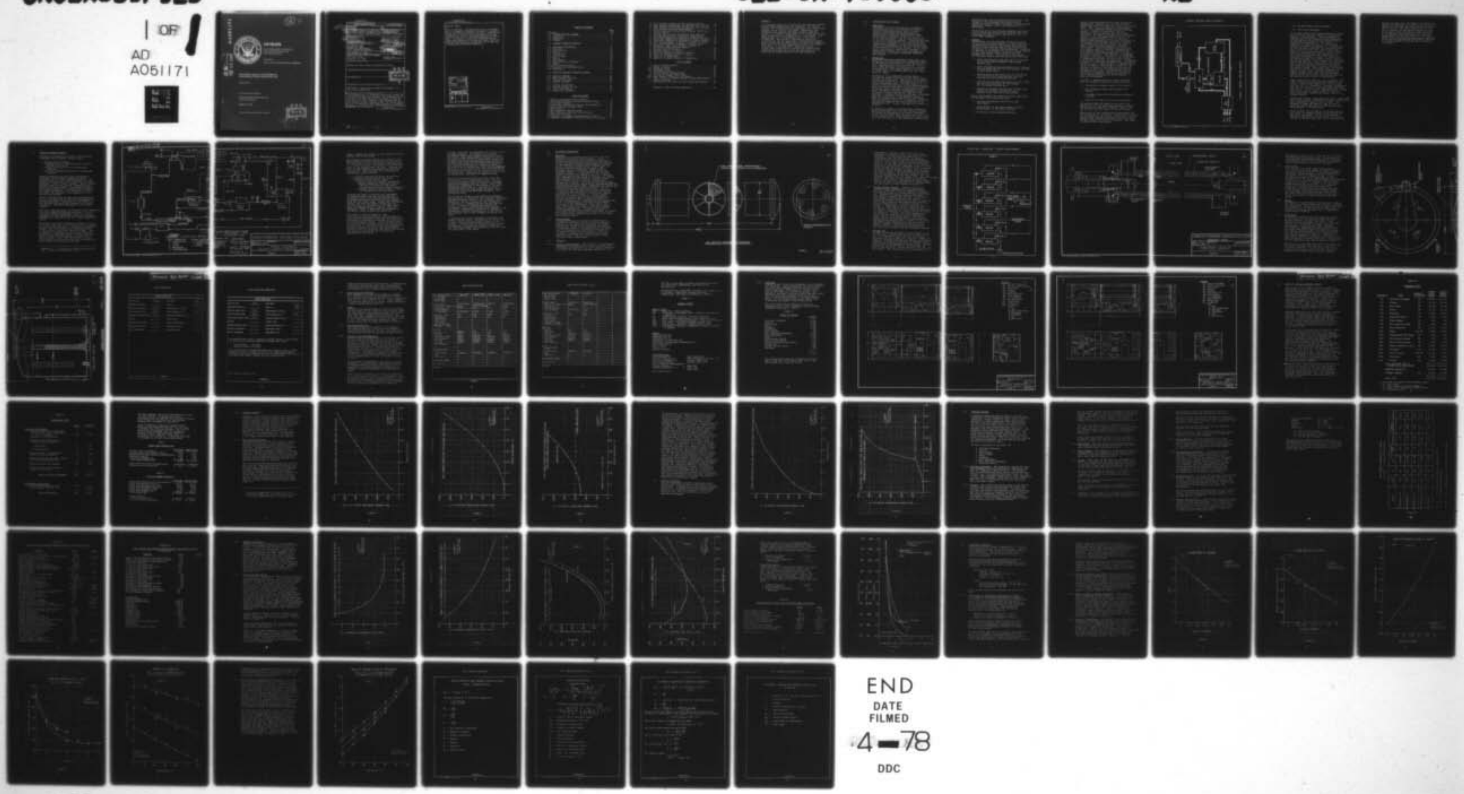

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CIVIL ENGINEERING LABORATORY
Naval Construction Battalion Center
Port Hueneme, California

Sponsored by

NAVAL FACILITIES ENGINEERING COMMAND

PRELIMINARY DESIGN OF AN EXPERIMENTAL CONTAINERIZED FREEZE DESALINATION UNIT

December 1977

An Investigation Conducted by

CONCENTRATION SPECIALISTS, INC.
Andover, Massachusetts

N68305-77-C-0019

UNCLASSIFIED

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A preliminary design of a 20,000 gpd containerized indirect contact freeze desalinator was completed. The system consists of a freezer with a defrosting cycle; wash column; indirect contact melter; and pumps, compressors, heat exchangers and controls necessary to make a complete system. At a nominal design point of 3.5% feed and at 70°F, power consumption is estimated

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to be 77 kw-hr/kgal. The construction cost of a prototype unit is estimated to be $280,000. The system is designed to operate on virtually any type of feed water. Projected performance data is given for the system operating with feed salinities from .5% to 5% and feed temperatures from 40°F to 100°F. In addition to a system with a defrosted freezer, studies were conducted of a scraped tube freezer which indicated that both the capital and operating cost of that type of freezer would be much more expensive.

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ABSTRACT

A preliminary design of a 20,000 gpd containerized indirect contact freeze desalinator was completed. The system consists of a freezer with a defrosting cycle; wash column; indirect contact melter; and pumps, compressors, heat exchangers and controls necessary to make a complete system. At a nominal design point of 3.5% feed and at 70°F. power consumption is estimated to be 77 kw-hr/kgal. The construction cost of a prototype unit is estimated to be $280,000. The system is designed to operate on virtually any type of feed water. Projected performance data is given for the system operating with feed salinities from .5% to 5% and feed temperatures from 40°F to 100°F. In addition to a system with a defrosted freezer, studies were conducted of a scraped tube freezer which indicated that both the capital and operating cost of that type of freezer would be much more expensive.

1.0 INTRODUCTION AND SUMMARY

1.1 Objective

The objective of the contract was to prepare a preliminary design of an experimental indirect contact freeze desalination unit to process seawater into potable water conforming to Environmental Protection Administration (EPA) drinking water standards to fit into an 8 ft. x 20 ft. ANSI container. The freezing system was to have the following features: A freezer employing a closed cycle refrigerant loop cooling through a surface heat exchanger. An ice wash column which limits the concentration of total dissolved solids in the product to less than 500 ppm. An ice melter which is an indirect contact heat exchanger. The design should reflect the latest knowledge and improvements in processes, equipment components and controls.

1.2 Background.

The Civil Engineering Laboratory (CEL) under the sponsorship of Naval Facilities Engineering Command is developing an experimental freeze desalination unit, containerized to fit into an 8 ft. x 8 ft. x 20 ft. ANSI container for application at advanced military bases. The target capacity of the unit is 20,000 gallons of potable water per day from a seawater supply.

The work is to be completed in three phases: preliminary design, final design and construction, and test and evaluation. This report covers Phase I, preliminary design of an experimental containerized freeze desalination unit. Now that Phase I is complete, CEL will determine if the proposed design offers advantages over other methods of desalination in capital cost, operating and maintenance costs, power consumption, logistic requirements, and/or the level of specialization and sophistication required to operate and maintain the unit. Depending upon CEL's evaluation of the proposed design, a Phase II contract may be awarded for final design and construction of the unit, followed by a Phase III contract for complete test and evaluation of the experimental unit. Phase I included two elements as follows:

An analytical study to size desalinator components based upon process flow, and heat and mass balances. The selection and design of individual components in the unit was based upon the least life cycle cost of the complete unit. Consideration was given to the fact that the end use of the developed containerized desalinator will be at advanced military bases. The designs of the components were carried out in sufficient detail to permit estimates of their overall dimensions, weights, costs, and effectivess (or efficiency).

Integration of all the selected equipment components into a preliminary design of the complete unit, and arranging the layout to fit into an 8 ft. x 8 ft. x 20 ft. ANSI container.

1.3 Summary

The statement of work has been carried out without exception and with one addition. It was found that scraped surface freezers were excessively expensive and would cause both the capital and operating cost of the unit to be greater than anticipated. Therefore, in addition to the design of a scraped surface freezer, a plain tube freezer with a defrost cycle was designed and costed. Based on these two concepts the following results and conclusions have been made:

- o Plain tube freezers have been used for freezing and are much more economical than scraped surface units.
- o Power consumption was calculated to be 76 kw/kgal. for the plain tube unit vs. 175 kw/kgal. for the scraped surface unit.
- o Capital costs of the first unit was calculated to be $278,000 for the plain tube unit and $425,000 for the scraped surface unit.
- o Life cycle costs were calculated to be $9.73/kgal. for the plain tube unit vs. $14.86/kgal. for the scraped surface unit.
- o Sensitivity studies indicate that the unit can operate on virtually any type of sea or brackish water that can be anticipated.

Freeze desalination has several potential advantages for portable, Naval desalting units:

- o Low power consumption and thus low fuel requirements.
- o Applicability to any water without extensive pretreatment, or corrosion problems.
- o No fouling of heat transfer surfaces.

Several freezing processes have been proposed for desalination. However, most of them are quite complicated and not suitable for small containerized units. This study was conducted to investigate the use of a relatively simple indirect freezing process. Indirect process means that the refrigerant does not contact the water being processed. All previous processes have used direct contact of the refrigerant with the water. An indirect process eliminates all of the steps involved with deaeration of the feed, stripping of refrigerants from effluent streams and removing non-condensibles from the process that cause direct contact freezing processes to become complicated. Indirect processes have been used to a limited extent in the food, pharmaceutical, and chemical industries for the production of such items as coffee and paraxylene. A schematic is shown in Figure 1. The basis of the freezing process is that when an ice crystal is frozen from a contaminated solution, the ice crystal formed is pure water excluding all impurities. The process consists of a feed heat exchanger where the feed is cooled to near its freezing point, a freezer where ice is produced, a wash column to purify the ice crystals, a melter to melt the ice crystals and recover the water and a refrigeration system to accomplish the cooling. Except for the freezer all of these components have been developed for other freezing processes and do not require extensive development.

In order to complete the work on this contract, work was primarily directed to these activities:

- o Developing a computer model of the process.
- o Heat transfer studies and design of the freezer.
- o Equipment selection and preliminary design of a system.
- o Sensitivity study of the resulting process.

The computer model has been used to select the optimum size for the feed heat exchanger, freezers, melter and heat rejection condenser, geometry of the freezers and flow rates through the freezers.

Three modes of heat transfer were investigated for the refrigerant in the freezer; nucleate pool boiling, forced convection boiling and sensible heat transfer. Sensible heat transfer with high circulation rates has been found to be the most efficient. Two types of freezers were investigated:

INDIRECT FREEZING PROCESS SCHEMATIC

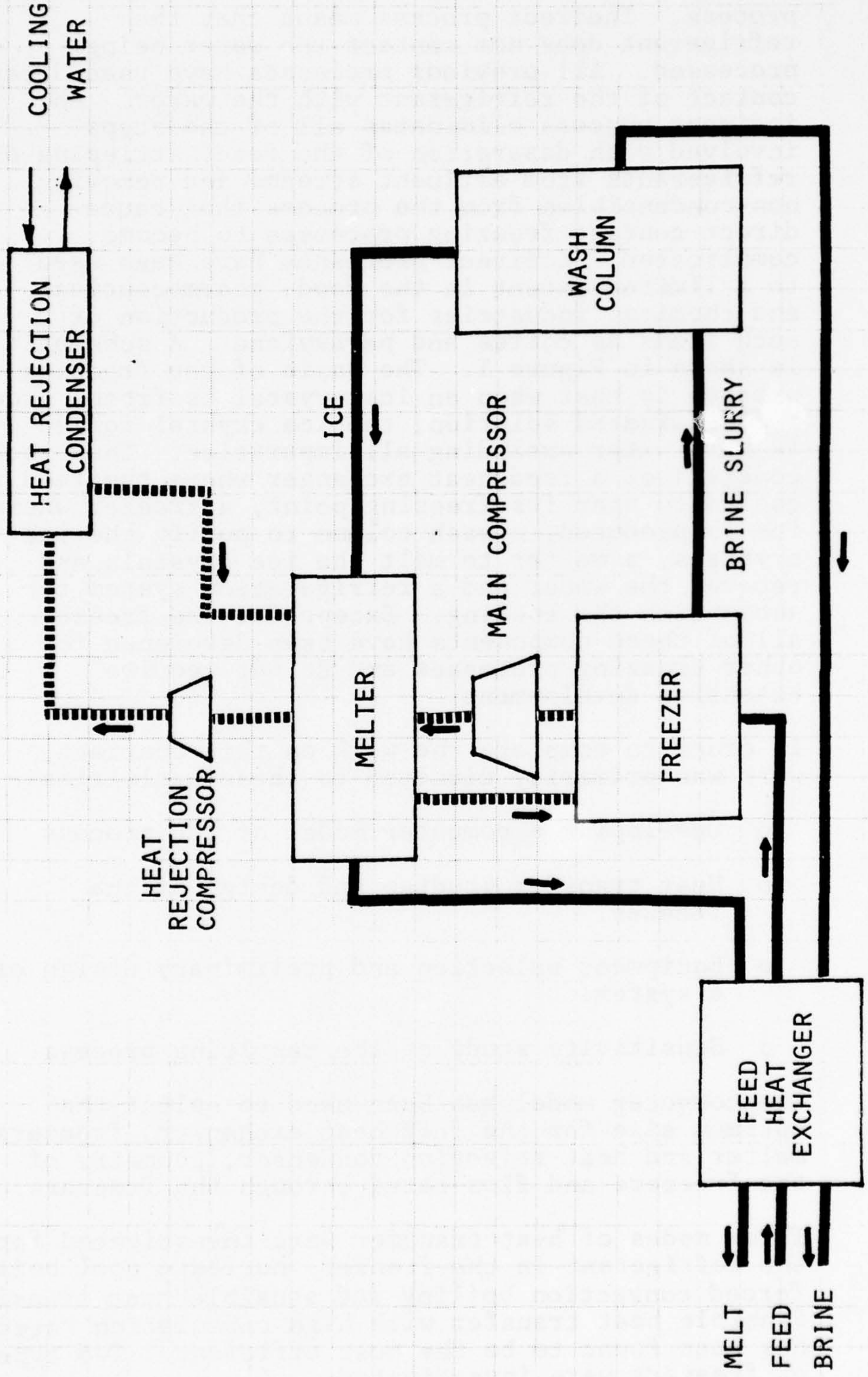

FIGURE 1 -INDIRECT FREEZING PROCESS

(1) Scraped surface heat exchangers

(2) Plain tube exchangers.

The scraped surface type would eliminate ice build-up on the heat transfer surface by continuously removing it mechanically. These exchangers are commonly used in the chemical process industries for heating and cooling viscous solutions and in crystallizing applications. The costs of commercially available units is prohibitively expensive for desalting however, so a scraped unit has been designed especially for this application. Cost of this unit was still significantly more expensive than a plain tube unit. Three types of plain tube freezers have been considered; shell and tube heat exchangers with defrosting or ultrasonic vibration to prevent ice accumulation and fluidized beds with ceramic or metal beads to prevent ice buildup. The ultrasonic and fluidized bed systems have never been tested, but some test work has been done on a defrosted cycle system. This study was expanded to include evaluation of the defrost system. This system uses six freezers operating in parallel, allowing defrosting one of them at a time. Tests have been conducted (by others) demonstrating that the period between defrost cycles can be quite long (hours) and that the defrost cycle is short (a few minutes).

Pumps, compressors, controls, heat exchangers, and other standard equipment have been specified and priced. A design was completed on a wash column that will fit in a container. Equipment arrangements were made to show how a 20,000 gpd plant would fit into a 8 ft. x 8 ft. x 20 ft. ANSI container. Plants with either type of freezer would fit, however space constraints limited the scraper type to 18 freezer modules.

Sensitivity studies to determine operating capabilities were conducted over a feed salinity range of .5% to 5% and a temperature range of 40°F to 100°F. The extremes of power and capacity over the entire range was from 19,000 gpd to 22,600 gpd and from 59 kw-hr/kgal. to 94 kw-hr/kgal.

Feed strainers are included in the system and the only limit on suspended solids is the quanity that the filter can handle. Solids below the size removed by the filter will not effect the process. The only pollutants that will not be removed by the process are those that are mixed or dispersed (as opposed to dissolved) in the feed water and have an affinity for ice such that they will not be washed off of the ice. The exact quanity of oils that can be tolerated in the feed is unknown, in that no testing of this type has ever been conducted. It is estimated that quanities of at least five and probably ten ppm can effectively be removed. Higher quantities than that which can be washed off of the ice will not impare the operation, but will be carried through the process and be contained in the melt.

2.0 INDIRECT FREEZING PROCESS

Freezing as a desalination process offers several advantages over processes currently used.

- o Absence of pretreatment.
- o Applicability to a wide variety of solutions without modification.
- o Reduced corrosion due to low temperature operation.
- o Low power consumption compared to evaporative processes.

Freezing processes that have been developed for desalting[1] are relatively complicated and not suited for use in a small containerized unit. The complications arise out of the advantages of these processes -- direct contact freezing. Direct contact freezing leads to very low power consumption and absence of scaling; however, it requires deaeration (for vacuum processes) or stripping (for secondary refrigerant processes), a non-condensibles system, and causes the compressor to be exposed to a very corrosive environment.

Indirect freezing does not have these disadvantages but it still maintains all the above advantages of freezing processes. The penalty is the addition of heat transfer surface and an increase in the power consumption. The major advantage is a far simpler process.

Even with these penalties, the process can be competitive with other competing desalting processes. The heat transfer surface is comparable to that of a vapor compression or MSF evaporation process and the power consumption is similar to that of a vapor compression system.

The process, as are all freezing processes, is based on the fact that when ice crystals form from a solution, the crystals are pure water, excluding all impurities. However, they are coated with a concentrated brine of the sea water from which they were formed. Thus, in addition to freezing, two other major steps are required for a freezing process -- washing the brine off of the crystals and melting the ice to form a useful product. Other steps are required to grow the crystals so they can be washed and to recover the energy required to freeze the crystals.

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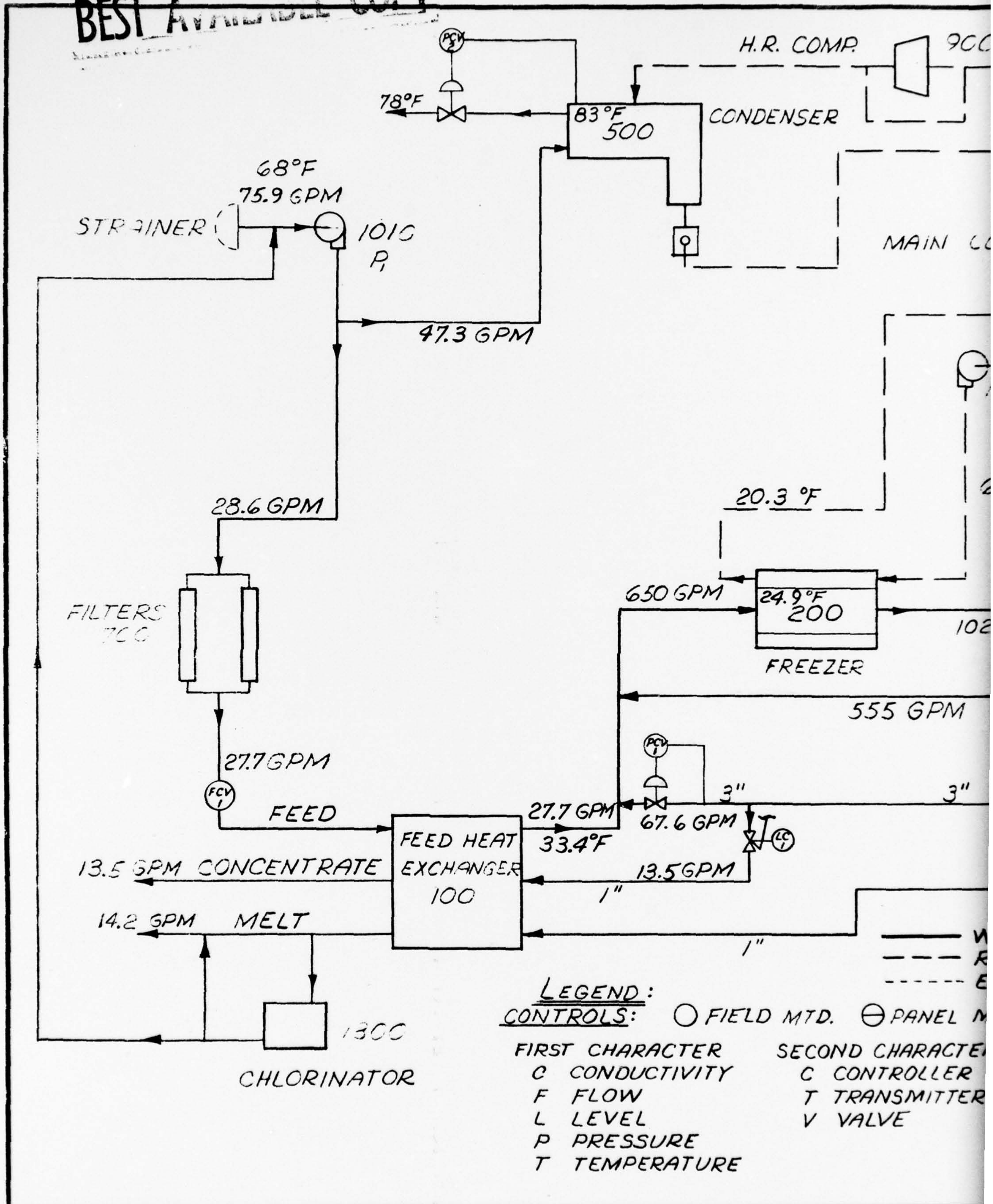
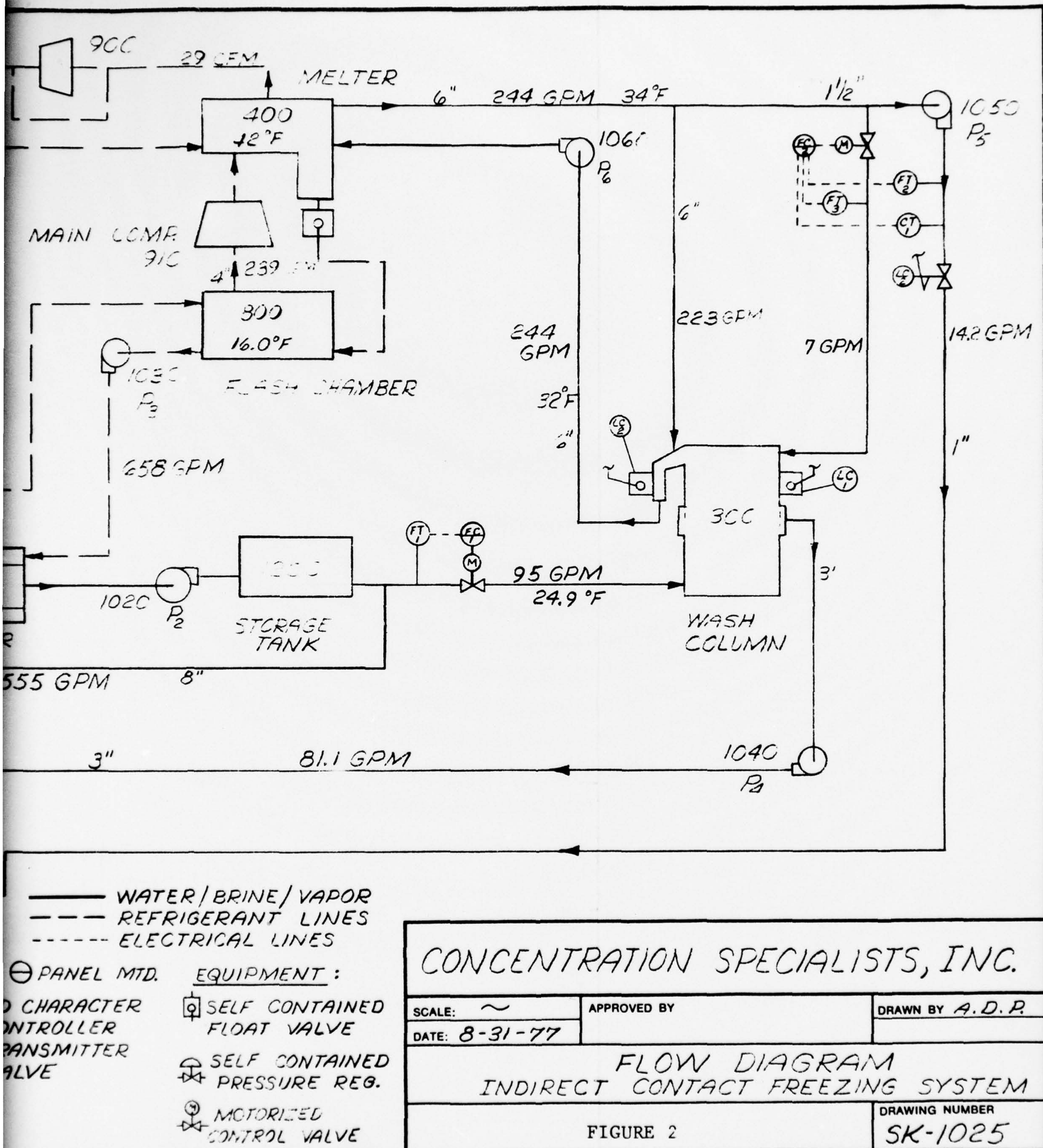
1 Barduhn, A.J., "The Status of Freeze Desalination", CEP, 71 (No. 11), 80-87 (Nov. 1975)



H.R. COMP.
900
78°F
83°F
500
CONDENSER
68°F
75.9 GPM
STRAINER
1010
P_1
MAIN
47.3 GPM
28.6 GPM
20.3 °F
FILTERS
700
650 GPM
24.9°F
200
FREEZER
555 GPM
27.7 GPM
FEED
3"
27.7 GPM
67.6 GPM
33.4°F
FEED HEAT EXCHANGER 100
13.5 GPM CONCENTRATE
13.5 GPM
1"
14.2 GPM MELT
1"
1300
CHLORINATOR
LEGEND:
CONTROLS: FIELD MTD. PANEL
FIRST CHARACTER
C CONDUCTIVITY
F FLOW
L LEVEL
P PRESSURE
T TEMPERATURE
SECOND CHARACTER
C CONTROLLER
T TRANSMITTER
V VALVE


900
29 CFM
MELTER
400
42°F
6"
244 GPM
34°F
1½"
1050
P_5
1060
P_6
MAIN COMP.
910
4"
239
800
16.0°F
1030
P_3
FLASH CHAMBER
658 GPM
244 GPM
32°F
6"
223 GPM
7 GPM
14.2 GPM
1"
300
WASH COLUMN
95 GPM
24.9°F
3'
1020
P_2
STORAGE TANK
555 GPM
8"
3"
81.1 GPM
1040
P_4
WATER/BRINE/VAPOR
REFRIGERANT LINES
ELECTRICAL LINES
PANEL MTD.
CHARACTER
CONTROLLER
TRANSMITTER
VALVE
EQUIPMENT:
SELF CONTAINED FLOAT VALVE
SELF CONTAINED PRESSURE REG.
MOTORIZED CONTROL VALVE
CONCENTRATION SPECIALISTS, INC.
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DRAWN BY A. D. P.
FLOW DIAGRAM
INDIRECT CONTACT FREEZING SYSTEM
DRAWING NUMBER
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FIGURE 2

Figure 2 shows the schematic of the indirect freezing process used in this study.

Feed enters the system through a dual filter system which removes particles that might clog the system. The feed passes through a plate heat exchanger where it is cooled to within a few degrees of its freezing point by the outgoing melt and brine. The feed sea water is then mixed with recirculating brine and ice crystals. Recirculation thru the freezer is required for several reasons:

- o To provide seed crystals which can grow and reduce nucleation on the tube walls
- o To maintain a high velocity which:
  - provides good heat transfer coefficients
  - reduces temperature difference between the wall and brine and thus limits nucleation and crystal growth on the wall.
  - Provides turbulence which tends to limit ice formation on the tube walls.

Although two types of freezers were considered, the scraped surface and the multiple section defrosted freezer, both are functionally similar. A refrigerated jacket surrounds the tube through which the ice slurry flows. Evaporating refrigerant removes heat from the slurry forming more ice. The ice slurry is transferred to the wash column where the adhering brine is removed, while the refrigerant vapor is compressed and condensed in the melter, where it melts the washed ice.

Both types of freezers depend on high velocities to scrub the surface, recirculation of ice to provide nucleation sites and low temperature differences to minimize or eliminate the build up of ice on the freezer surface. In the scraped surface unit the scraper is used to insure that ice build up is minimized. In the defrost type the refrigeration suction is always shut off to one of the several sections while the relatively warm feed flows through that same section causing any ice which has accumulated to be melted.

The wash column has been developed and used in several freezing processes. It consists of a vertical cylindrical vessel with screens located about half way up the column through which the brine is removed, a scraper at the top of the column which removes the ice and a wash water distribution system at the top. Ice enters the bottom of the column and consolidates into a porous piston which is propelled upward by the flow of brine through the piston (not by bouyancy). The brine exits at the screens, the piston continuing upward with the majority of the brine drained off due to gravity flow. Wash water is applied to the top of the column which displaces the remaining brine.

The ice is scraped off the top of the column and fluidized by a recirculating melt flow. The ice is circulated through a shell and tube heat exchanger with the refrigerant from the freezer, having been compressed, condensing on the outside of the tubes causing melting of the ice. The melted ice and concentrate from the wash column are pumped out of the system through the feed heat exchanger, thus cooling the feed.

Due to lack of closure on the feed heat exchanger and heat input to the process due to pumps, compressors and ambient losses, there is not sufficient cooling capacity in the ice to cause all the refrigerant to condense at a temperature near the melting point of ice. In order to efficiently condense this refrigerant a heat rejection system is employed utilizing a compressor to increase the pressure of the uncondensed (excess) refrigerant to a sufficiently high pressure that it will condense on ambient sea water.

A chlorination system is provided to chlorinate the product and feed lines. Chlorination of the product is used to insure bacteriological purity of the product. The feed lines to the system must be dosed periodically to prevent any growth of organisms in the feed system which would eventually grow and block the feed lines to the system.

3.0 EQUIPMENT DESCRIPTION

3.1 Materials

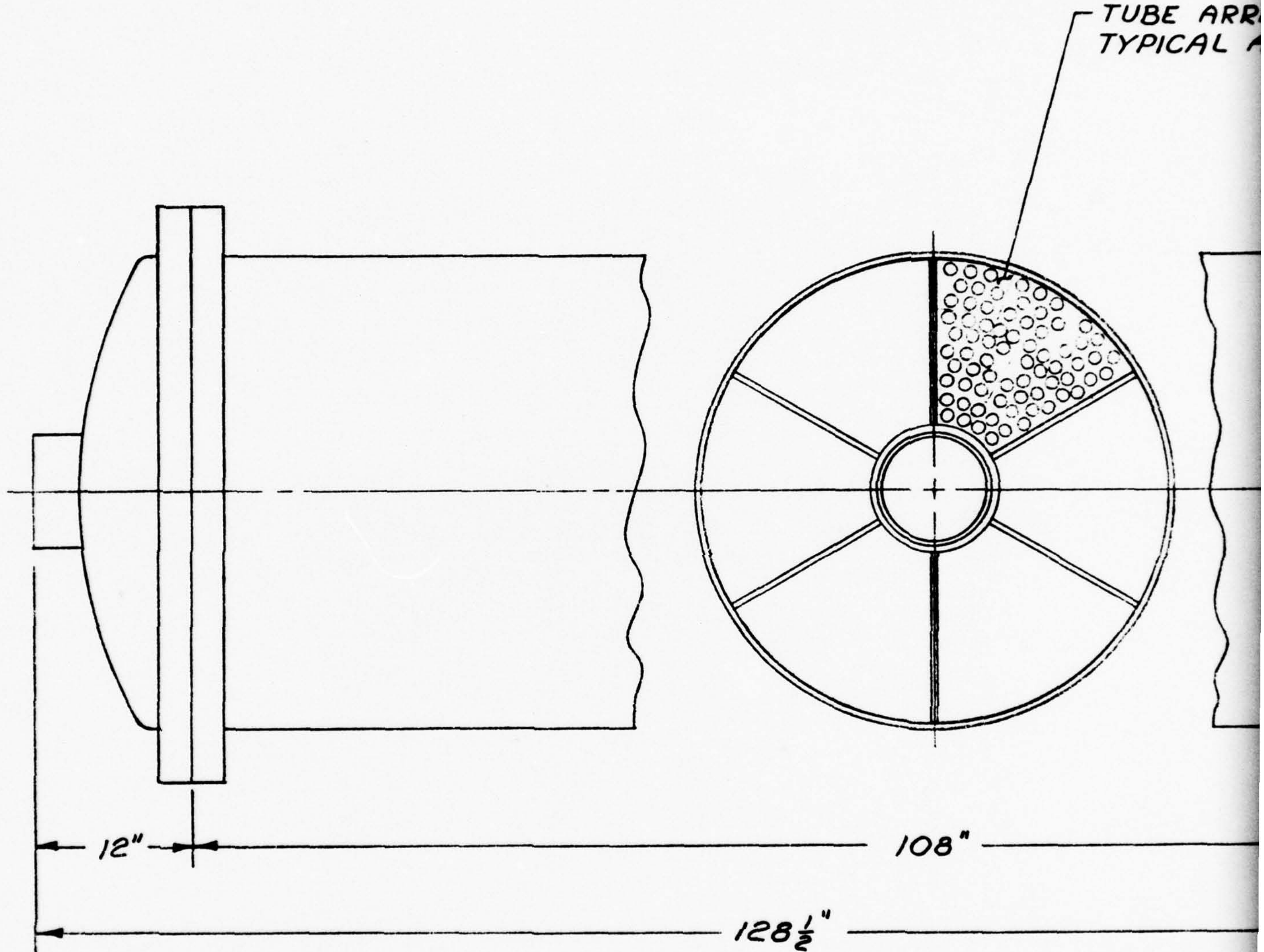
Because of the intended applications for this unit, material selection is very stringent. A very conservative selection of materials has been made, so that the desalting unit can operate with any type of feed whether it contains large amounts of heavy metals, various polluted waters, including H_2S, or the high chloride content of sea water. This precludes the use of aluminum, copper alloys or stainless steels up to and including 316 for heat transfer surfaces. A relaxation of this criteria would have significant impact on the cost. The feed heat exchanger is constructed of titanium, the freezer and condenser have 26-1 stainless tubing and the melter, which is only in contact with melt and refrigerant is 316 SS. For vessels that have thicker sections the corrosion impact is not significant and less costly materials can be used. Therefore, all vessels are constructed of epoxy coated mild steel. 26-1 is a relatively new stainless that has been developed to resist pitting and crevice corrosion in chloride environments and is superior to 316 in this respect while not as costly as alloy 20. Copper nickel was not selected because of its poor performance in polluted waters, while aluminum is subject to attack by heavy metals. Aluminum has been demonstrated to be a suitable material for desalting processes operating at low temperatures, below 140°F, with over a 100 plants in operation using 5052 aluminum tubing for evaporators. The use of this material in a freezing plant for a "normal" sea water application would significantly reduce costs and should be considered when comparing the costs of this plant with other processes.

3.2 Feed Strainer

A duplex strainer with automatic backwashing is provided in the feed prior to the heat exchanger to prevent dirt and debris from entering the system. Although the system is not especially susceptible to suspended solids, it is good practice to prevent them from entering the system where they can silt out and eventually clog any orifices. Tube type strainers, available from several manufacturers, have been used. They are specified with 40 mesh screens of monel construction.

3.3 Freezers

3.3.1 Defrost Cycle Freezer. This freezer is the preferred embodiment of the process because of its lower power consumption and capital cost. This freezer (Figure 3) is a shell and tube heat exchanger divided into six



SIX SECTION FREEZE

2

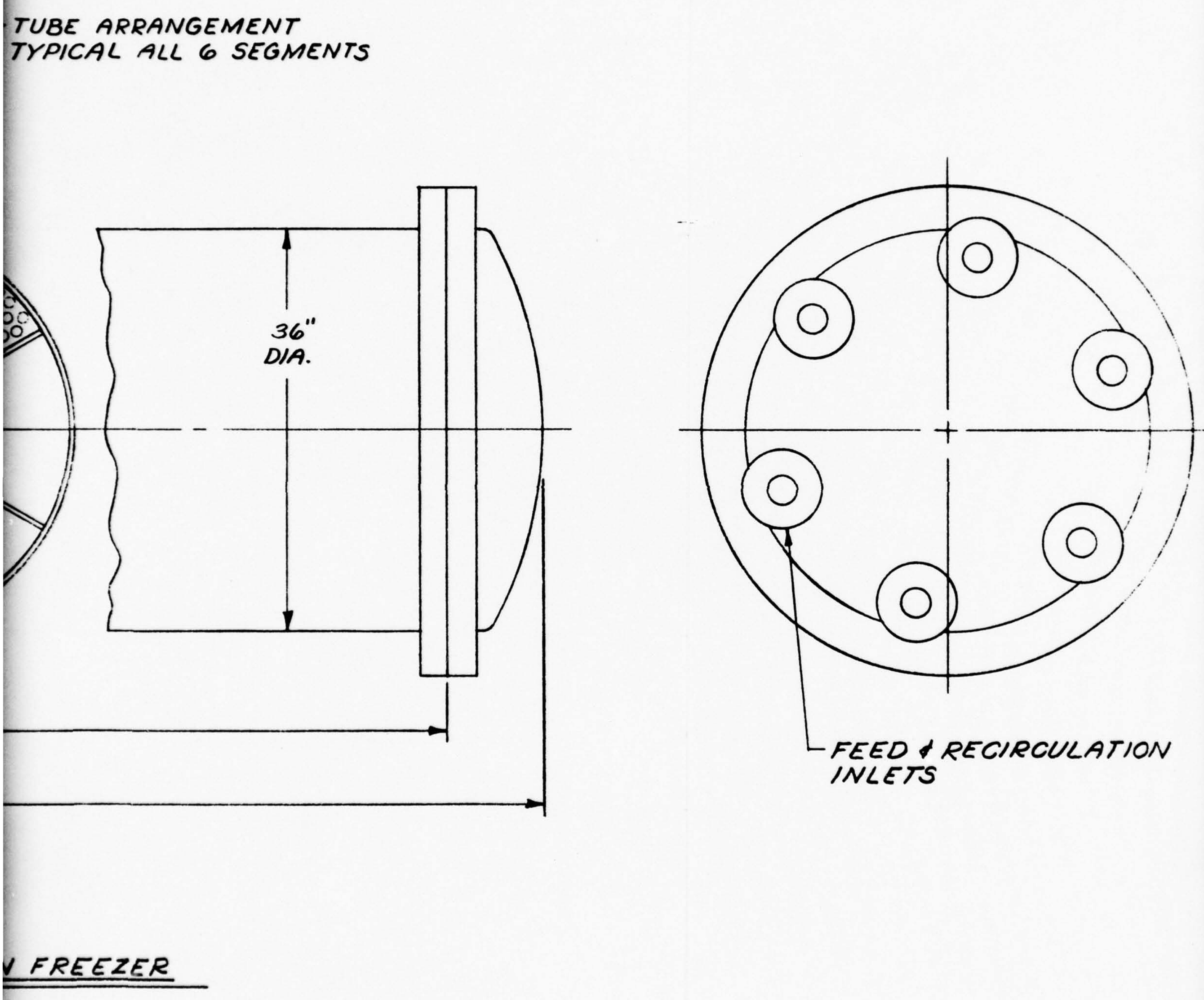


FIGURE 3

SK-1037

longitudinal sections, five of which would be freezing at a given time while the other is being defrosted. To accomplish the defrosting the refrigerant line from the freezer to compressor (suction line) would be shut off during the defrost cycle so that no heat could be extracted from that section. At the same time the feed to the freezer, which exits the feed heat exchanger at 33.4°F, (4.8°F above its freezing point of 28.6°F), would be routed to the section of the heat exchanger that is being defrosted. Figure 4 shows the valving requirements. The sensible heat in the feed would remove any ice that had accumulated on the tube surface. The heat exchanger contains 432-1 in. tubes, 9 ft. long; 72 in each section. The tubes are 26-1 stainless while the shell is carbon steel. Refrigerant is recirculated through the tubes to obtain high heat transfer rates. (See discussion of heat transfer in Section 5.)

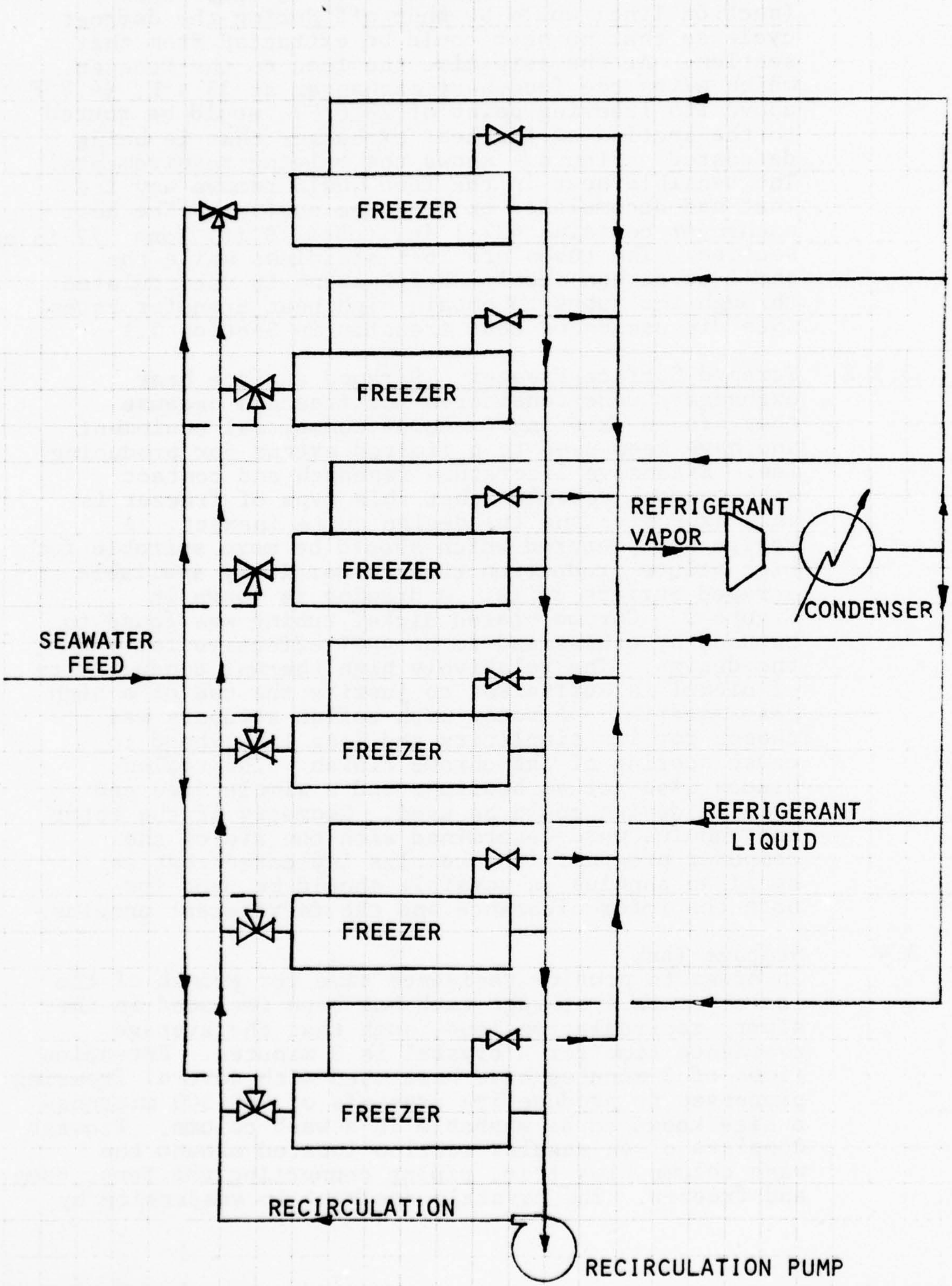
3.3.2 Scraped Surface Freezer. Scraped surface heat exchangers were considered for freezers because they are an existing piece of commercial equipment and have been used to a limited extent for producing ice. Extensive literature research and contact with vendors revealed that this type of freezer is very expensive and its design quite inexact. A design was prepared which should be more suitable for high volume production than commercially available scraped surface units. A drawing is shown in Figure 5. Chrome plated nickel tubing was found to be used by others and to be cost effective for the design. The relatively high thermal conductivity of nickel is sufficient to justify the use of a high cost material. A rotor with teflon scrapers was chosen for its simplicity and less likelyhood to cause scoring of the chrome finish. The teflon blades also act as bearings and a simple free end bearing design could be used. Geometry of the rotor and annulus were determined with the aid of the computer program. The results indicated that as small an annulus as possible should be used for both the rotor clearance and the refrigerant annulus.

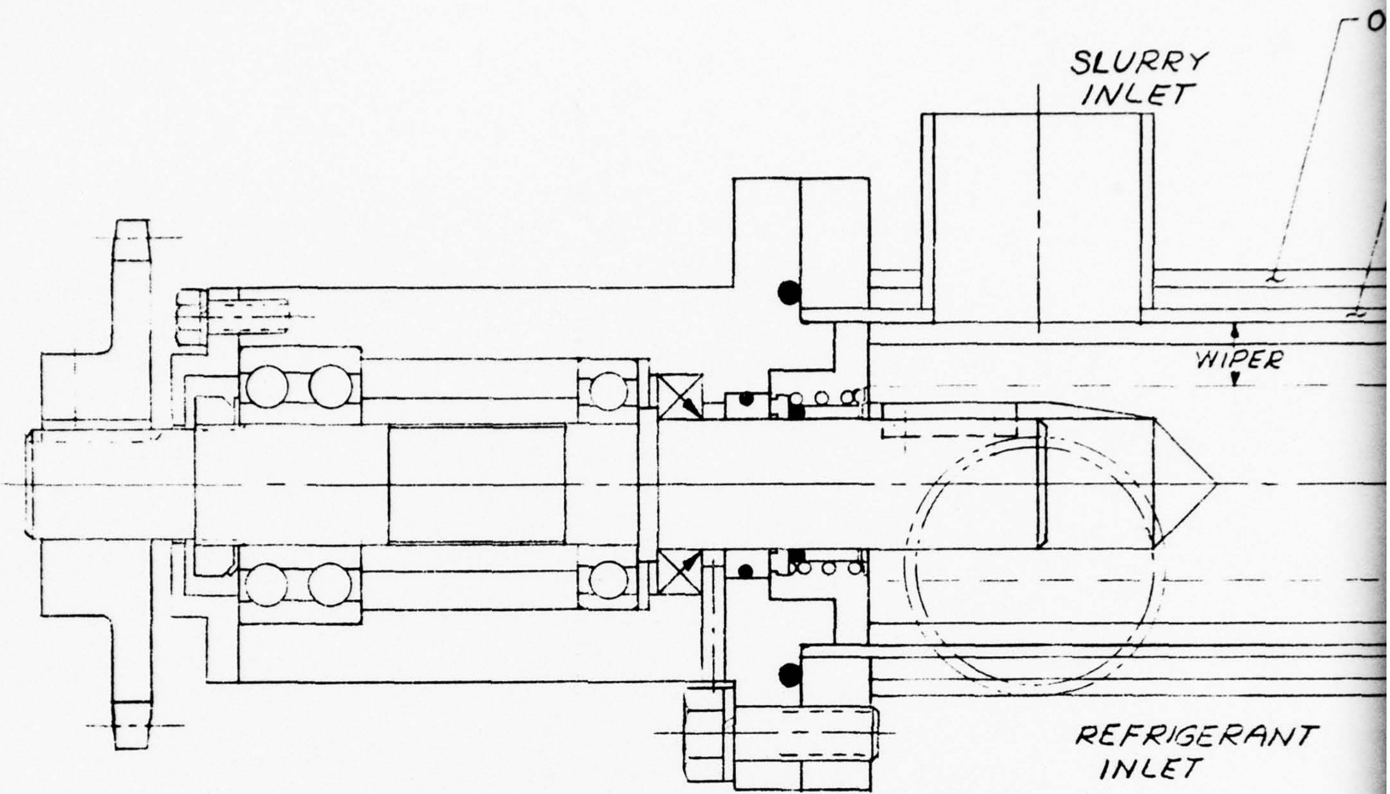
3.4 Storage Tank

In order to provide residence time for growth of the ice crystals a storage tank has been included in the slurry recirculation loop, such that the average residence time for a crystal is 3 minutes. Retention times of 3 minutes have been used with several freezing processes to produce ice crystals of 200-300 microns -- a size known to be washable in a wash column. Storage consists of an annular section located around the wash column plus 8 in. piping connecting the tank, pump and freezer. The crystals are kept in suspension by

PLAIN TUBE - DEFROSTING - FREEZER FLOW SCHEMATIC

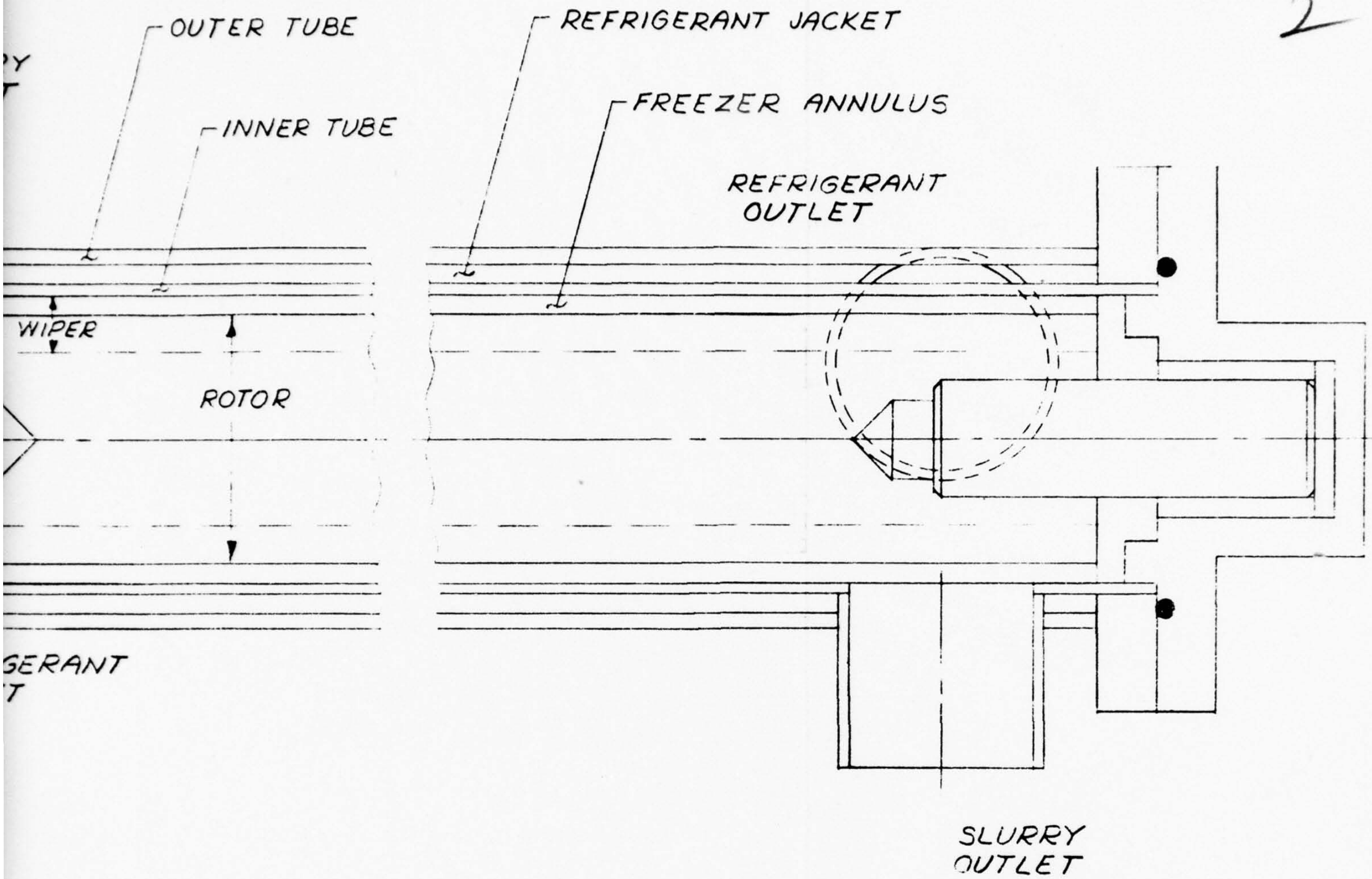
FIGURE 4





SLURRY INLET
WIPER
REFRIGERANT INLET

2



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ANDOVER, MASS.

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FREEZER LAYOUT

FIGURE 5

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maintaining a velocity of at least one ft. per second throughout the recirculation loop. Tests made with other freezing systems have shown that this velocity is sufficient to keep the ice crystals in suspension rather than floating to the top of the pipe.

3.5 Wash Column

A 5½ ft. ID, 7 ft. high, epoxy coated vessel is provided for the wash column as shown in Figure 6. Ice slurry enters the bottom of the column through 4 tangential nozzles located at the mean diameter of the column. Brine is removed through 7 brine drainage tubes, one in the center and the other 6 located on a 50 in. circle. The drainage screen consists of a series of 1/8 holes drilled in PVC pipe about midway up the column. A scraper at the top of the column cuts the ice and moves it to a tangential outlet on one side of the column where it is mixed with a recirculating flow of melt and pumped to the melter. Wash water is distributed from a rotating header located behind each cutter blade. The slurry retention tank referred to in Section 3.4 can be seen around the lower portion of the wash column.

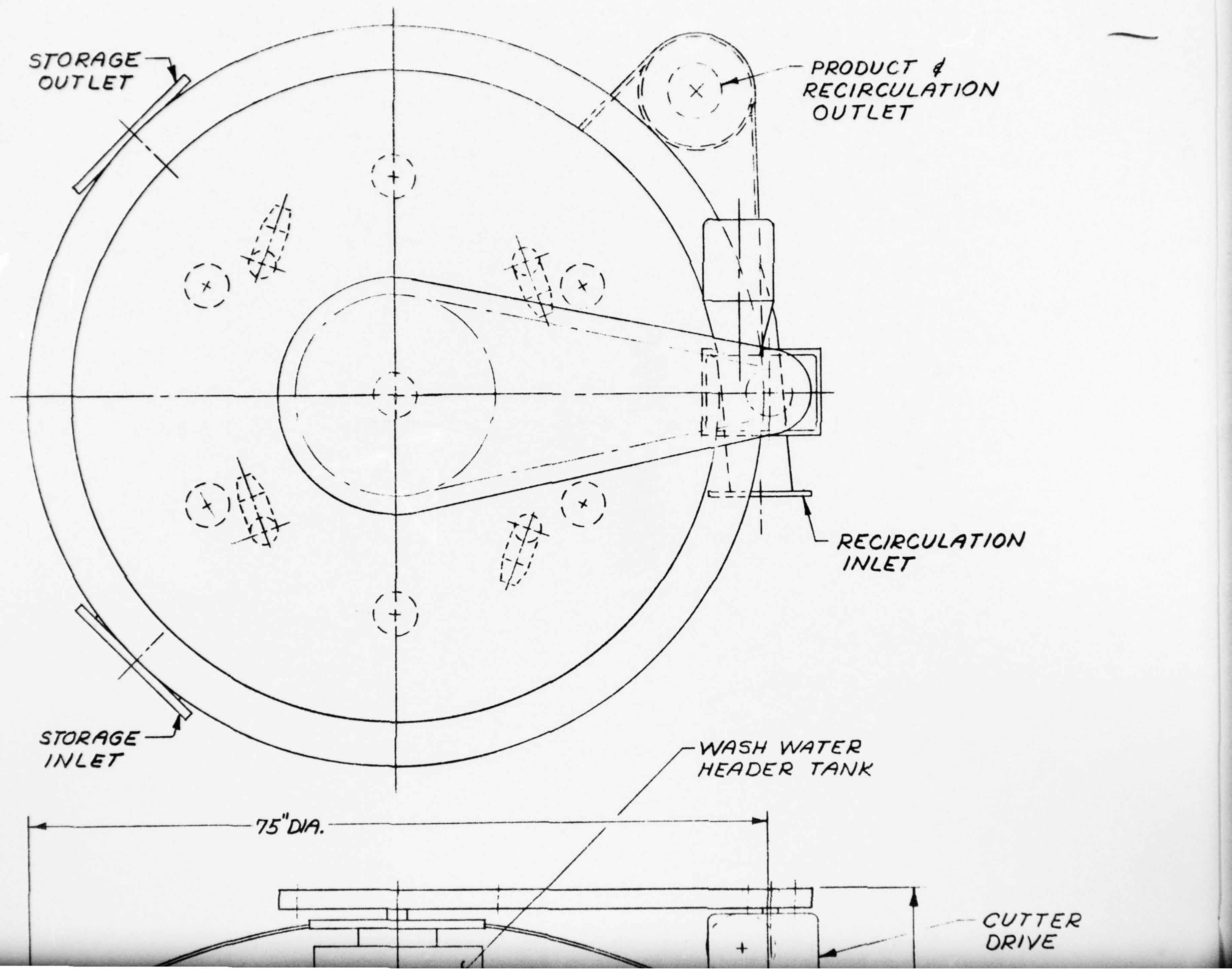
3.6 Melter

The melter is a conventional shell-and-tube condenser. One inch 316 stainless finned tube is used to condense the refrigerant. An overall heat transfer coefficient of 82 BTU/hr-ft.^{2}-°F was calculated. This gives a condenser containing 232 tubes, 10 ft. long with a finned area of 1,389 ft.^{2}

3.7 Compressors

An extensive evaluation of compressors was made and it was found that reciprocating compressors were the most efficient in this size range. Screw compressors were considered, but in this size range they are not as efficient or compact as either centrifugals or reciprocating types. Although reciprocating compressors are more efficient, they are larger than centrifugals and therefore because of packaging considerations a centrifugal has been used for the main compressor with a reciprocating heat removal compressor. This is a significant cost penalty since the centrifugal costs about three times as much as a reciprocating type. Specifications for the compressors are given in Figures 7 and 8.

During start up, the main compressor is by-passed and the entire refrigeration load taken by the heat rejection compressor. For this reason and to have better control of the process the heat rejection compressor is sized at about twice its required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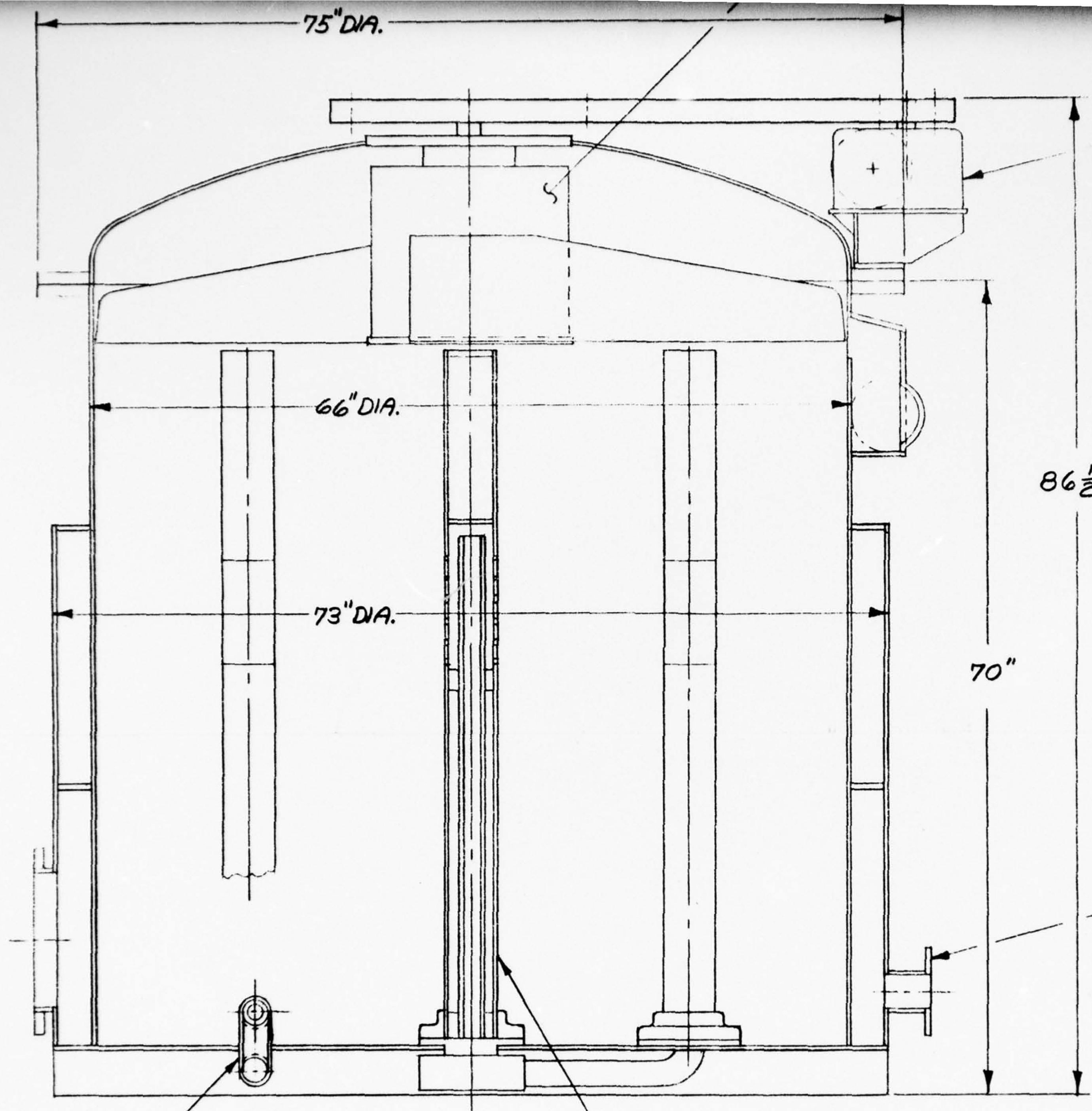
PRODUCT & RECIRCULATION OUTLET
RECIRCULATION INLET
STORAGE INLET
WASH WATER HEADER TANK
75" DIA.
CUTTER DRIVE



2

WASH COLUMN

FIGURE 6

SK-1026

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MAIN COMPRESSOR

DESIGN CONDITIONS			
Type of Gas	R-22	Flow ACFM	255
Relative Humidity %		Mole Wt.	86.5
Specific Heat Ratio	1.283	Compressibility Factor	
Inlet Pressure PSIA	47.5	Inlet Temperature °F	10
Discharge Pressure PSIA	83.2	Discharge Temperature °F	
Coolant		Coolant Flow	
Coolant Pressure PSIG		Coolant Temperature °F	
Required Horsepower		Speed RPM, Pinion	
Driver Horsepower		Speed RPM, Motor	

Ref: Letter of June 21, 1977

FIGURE 7

HEAT REJECTION COMPRESSOR

DESIGN CONDITIONS			
Type of Gas	R-22	Flow ACFM	92
Relative Humidity %		Mole Wt.	86.5
Specific Heat Ratio	1.35	Compressibility Factor	
Inlet Pressure PSIA	81.8	Inlet Temperature °F	39
Discharge Pressure PSIA	177.9	Discharge Temperature °F	
Coolant		Coolant Flow	
Coolant Pressure PSIG		Coolant Temperature °F	
Required Horsepower		Speed RPM, Pinion	
Driver Horsepower		Speed RPM, Motor	

The compressor must be able to operate (at reduced capacity) at the following condition when the condensing temperature is higher than average.

Inlet Pressure - 81.8 psia
Discharge Pressure- 225.5 psia

The compressor must be equipped with some sort of capacity control to permit varying the capacity to between 25 and 40% of the nominal design capacity. Depending on the type of compressor this could be accomplished by means such as inlet guide vanes, bypass, cylinder cut outs, or speed control.

Ref: Letter of June 21, 1977

FIGURE 8

capacity during steady state operation. Reciprocating compressors are good for this type of application since with cylinder unloading they operate nearly as efficiently at part load as they do at full load.

3.8 Heat Rejection Condenser

This condenser is a plain tube condenser, because 26-1 stainless steel is not available in a finned tube design. The condenser has 373 3/4 in. tubes 10 ft. long for an overall area of 732 ft.² This condenser is oversized relative to its normal leading because of a higher loading during start up.

3.9 Pumps

Standard close coupled pumps are used in the system. The feed pump will be of alloy 20 construction, while the other pumps in the system have cast iron casings with 316SS trim. Experience has shown that for pumps and other equipment with heavy walls, cast iron is an acceptable material as long as the temperature is low. All pumps will have mechanical seals. Specifications for the pumps are given on Figure 9.

3.10 Chlorination System

The chlorination system consists of two chemical feed pumps mounted on a 30 gallon drum feeding a concentrated hypochlorite solution. One pump treats the melt and the other is used for the feed system.

3.11 Controls and Instrumentation

Controls have been simplified to the greatest extent so that operation of the system is as simple as possible. Self contained regulators have been used wherever possible to simplify the system and reduce maintenance problems. Levels are controlled by float operated control valves. Conventional electronic controllers are used to control the wash water and slurry flow. The capacity of the heat removal compressor is controlled by sensing the temperature of the melt and unloading compressor cylinders at various temperatures. A list of controls is given in Table I.

The system will automatically shut down in case of equipment or system failure. Sensors are included to determine when any malfunction occurs and an annunciator to indicate where the malfunction is located. A list of alarm points is also given in Table I.

Start up and shut down of the system is controlled by a single start/stop button. A programmable logic controller (PLC) is used to sequence events so that no operator intervention is required during start up. A manual mode is also provided for trouble shooting so that individual components can be operated.

PUMP SPECIFICATIONS

Pump Identification	FEED-1010	RECIR.-1020	REFRIG.-1030	CONC.-1040
No. of Pumps				
Manufacturer				
Type of Pump				
Liquid Pumped	3.5% Sea H_2O	7.5% Sea H_2O	Freon 22	7.5% Sea H_2O
Description (Slurry, etc.)	Clear	20% Ice Fraction		Clear
Temperature - °F	45-85	24.4	8.3	24.4
Suction Pressure				
S.C./Viscosity	1.030/1.397	1.034/	1.344/	1.054/
Performance - GPM	113	518	1696	81.2
TDH-Req'd-Ft.	80	23	7.6	68
NPSH-Avail-Ft.				
BHP-Op. Cond.				
Curve Number				
Impeller Dia./Max.				
Size-Suct. x Disch.				
Materials				
Casing	20cb-3	C.I.	C.I.	C.I.
Rear Cover	20cb-3	C.I.	C.I.	C.I.
Impeller	20cb-3	316 S.S.	C.I.	316 S.S.
Shaft/Shaft Sleave	20cb-3	316 S.S.	316 S.S.	316 S.S.
Seal Hardware	20cb-3	316 S.S.	316 S.S.	316 S.S.
Seal Seat	Ceramic	Ceramic	Ceramic	Ceramic
Seal Ring	Carbon	Carbon	Carbon	Carbon
Seal Elastomer				
Motor				
Horsepower/RPM				
Frame				
Ph/Cycle/Volts	3/60/208	3/60/208	3/60/208	3/60/208
Type				
Insulation	F	F	F	F
Coupling-Mfg.-Type				
Total Unit Wt.				
Comments:				

FIGURE 9

PUMP SPECIFICATIONS Cont'd

Pump Identification	MELT-1050	W-C-1060		
No. of Pumps	1	1		
Manufacturer				
Type of Pump				
Liquid Pumped	Fresh H_2O	Fresh H_2O		
Description (Slurry, etc.)	Clear	5% Ice Fraction		
Temperature - °F	34	32		
Suction Pressure				
S.G./Viscosity	1.0017/1.34	.997/		
Performance - GPM	14.2	265		
TDH-Req'd-Ft.	33	19		
NPSH-Avail-Ft.				
BHP-Op. Cond.				
Curve Number				
Impeller Dia./Max.				
Size-Suct. x Disch.				
Materials				
Casing	C.I.	C.I.		
Rear Cover	C.I.	C.I.		
Impeller	316 S.S.	316 S.S.		
Shaft/Shaft Sleave	316 S.S.	316 S.S.		
Seal Hardware	316 S.S.	316 S.S.		
Seal Seat	Ceramic	Ceramic		
Seal Ring	Carbon	Carbon		
Seal Elastomer				
Motor				
Horsepower/RPM				
Frame				
Ph/Cycle/Volts	3/60/208	3/60/208		
Type				
Insulation	F	F		
Coupling-Mfg.-Type				
Total Unit Wt.				

Comments:

The PLC is also used to control the defrosting cycle, the feed filter backwashing, and feed system chlorination.

Instrumentation is provided to monitor the conductivity of the melt, significant process pressures, temperatures, and flows as indicated on Table I.

TABLE I

CONTROL SYSTEM

Control Loops

FR1 - Feed Flow - Flow regulator
PR1 - Condenser cooling water flow - Pressure actuated flow regulator
PR2 - Recirculation pressure - Pressure regulator
LC1 - Brine level in wash column - Float operated valve
LC2 - Melt level in fluidizing chamber - Float operated valve
FIC1 - Wash water - Electronic process control loop
FIC2 - Slurry flow - Electronic process control loop
TIC1 - Heat Removal Capacity - Temperature actuated cylinder unloading.

Alarms

Pump pressure (5)
Compressor ΔP (2)
Compressor ΔT (2)
Compressor oil pressure (2)
Compressor cooling water temperature (2)
Melt conductivity
Melter level
Filter ΔP (2)
Scraper rotation

Instrumentation

Pump pressure (6)	Feed temperature
Filters pressures (2)	Heat exchanger cold temp. (3)
Feed heat exchanger (3)	Freezer temperature
Wash column pressure	Cooling water exit
Melter pressure (refrigerant)	
Freezer Pressure (")	Feed flow
Condenser Pressure (")	Melt flow
	Brine flow
Melt conductivity	

3.12 Packaging

Figures 10 and 11 show the arrangement of the equipment in a container. Both arrangements are quite similar. The wash column dominates one end with the control center along side. The freezer is located on the side on the other end (with access doors). The mechanical equipment is located along side the freezer with the melter and condenser above the freezer. Doors are provided for access to the motor control center. On the scraped surface unit another door is provided for access to the scraper drive.

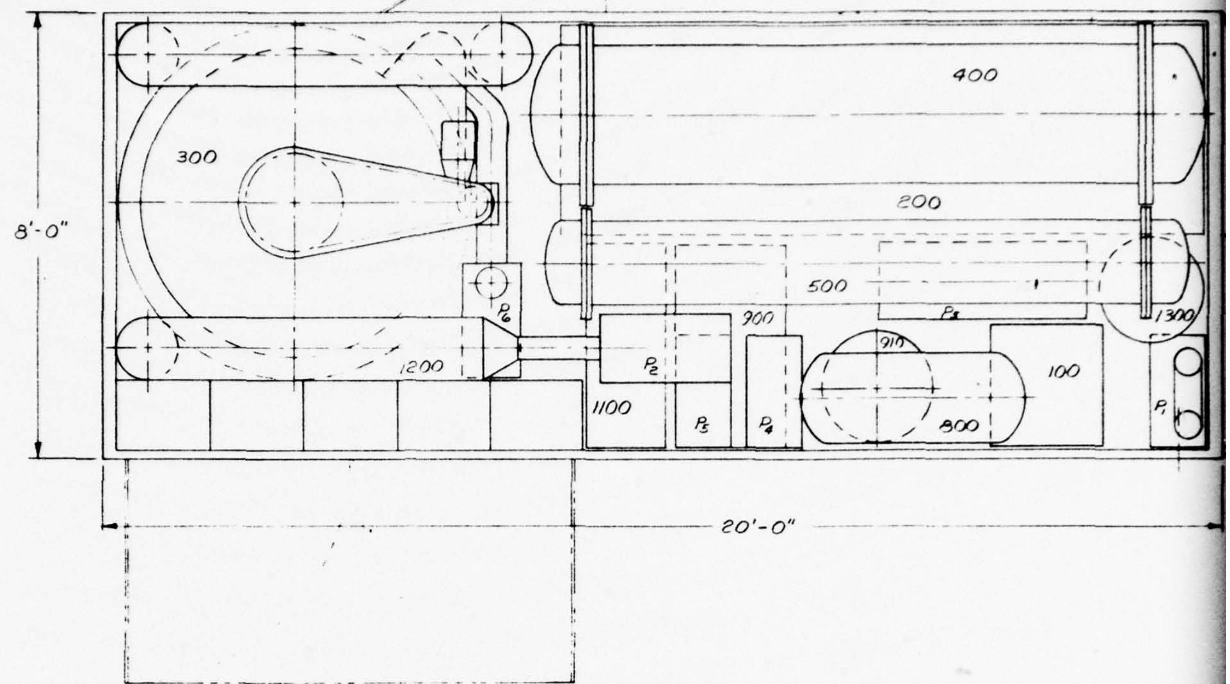
The total weight of the defrosted freezer is estimated to be 42,150. Weights of the individual components are given in Table II.

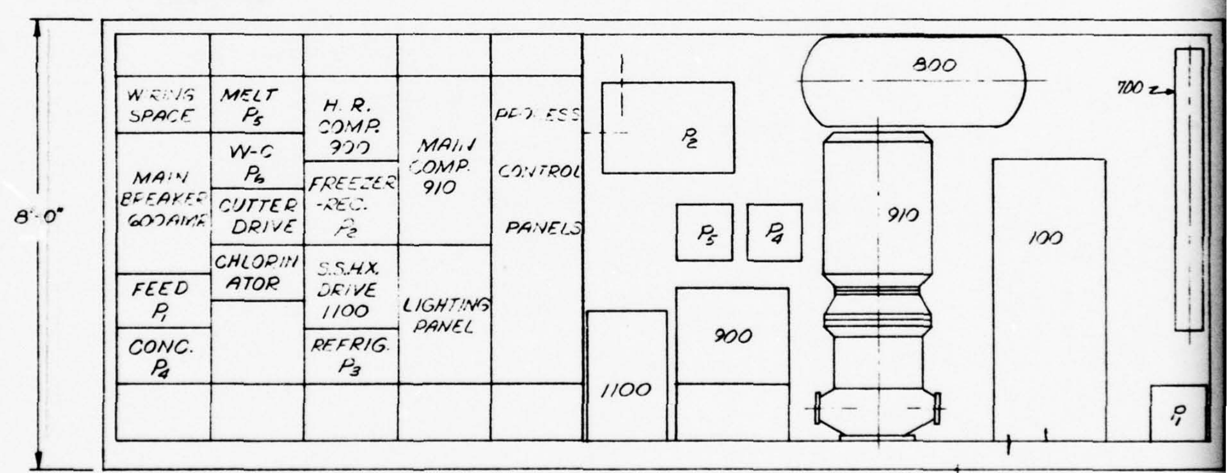
TABLE II

WEIGHT OF SYSTEM

	POUNDS
Feed Heat Exchanger	2,800
Freezer	8,700
Wash Column	6,200
Melter	5,000
Condenser	3,100
Feed Filter	260
Flash Chamber	300
Heat Rejection Compressor	1,770
Main Compressor	2,480
Pumps	2,360
Chlorination System	70
Motor Control Center	2,650
Process Controls & Instruments	600
Insulation	660
Piping, Control Valves	1,200
Container	4,000
	42,150

The scraped surface freezer assembly would weigh about 11,500 lbs. thus adding 2,800 lbs. to the above weight for a total of 44,950.


8'-0"
300
400
200
500
900
910
100
1300
1200
1100
800
20'-0"

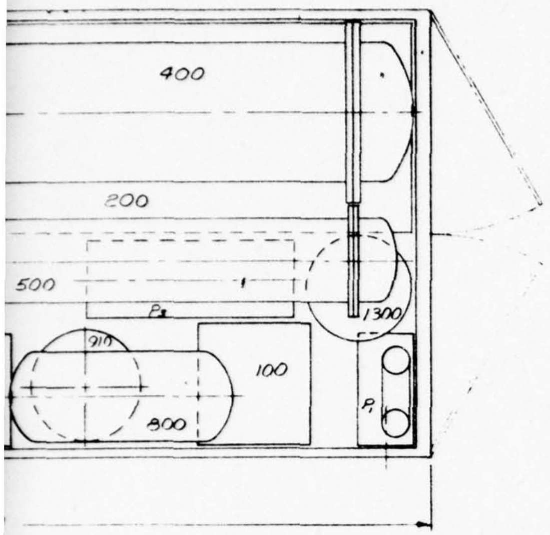

8'-0"
SPACE
MELT
H. R.
COMP.
MAIN
COMP.
910
CONTROL
PANELS
W-C
FREEZER
MAIN
CUTTER
DRIVE
CHLORIN
ATOR
S.S.H.X.
DRIVE
1100
FEED
LIGHTING
PANEL
CONC.
REFRIG.
800
910
100
900
1100

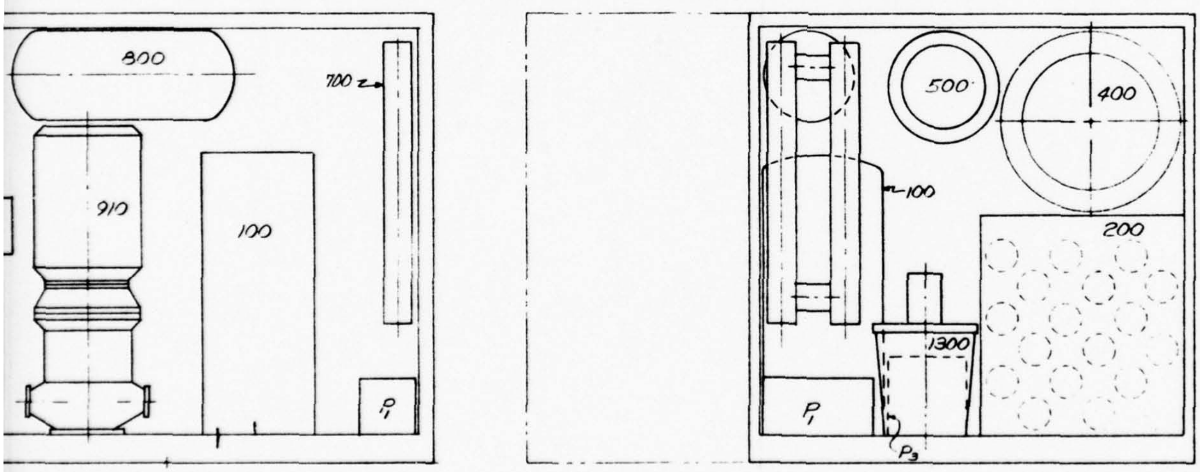
LEGEND:

100	FEED HEAT EXCHANGER
200	FREEZER - SCRAPED SURFACE
300	WASH COLUMN
400	MELTER
500	CONDENSER
700	DUAL FILTER - AUTO.
800	FLASH CHAMBER
900	H.R. COMPRESSOR
910	MAIN COMPRESSOR
1100	FREEZER DRIVE
1200	STORAGE TANK
1300	CHLORINATOR

PUMPS:

- P_1 FEED
- P_2 BRINE RECIRCULATION
- P_3 REFRIGERANT
- P_4 CONCENTRATE
- P_5 MELT
- P_6 WASH COLUMN



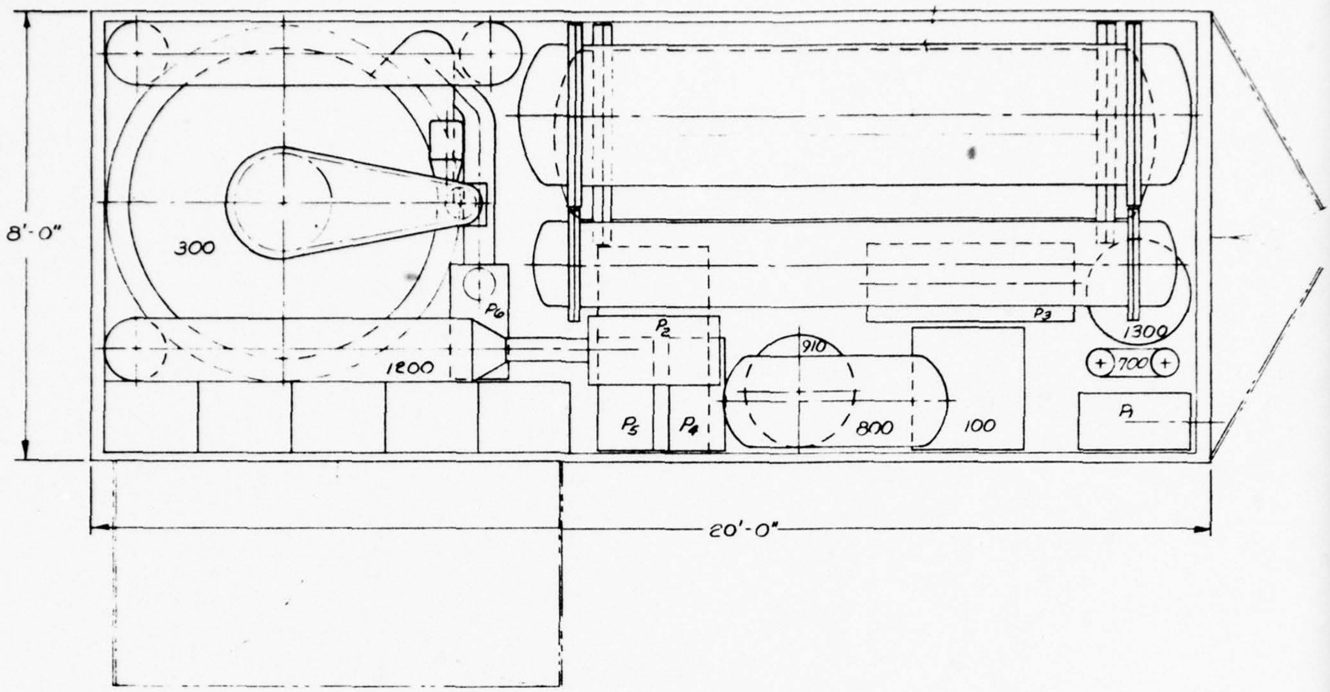


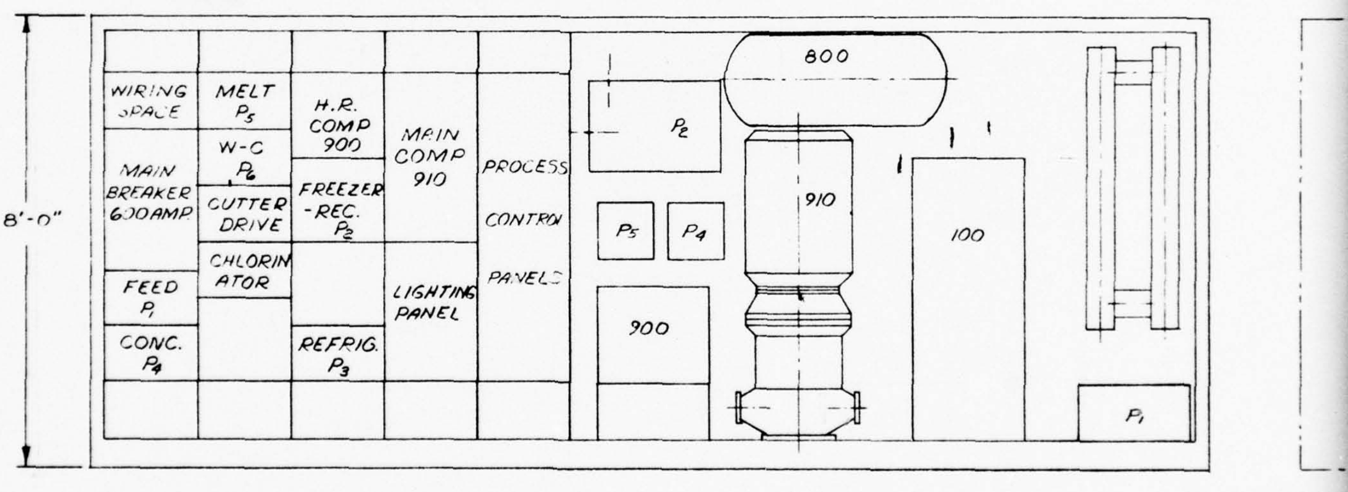
CONCENTRATION SPECIALISTS, INC.
ANDOVER, MASS.

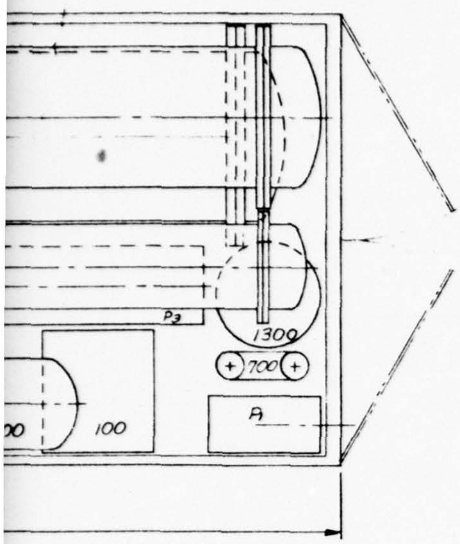
SCALE ~	APPROVED BY	DRAWN BY A.D.P.
DATE 10-25-77		REVISED

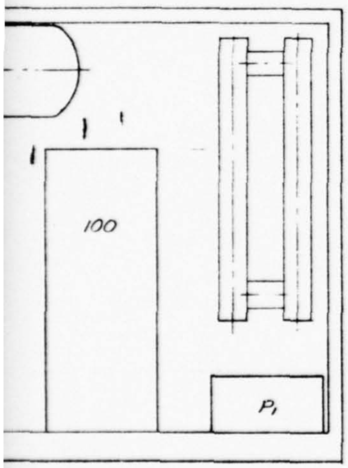
COMPONENT ARRANGEMENT
INDIRECT CONTACT FREEZING SYSTEM - SCRAPED

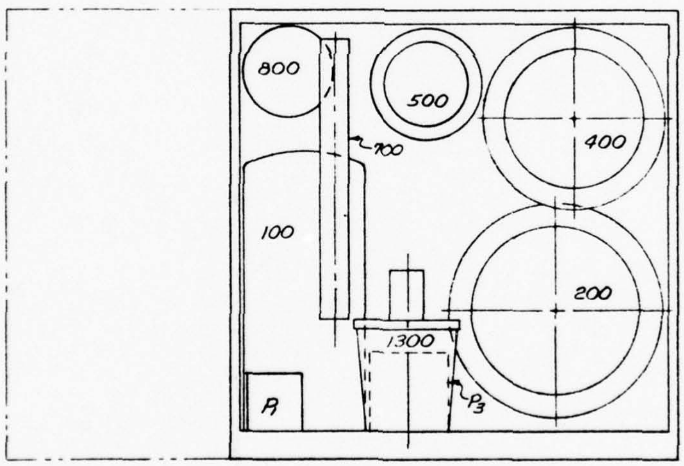
FIGURE 10 — DRAWING NUMBER SK-1040


8'-0"
300
1200
P6
P2
P3
1300
700
910
800
100
P5
P4
P1
20'-0"


8'-0"
WIRING SPACE
MAIN BREAKER 600 AMP.
FEED P1
CONC. P4
MELT P5
W-C P6
CUTTER DRIVE
CHLORIN ATOR
H.R. COMP 900
FREEZER -REC. P2
REFRIG. P3
MAIN COMP 910
LIGHTING PANEL
PROCESS CONTROL PANELS
800
P2
910
P5
P4
900
100
P1







LEGEND

100	FEED HEAT EXCHANGER
200	FREEZER - DEFROSTER
300	WASH COLUMN
400	MELTER
500	CONDENSER
700	DUAL FILTER - AUTO.
800	FLASH CHAMBER
900	H. R. COMPRESSOR
910	MAIN COMPRESSOR
1200	STORAGE TANK
1300	CHLORINATOR
	PUMP
P_1	FEED
P_2	BRINE RECIRCULATION
P_3	REFRIGERANT
P_4	CONCENTRATE
P_5	MELT
P_6	WASH COLUMN

CONCENTRATION SPECIALISTS, INC.
ANDOVER, MASS.

SCALE ~	APPROVED BY	DRAWN BY A.D.P.
DATE 10-21-77		REVISED

COMPONENT ARRANGEMENT
INDIRECT CONTACT FREEZING SYSTEM - DEFROSTER

FIGURE 11

DRAWING NUMBER SK-1041

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4.0 COSTS OF INDIRECT FREEZING SYSTEMS

The costing of the indirect freezing systems was based primarily on vendor quotes or vendor estimates for specific equipment for most components. Other costs are based on quotes for similar equipment or engineering estimates for similar applications. A cost summary and source of the cost data for the equipment is given in Table III. These costs should be considered as "study" accuracy with an accuracy of ±25%. From this it can be seen that the defrost cycle freezer has a tremendous advantage, the scraped surface unit costing more than 50% more than the defrost cycle system.

Although these costs are high they are for a system made of premium materials, tightly packaged in a container. A system made of more conventional materials and skid mounted might cost $50,000 less. Also this system includes a feed pump, filters, and chlorination system which add over $10,000 to the cost. These costs are also for the first of a kind and further reductions would be expected in production units.

Maintenance costs are tabulated on Table IV. These costs are quite speculative since a unit of this type has not been built. The system is composed of relatively simple pieces of equipment and should be easy to maintain without any special knowledge. Based on experience these costs should be conservative, assuming a mature design. No differentiation was made for the scraped surface unit, but it would be expected to have higher costs due to the added mechanical complexity of the freezers. Power costs are also much more favorable for a defrost cycle system. The horsepower for the two systems are 76 and 175 hp (77 and 178 kw-hr/ 1,000 gal., assuming a motor efficiency of 87%). At an operating factor of 83.3% and electrical cost of 3.5¢/kw-hr this is a difference between $49,498 and $70,833 per year between the defrost cycle and the scraped surface freezers.

Operating labor, should be limited to that required to check out the unit, start it up and inspect it every 4 to 8 hours to see that everything is operating satisfactorily. In most environments this labor would be shared with other types of equipment being operated. Thus, the equivalent of one full time operator has been assumed for the operating labor.

TABLE III

COMPONENT COSTS

ITEM NO. (1)	ITEM	SOURCE (2) OF COSTING	DEFROST CYCLE SYSTEM	SCRAPED SURFACE SYSTEM
100	Feed Heat Exchanger	VQ	$ 20,150	$ 20,150
200	Freezer	EE	31,125	155,560
300	Wash Column	VE	10,500	10,500
400	Melter	EE	25,700	25,700
500	Condenser	EE	12,900	12,900
700	Dual Feed Filter	VQ	4,925	4,925
800	Flash Chamber	EE	800	500
900	Heat Rejection Comp.	VQ	2,906	2,906
910	Main Compressor	VQ	16,700	16,700
1000	Pumps	VQ, SE	13,600	13,600
1200	Storage Tank (PVC Pipe)	VE	805	805
1300	Chlorination System	VE	945	945
1800	Motor Control Center	VQ	7,000	7,000
1900	Process Controls & Inst.	SE	20,153	20,153
2500	Container	VE	11,200	11,200
2600	Control Valves	VE, SE	10,124	2,372
3000	Insulation	EE	7,500	7,500
5000	Piping	EE	8,900	8,900
	TOTAL EQUIPMENT COST TO CONCENTRATION SPECIALISTS		$205,633	$322,316
	HANDLING CHARGE @21%		43,182	67,686
	ASSEMBLY CHARGES	EE	29,570	35,065
	TOTAL COST		$278,385	$425,067

(1) Same number that appears on system schematic Figure 2.

(2) VQ - Vendor quote
VE - Vendor estimate for specific equipment
SE - Vendor quote or estimate for similar equipment
EE - Engineering estimate

TABLE IV

MAINTENANCE COSTS

	HOURS	MATERIAL
SCHEDULED MAINTENANCE		
Scheduled inspection, replacement of pump seals, 8 hrs./pump once per year -- 6 pumps Material 6 seals @$150.00	48	$ 900
Scheduled inspection/repair of refrigeration compressors:		
- Centrifugal	32	150
- Reciprocating	64	400
Annual checkout, calibration of controls and instruments	40	100
General Inspection and minor repairs of system during shut down	40	500
General painting and cleaning	80	200
Routine inspection/maintenance -- weekly by operators	-	-
TOTAL SCHEDULE MAINTENANCE	304	$2,250
UNSCHEDULED MAINTENANCE		
5% of original assembly labor	100	
1% of original equipment cost		$2,500
TOTAL MAINTENANCE	404	$4,750

The only chemical cost is for hypochlorite to treat the feed and melt. A dosage of 2 ppm on a continuous basis for the melt and 5 ppm for 5 minutes every 8 hours for the feed, has been assumed. Hypochlorite is assumed to cost 61¢/lb.

Table V summarizes costs on a current annual operating cost basis and Table VI does the same on present value life cycle basis. For the life cycle costs an equipment life of 10 years is assumed, with 6% annual real cost growth for electrical power, with an initial value of electricity of $.035/kw-hr. A one year lead time is included in the life cycle calculations.

TABLE V

CURRENT ANNUAL OPERATING COST

	PLAIN TUBE	SCRAPED SURFACE
Electrical Cost at $.035/kw-hr	$ 16,476	$ 37,811
Operating Labor-1 man @ $14,000 + 66% OH	23,240	23,240
Maintenance Labor-404hrs.@$9.00/hr.+66%OH=$15.00/hr.	4,848	4,848
Maintenance Supplies	6,060	6,060
Hypochlorite-302lb.@$.61/lb.	184	184
TOTAL ANNUAL OPERATING COST	$ 50,808	$ 72,143
Annual Production at 83.3% Operating Factor	6,080,900gal.	6,080,900gal.
Operating Cost per 1,000 gallons	$ 8.35	$ 11.86

TABLE VI

LIFE CYCLE PRESENT VALUE COSTS

	PLAIN TUBE	SCRAPED SURFACE
Present Value Electrical with 6% real growth/yr.	$ 132,681	$ 304,492
Present Value Operating Labor Cost	136,210	136,210
Present Value Maintenance Labor Cost	28,414	28,414
Present Value Maintenance Supply Costs	27,840	27,840
Present Value Hypochlorite Costs	1,078	1,078
Present Value Equipment Costs	265,579	405,514
TOTAL PRESENT VALUE	$ 591,802	$ 903,548
Present Production		
- Annual Production x 10	60,809,000	60,809,000
Life Cycle Cost per 1,000 gallons	$ 9.73	$ 14.86

5.0 ANALYSIS METHODS

Analysis of the indirect freezing system was conducted primarily using a computer program which modeled the various components of the process. This model was used to determine optimum configurations for the two freezing systems studied, and then, with the optimum values fixed, sensitivity studies were conducted. Before the model could be constructed two preliminary analysis tasks were completed: (1) determination of the heat transfer mode in the freezer, and (2) determination of the refrigerant to be used. These two tasks plus a description of the computer model are given in this section.

5.1 Heat Transfer Analysis.

Two types of heat transfer were studied -- forced convection and boiling. Forced convection heat transfer was analyzed using the Dittus Boelter equation. (See Appendix A for equations referred to in this section.) Using this mode of heat transfer high velocities were required to achieve good heat transfer coefficients, Figure 12. At high velocities the pressure drop and corresponding pump work was high so investigation into boiling heat transfer was conducted to determine whether or not improved results could be obtained.

The first type of boiling investigated was nucleate pool boiling. While pool boiling of freon-22 can give high heat transfer coefficients this requires a higher temperature difference than is usable in the freezer (Figure 13).* At low temperature differences nucleate pool boiling heat transfer can be improved by superimposing a forced convection effect (Figure 14) but this still does not raise the heat transfer coefficient substantially above what you get with forced convection sensible heat transfer.

---

* The excess temperature in Figure 13 is the difference between the wall temperature and the saturation temperature of the boiling liquid.

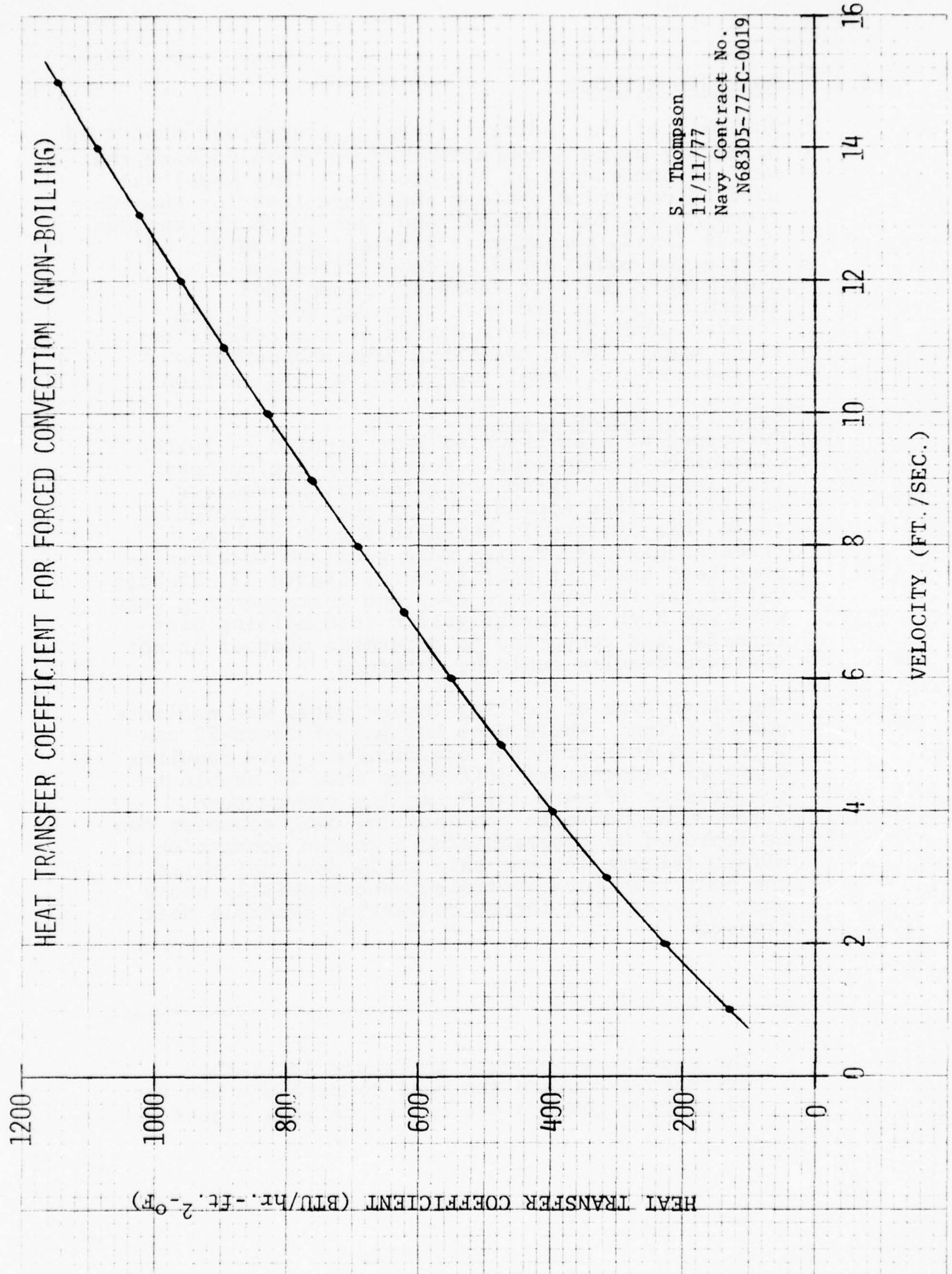

HEAT TRANSFER COEFFICIENT FOR FORCED CONVECTION (NON-BOILING)
S. Thompson
11/11/77
Navy Contract No.
N683D5-77-C-0019
VELOCITY (FT./SEC.)
0
2
4
6
8
10
12
14
16
HEAT TRANSFER COEFFICIENT (BTU/hr.-ft.^{2}-°F)
0
200
400
600
800
1000
1200

FIGURE 12

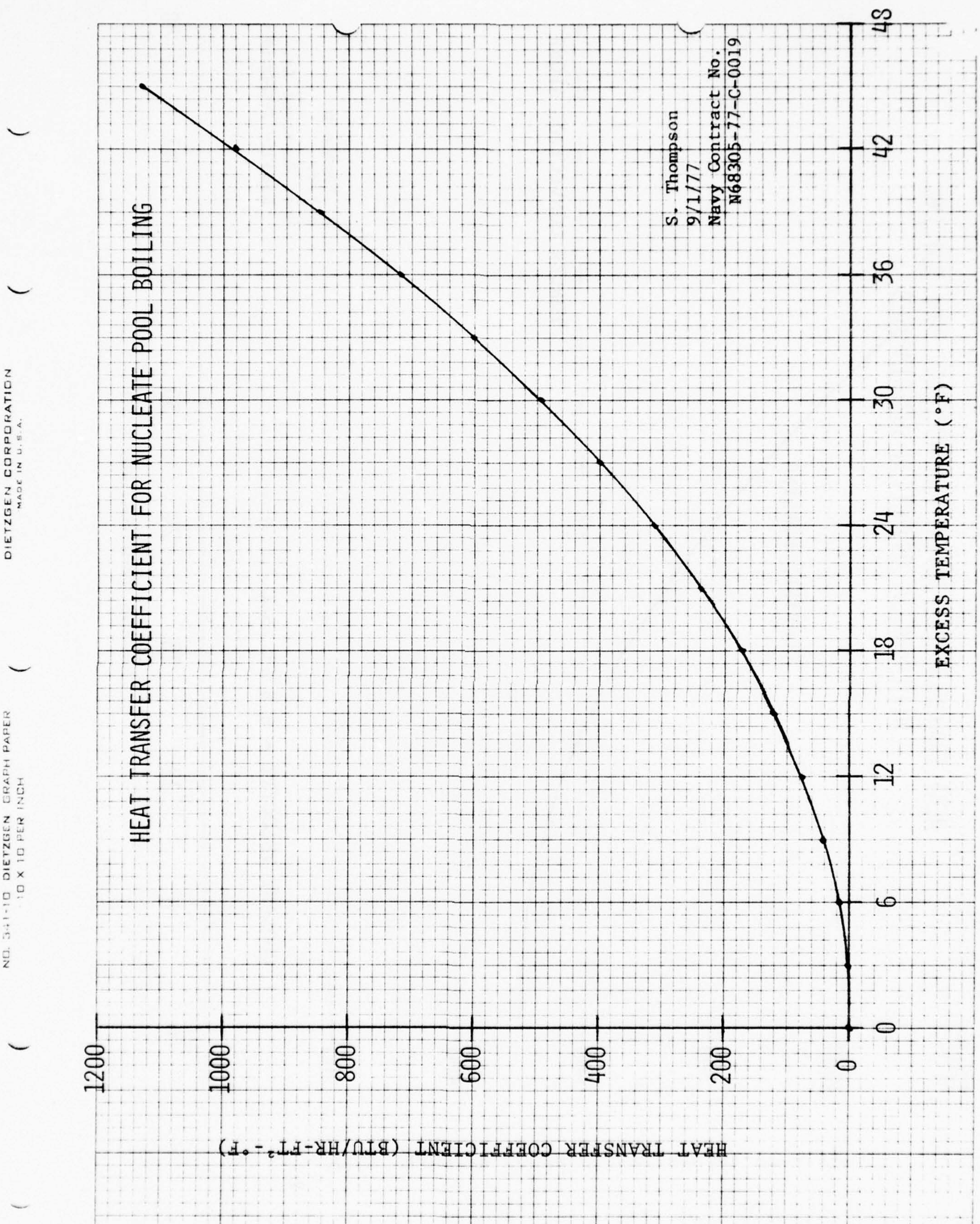

HEAT TRANSFER COEFFICIENT FOR NUCLEATE POOL BOILING
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019
HEAT TRANSFER COEFFICIENT (BTU/HR-FT²-°F)
EXCESS TEMPERATURE (°F)
0
200
400
600
800
1000
1200
0
6
12
18
24
30
36
42
48
NO. 341-10 DIETZGEN GRAPH PAPER
10 X 10 PER INCH
DIETZGEN CORPORATION
MADE IN U.S.A.

FIGURE 13

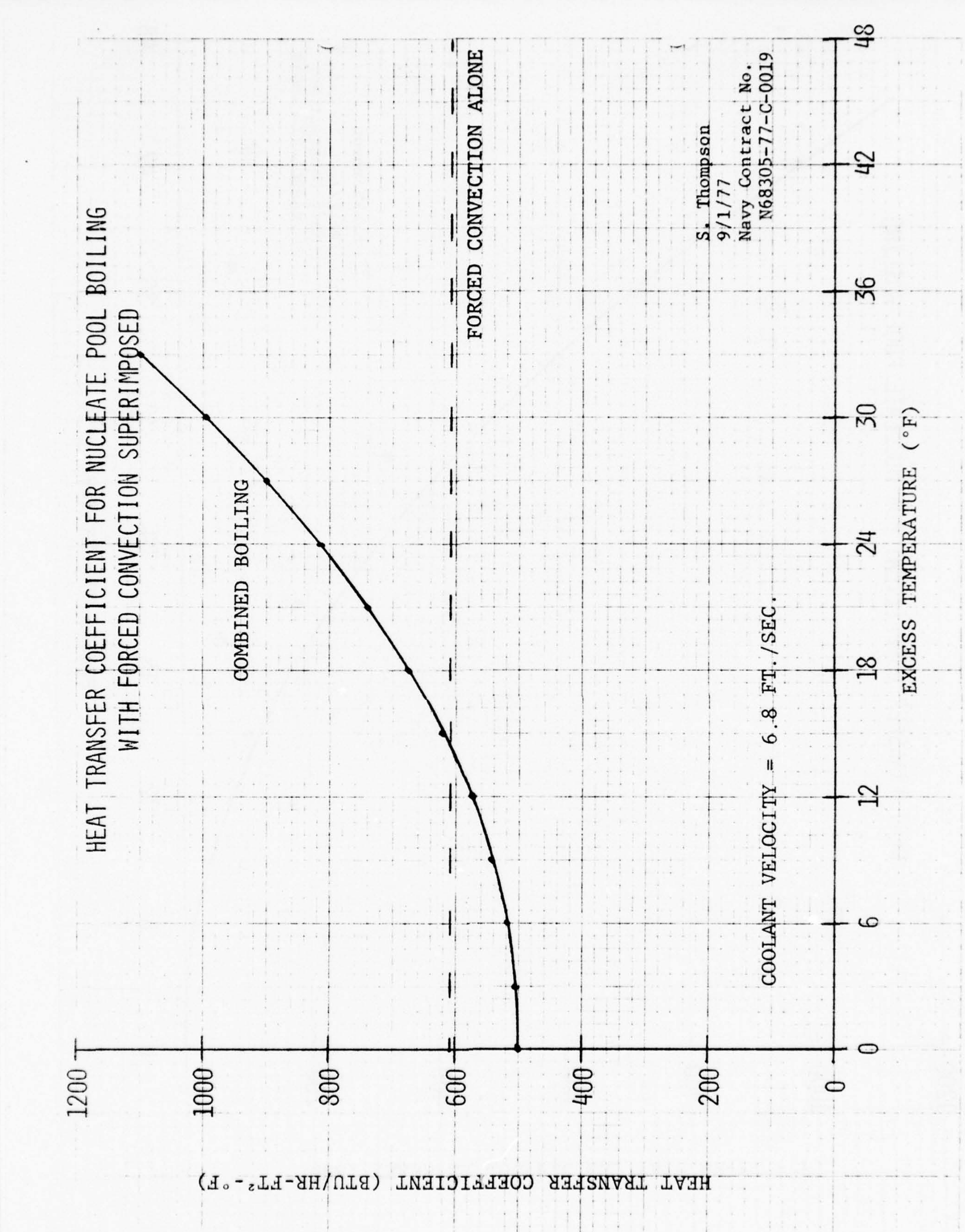

HEAT TRANSFER COEFFICIENT FOR NUCLEATE POOL BOILING
WITH FORCED CONVECTION SUPERIMPOSED
COMBINED BOILING
FORCED CONVECTION ALONE
COOLANT VELOCITY = 6.8 FT./SEC.
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019
HEAT TRANSFER COEFFICIENT (BTU/HR-FT²-°F)
EXCESS TEMPERATURE (°F)
0
200
400
600
800
1000
1200
0
6
12
18
24
30
36
42
48

FIGURE 14

An alternative to nucleate pool boiling is forced convection boiling. Forced convection boiling is not the same thing as pool boiling with a forced convection effect. Pool boiling with a forced convection effect is simply a flowing liquid undergoing nucleate pool boiling. Forced convection boiling is a term used to refer to a whole series of regimes of which nucleate boiling may play only a small part. Correlations for forced convection boiling only exist for specific fluids under specific conditions and in specific geometries. After a search of the available literature it was concluded that the best relationship was one of the forms proposed by Bo Pierre. (See Figure 15 and Appendix A - Heat Transfer Equations). This relationship is applicable to a regime where the inlet quality is greater than 15% and the outlet quality is less than 80%. Since the heat transfer coefficient increases with change in quality the optimum condition is for an outlet quality of 80%. In order to use this relationship, however, we must account for the process of raising the quality to 15%. The most likely mechanism in this region seems to be nucleate pool boiling, which reduces the overall heat transfer coefficient. From a combination of Bo Pierre's relationship and nucleate pool boiling an overall relationship was developed (Figure 16). This relationship also gave a lower overall heat transfer coefficient than forced convection sensible heat transfer, at least with the temperature difference in use in the freezer. From this it was concluded that the analytical results from boiling heat transfer would be poorer than those from sensible heat transfer. Experimental data would be necessary to further evaluate the potential of boiling heat transfer.

5.2 Coolant Selection.

Data was gathered on several refrigerants and/or coolants that could be used on the cooling side of the freezer. A tabulation of properties is attached (Table VII). A parameter $H_c/\Delta P$ was defined which indicates the relative efficiency of each fluid. Based on this, R-22 was selected as the most desirable refrigerant.

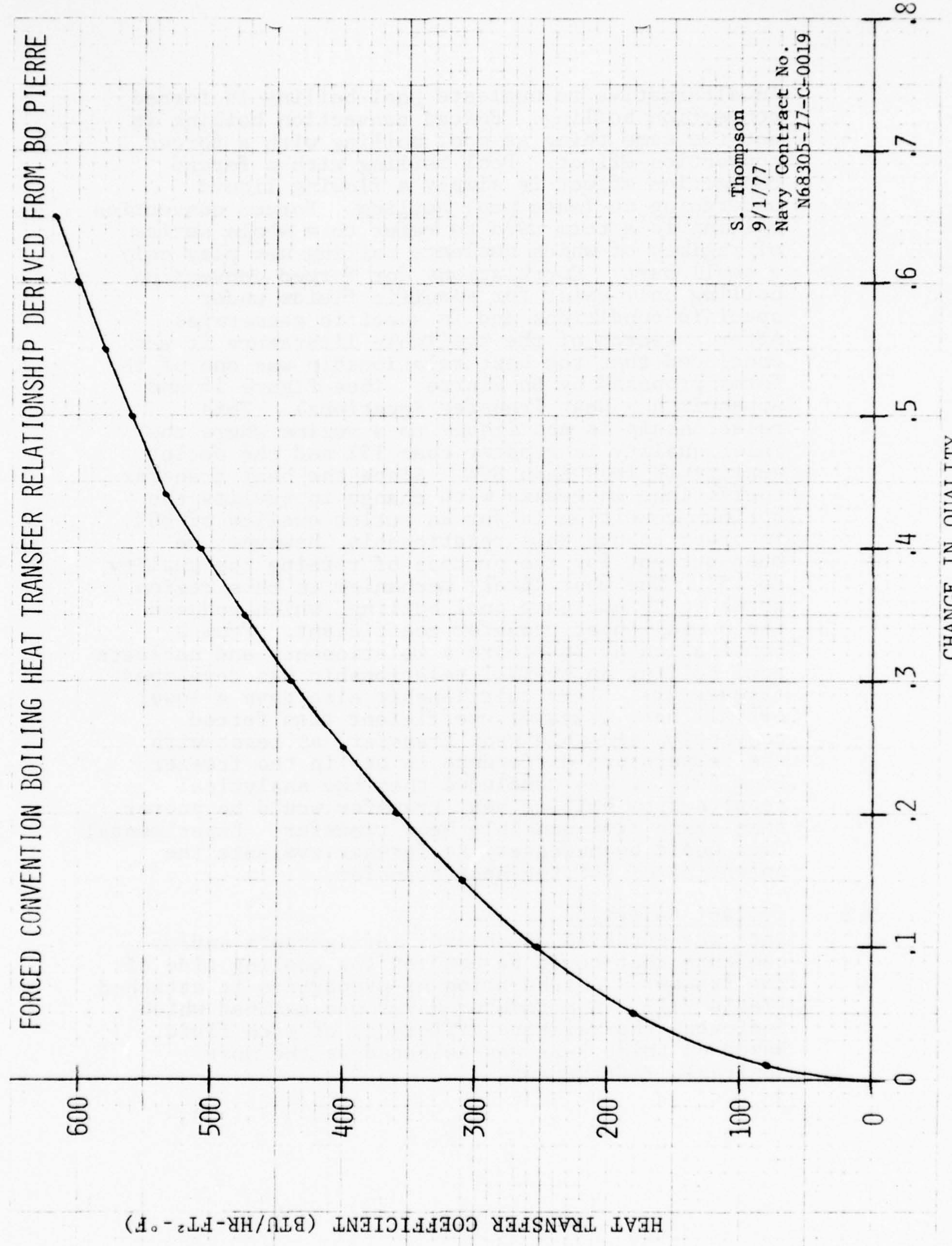

FORCED CONVENTION BOILING HEAT TRANSFER RELATIONSHIP DERIVED FROM BO PIERRE
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019
HEAT TRANSFER COEFFICIENT (BTU/HR-FT²-°F)
CHANGE IN QUALITY
0
100
200
300
400
500
600
0
.1
.2
.3
.4
.5
.6
.7
.8

FIGURE 15

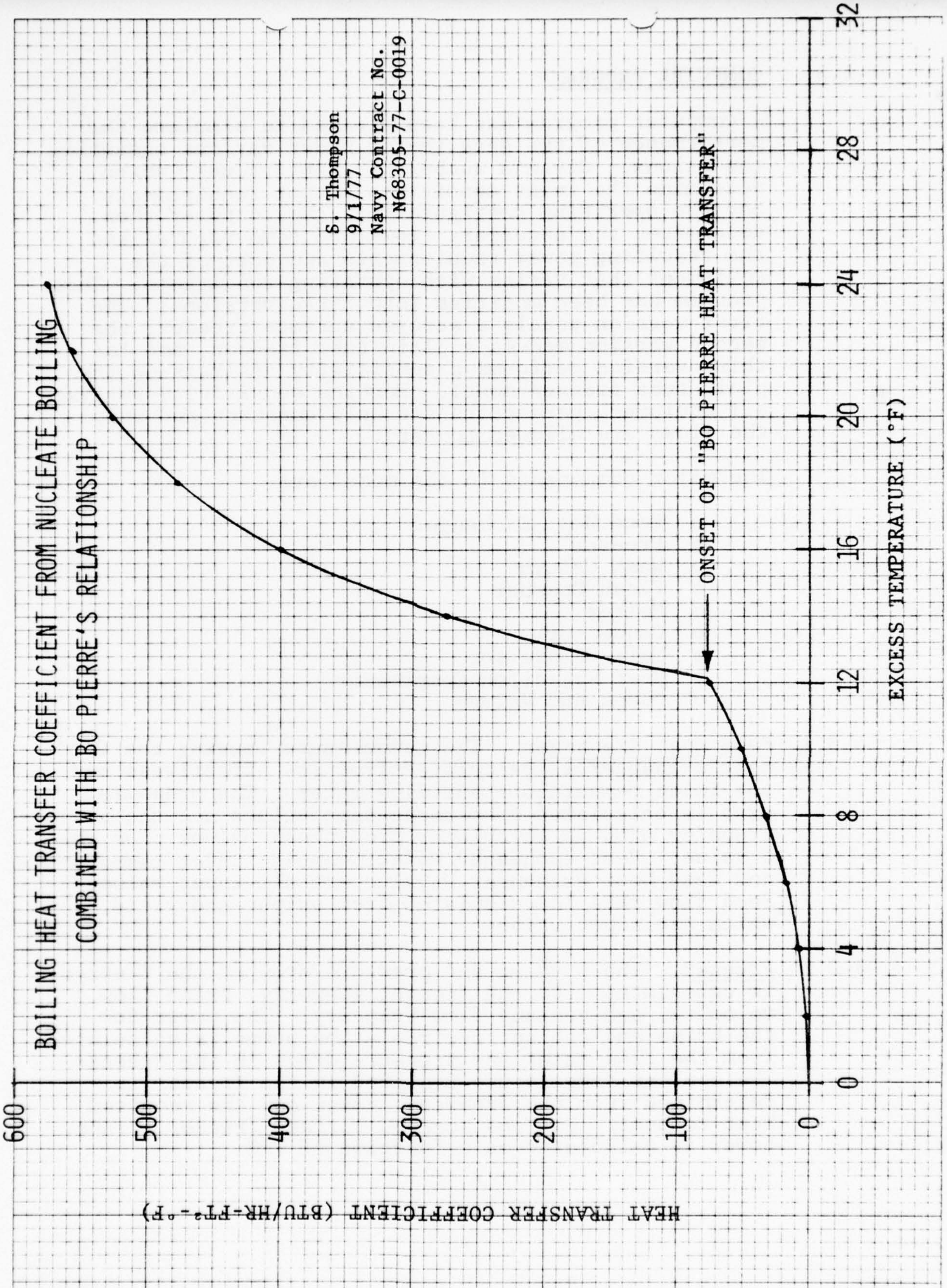

BOILING HEAT TRANSFER COEFFICIENT FROM NUCLEATE BOILING
COMBINED WITH BO PIERRE'S RELATIONSHIP
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019
ONSET OF "BO PIERRE HEAT TRANSFER"
EXCESS TEMPERATURE (°F)
HEAT TRANSFER COEFFICIENT (BTU/HR-FT²-°F)
0
4
8
12
16
20
24
28
32
0
100
200
300
400
500
600

FIGURE 16

5.3 COMPUTER PROGRAM

A computer program has been created to simulate the performance of the desalinator under various operating conditions. Given a number of input values, which determine the inlet conditions output requirements, and the operating configuration, the computer then determines the necessary flows and temperatures and the resulting power requirements and cost of the equipment that varies with operating conditions. This program has been modified into three versions. The first version models a scraped surface freezer, the second version models a plain tube freezer, and the third version performs a sensitivity analysis on the plain tube freezer model. For the sensitivity analysis all equipment parameters are held constant and only the flows are allowed to vary.

The major components in the model are:

- Feed Heat Exchanger
- Freezer
- Wash Column
- Flash Chamber
- Melter
- Condenser
- Main Compressor
- Heat Rejection Compressor
- Pumps and Valves

5.3.1 Feed Heat Exchanger. The temperature leaving the feed heat exchanger is calculated from its effectivess and the inlet temperature and flow rates. The pressure drop is calculated from these same values and the heat transfer area is calculated from these values plus the U value. Heat exchanger effectiveness, inlet temperature, and feed heat exchanger U value are all input values. Feed flow rate is calculated from output and yield, which are input values.

5.3.2 Freezer. The U value and pressure drop of the freezer is calculated from the freezer geometry, brine flow velocity, and coolant flow velocity. The power for the brine and coolant recirculation pumps is calculated from the flow rates and pressure drops in the freezer. The coolant temperature required in the freezer is determined from the freezer U value and the required heat removal. For the scraped surface freezer scraper power is calculated as a function of RPM.

For the scraped surface freezer the geometry consists of two concentric annuli. The outer annulus is the coolant chamber. The inner annulus, which is formed between the scraper shaft and the wall separating the inner and outer annulus, is the brine chamber. The scraper scrapes the inside of the wall separating the two annuli.

For the plain tube freezer the geometry is that of a shell and tube heat exchanger. Although in practice the shell side will be divided into six separate chambers the computer models a simple shell and tube exchanger. Results with a six chamber exchanger are expected to be similar.

Input values for freezer calculations are: geometry parameters, brine flow velocity, and coolant flow velocity (coolant flow rate in the plain tube version).

5.3.3 Wash Column. The wash column was not modeled except that input values were given for the brine and product water pressure drops through the column.

5.3.4 Flash Chamber. The temperature in the flash chamber is determined from the temperature of the coolant in the freezer and the heat from the coolant recirculation pump, which must be compensated for.

5.3.5 Melter. The U value of the melter is calculated from the film coefficient of the freon (shell) side, which is an input value, the calculated tube wall resistance, and the water (tube) side film coefficient, which is calculated from the water velocity and the tube ID.

The heat transfer area of the melter is calculated from the heat to be removed by the water, the freon temperature, the water exit temperature, and the U value of the melter.

The pressure drop is calculated from the water velocity and the tube geometry.

Input values are: shell side film coefficient, tube ID, water velocity, freon temperature, and water exit temperature.

5.3.6 Condenser. The U value of the condenser is calculated from the film coefficient of the freon (shell) side, which is an input value, the calculated tube wall resistance, and the brine (tube) film coefficient, which is calculated from the brine velocity and the tube ID.

The heat transfer area of the condenser is calculated from the U value, the quantity of heat to be transferred, the freon temperature, the brine inlet temperature, and the brine exit temperature.

The pressure drop is calculated from the condenser geometry and the brine velocity.

Input values are: shell side film coefficient, brine velocity, tube ID, freon temperature, and approach temperature of brine in condenser.

5.3.7 Main Compressor. The flow required for the main compressor is calculated from the heat to be removed from the flash chamber and the flash chamber and melter temperatures. The power of the main compressor is calculated from the flow and the flash chamber and melter freon pressures.

5.3.8 Heat Rejection Compressor. The power for the heat rejection compressor is calculated from the flow and the freon temperatures in the melter and condenser. The heat to be removed by the compressor from the melter is determined from the heat removed from the flash chamber, the heat produced by the main compressor, and the heat removed by the water in the melter. The flow required for the heat rejection compressor is calculated from the heat to be removed from the melter by the compressor, the temperature of the freon in the melter, and the temperature of the freon in the condenser.

5.3.9 Pumps and Valves. The power for the pumps is determined by their flows and the pressure drops they must pump through. Input values are given for valve size, and the pressure drop through the valves is accounted for in determining pressure drops for pumps. Input values are given for the efficiency of the pumps and compressors.

The above explanations have been given to give a general idea of the operation of the computer program. They are not rigorous explanations and many details of the program have been omitted.

5.3.10 Present Value Analysis. The computer program determines costing, as well as performance, of the components that change with operating conditions. This permits the design to be optimized for cost as well as performance. The following is a list of the components which were included in the present value analysis, along with the form of their costing equations:

Feed Heat Exchanger	$K_1 + K_2 * Area$
Freezer	$K_2 * Area$
Melter	$K_2 * Area$
Condenser	$K_2 * Area$
Pumps	$K_1 + K_3 * Hp$
Compressors	$K_1 + K_3 * Hp + K_4 * Cfm$

K_1 = Fixed cost for component
K_2 = Cost per unit area of component
K_3 = Cost per horsepower of component
K_4 = Cost per Cfm of component

The present value analysis was programmed with an assumed equipment life of 10 years, a 6% annual real cost growth for electrical power and an initial value for electrical costs of \$.035 per kw-hr. All values assumed for the present value analysis are at the end of Table VIII.

TABLE VII

HEAT TRANSFER PROPERTIES OF FLUIDS

at $20^{o}F$ 15 Ft/sec Flow Velocity

FLUID	$h_c/\Delta P_m$ (1)	h_c	Re (2)	SPECIFIC HEAT	VISCOSITY	THERMAL CONDUCTIVITY	DENSITY
Refrigerant 114	.612	594	353,600	.222	1.23	.042	96.63
Refrigerant 12	.889	784	587,800	.220	.678	.0468	88.53
Refrigerant 22	1.216	988	613,200	.276	.599	.0598	81.60
30% Ethylene Glycol	1.105	869	25,270	.93	11.62	.28	65.24
30% Dowfrost	.721	593	12,370	.92	23.47	.263	64.49

(1) h_c = Heat Transfer Coefficient ($BTU/hr\text{-}Ft^2\text{-}^oF$)

ΔP = Pressure Drop (lb/Ft^2)

(2) Re = For a typical geometry and 15 Ft/sec flow velocity.

TABLE VIII

INPUT VALUES FOR INDIRECT CONTACT FREEZE DESALINATOR PROGRAM

VARIABLE	VALUE	RANGE
Outside Diameter of Brine Chamber	4.00 in.	
Thickness of Wall Separating Brine and Coolant Chambers	.148 in.	0.625-.250
Brine Chamber Annulus Thickness	.250 in.	.125-.825
Number of Scrapers per Freezer Unit	2.00	
Scraper Thickness	.125 in.	
Coolant Chamber Annulus Thickness	.250 in.	.125-.5
Roughness Height of Coolant Chamber Walls	.0018 in.	
Roughness Height of Brine Chamber Outside Wall	10^{-7} in.	
Roughness Height of Shaft and Scraper Surfaces	250 x 10^{-6} in.	
Length of Freezer	9.0 ft.	
Number of Freezers	18	12-20
Scraper Force per ft. of Scraper Length	60.0 lbs.	
Scraper Speed	60.0 rpm.	
Thermal Conductivity of Freezer Wall	35.0 (BTU/hr.-ft.-°F)	8-40
Flow Fraction of Valves	.600	
Inside Diameter of Melter Tubes	.745 in.	
Roughness Height of Mleter Tubes	250. x 10^{-6} in.	
Effectiveness of Feed Heat Exchanger	.900	.800-.940
Velocity of Brine in Freezer	20.0 ft./sec.	2-52
Inlet Temperature	68.0°F	40-100
Fraction of Wash Column Output Used for Washing Ice	.050	
Temperature of Water Coming out of Melter	34.0°F	32.2-34.0
Ice Fraction Leaving Freezer	.150	
Ice Fraction in Slurry to Melter	.050	
Salinity at Inlet	.035	.005-.050
Velocity of Coolant in Freezer	7.0	2-20
Temperature of Coolant in Melter	41.0	36.0-44.0
Approach Temperature of Brine in Condenser	5.0°F	2.0-12.0
Pressure Drop through Orifice at Flash Chamber Inlet from Freezer	2.0 psi	0.0-2.0
Combined Pressure Drop through Cyclone and Filter	15.0 psi	
Pressure Drop of Brine through Wash Column	5.00 psi	
Capacity of Desalinator	20,000 gpd	
Yield of Desalinator	.500	.3-.8
Average Pressure Drop of Pipes	.500	
Pressure Drop through Storage Tank	0.0	
Pressure Drop of Product Water through Wash Column	0.0	
Thermal Conductivity of Melter Tubes	8.00	
Thermal Conductivity of Condenser Tubes	9.40	
Fouling Factor for Melter and Condenser Tubes	.0005	
Outside/Inside Area Ratio for Melter	3.07	
Outside/Inside Area Ratio for Condenser	3.38	
Velocity of Water in Melter Tubes	5.50 fps	4.0-12.0
Wall Thickness of Melter Tubes	.065	
Wall Thickness of Condenser	.065	
ID of Condenser Tubes	.495	
Roughness Height of Condenser Tubes	250 * 10^{-6}	
Velocity of Brine in Condenser	10.0 fps	4.0-16.0
Temperature Rise of Brine in Condenser	11.0°F	8.0-14.0

TABLE VIII

INPUT VALUES FOR INDIRECT CONTACT FREEZE DESALINATOR PROGRAM
(Continued)

VARIABLE	VALUE	RANGE
Number of Velccity Heads for Brine Side of Freezer	2.00	
Number of Velocity Heads for Coolant Side of Freezer	2.00	
Coefficient of Friction of Scraper against Wall	.300	
C_V Value for Feed Valve	9.00	
C_V Value for Brine Slurry Valve	29.0	
C_V Value for Concentrate Outlet Valve	2.00	
C_V Value for Melt Outlet Valve	9.00	
C_V Value for Wash Valve	2.00	
C_V Value for Main Coolant Valve	60.0	
Efficiency of Feed Pump	.600	
Efficiency of Brine Recirculation Pump	.800	
Efficiency of Concentrate Pump	.600	
Efficiency of Melt Slurry Pump	.600	
Efficiency of Melt Pump	.600	
Efficiency of Coolant Recirculation Pump	.750	
Efficiency of Main Compressor	.610	
Efficiency of Heat Rejection Compressor	.610	
Film Coefficient of Freon Side of Melter	220.	
Film Coefficient of Freon Side of Condenser	220.	
U Value of Feed Heat Exchanger	475.	
COSTING INPUTS		
Feed Heat Exchanger $/ft.²	37.00	
Feed Heat Exchanger Fixed $	1,500.00	
Pump Fixed $	1,833.00	
Pump $/Hp	203.50	
Freezer $/ft.²	750.00	
Melter $/ft.²	20.00	
Condenser $/ft.²	20.00	
Compressor Fixed $	1,660.00	
Compressor $/cfm	30.00	
Compressor $/Hp	50.00	
Project Lead Time	1.00 yr.	
Economic Life	10.0 yr.	
Differential Inflation Rate for Power	.06	
Days per Year	365	
Hours per Day	20	
Initial Cost per Kilowatt of Power	$.035	

6.0 RESULTS OF ANALYSIS

Optimization runs were made for both the scraped surface heat exchanger system and the defrost cycle system. A few of the more significant runs are presented in the next two sections. The program had the capability of varying all combinations of up to six parameters at a time. Normally, only the lowest cost combination was retained for use. However, some of the more significant variables were printed out and these are the ones presented in this section. Lowest present value cost for power plus capital cost for pumps, compressors and heat transfer surfaces were the only cost elements calculated for these optimization runs. Therefore, the present value cost listed on these curves only represent these items and only serve as a reference value.

6.1 Scraped Surface System.

One of the more important parameters is ΔT_b (brine to wall temperature difference). ΔT_b will be an important parameter influencing whether ice will freeze to the walls of the freezer. At high ΔT_b's the brine would have to be highly subcooled in order not to freeze on the walls. In a turbulent flow stream only relatively small amounts of subcooling are likely to be present. Therefore, small ΔT_b's are required. Figure 17 shows the relationship between ΔT_b and brine velocity in the freezer. Above approximately 44 ft./sec. the heat input, due to pump work, becomes so significant that lower ΔT's are not obtainable. Velocities above 20 to 28 ft./sec. produce diminishing returns in reducing ΔT_b. A value of 24 fps has been chosen as the design value. Whether the ΔT_b associated with this velocity is sufficiently low is unknown until tests are conducted. If a lower ΔT is required this would indicate a greater area (or lower capacity) is required.

As the number of freezer units are reduced the ΔT_b becomes larger at a given volocity, because the heat flux is increasing (for the same capacity). This is shown in Figure 18.

Figure 19 shows although 24 fps brine velocity is not optimum, the costs are not strongly affected by using such a high velocity.

Figure 20, Cost and Horsepower vs. Number of Freezer Units, is presented to show, that considering cost only, the optimum number of freezer units is 12. However, because of ΔT_b considerations more units are being used. Although this set of calculations is for 24 freezers, 18 were used in the design because of packaging considerations.

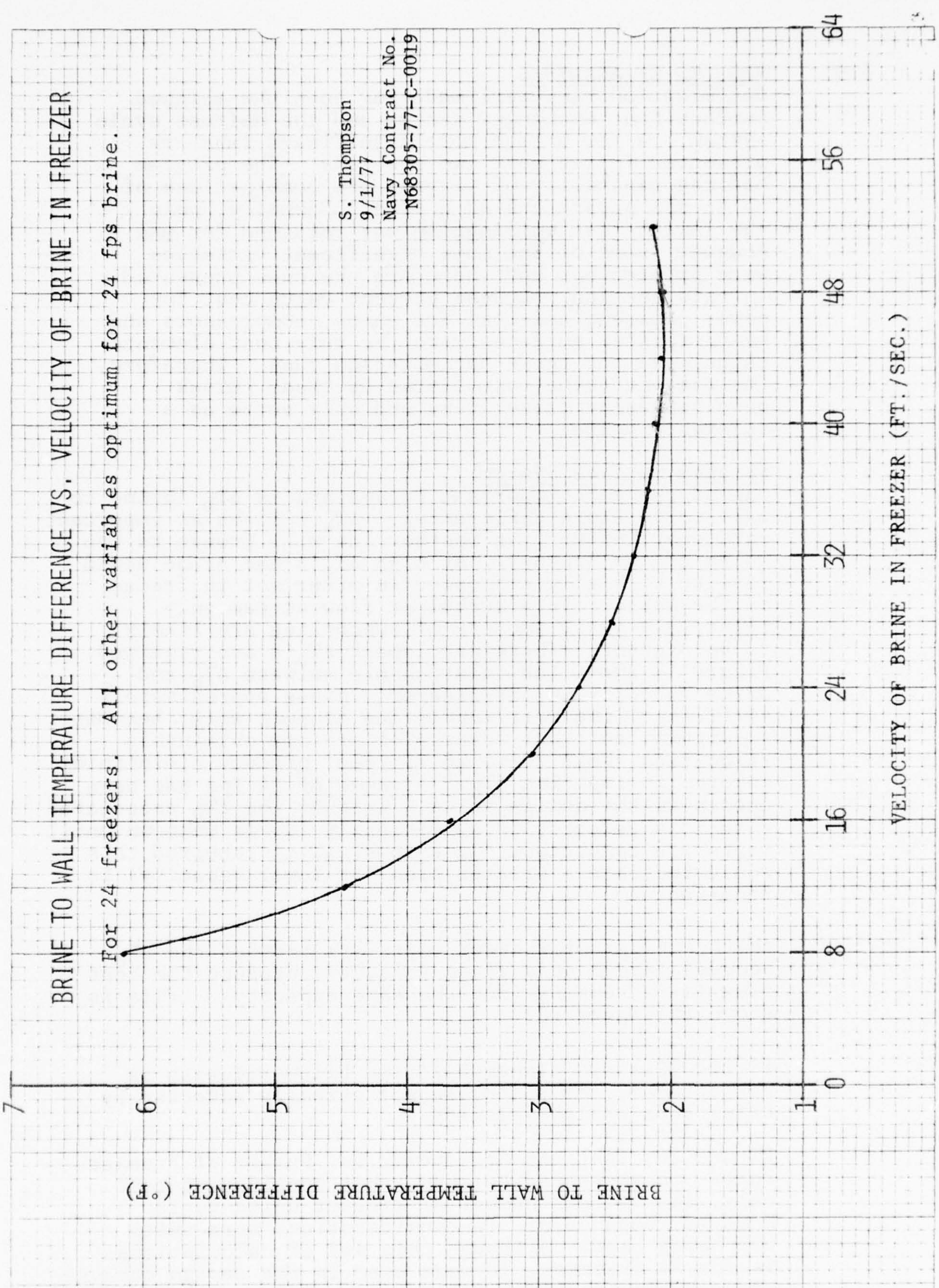

BRINE TO WALL TEMPERATURE DIFFERENCE VS. VELOCITY OF BRINE IN FREEZER
For 24 freezers. All other variables optimum for 24 fps brine.
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019
BRINE TO WALL TEMPERATURE DIFFERENCE (°F)
VELOCITY OF BRINE IN FREEZER (FT./SEC.)
7
6
5
4
3
2
1
0
8
16
24
32
40
48
56
64

FIGURE 17

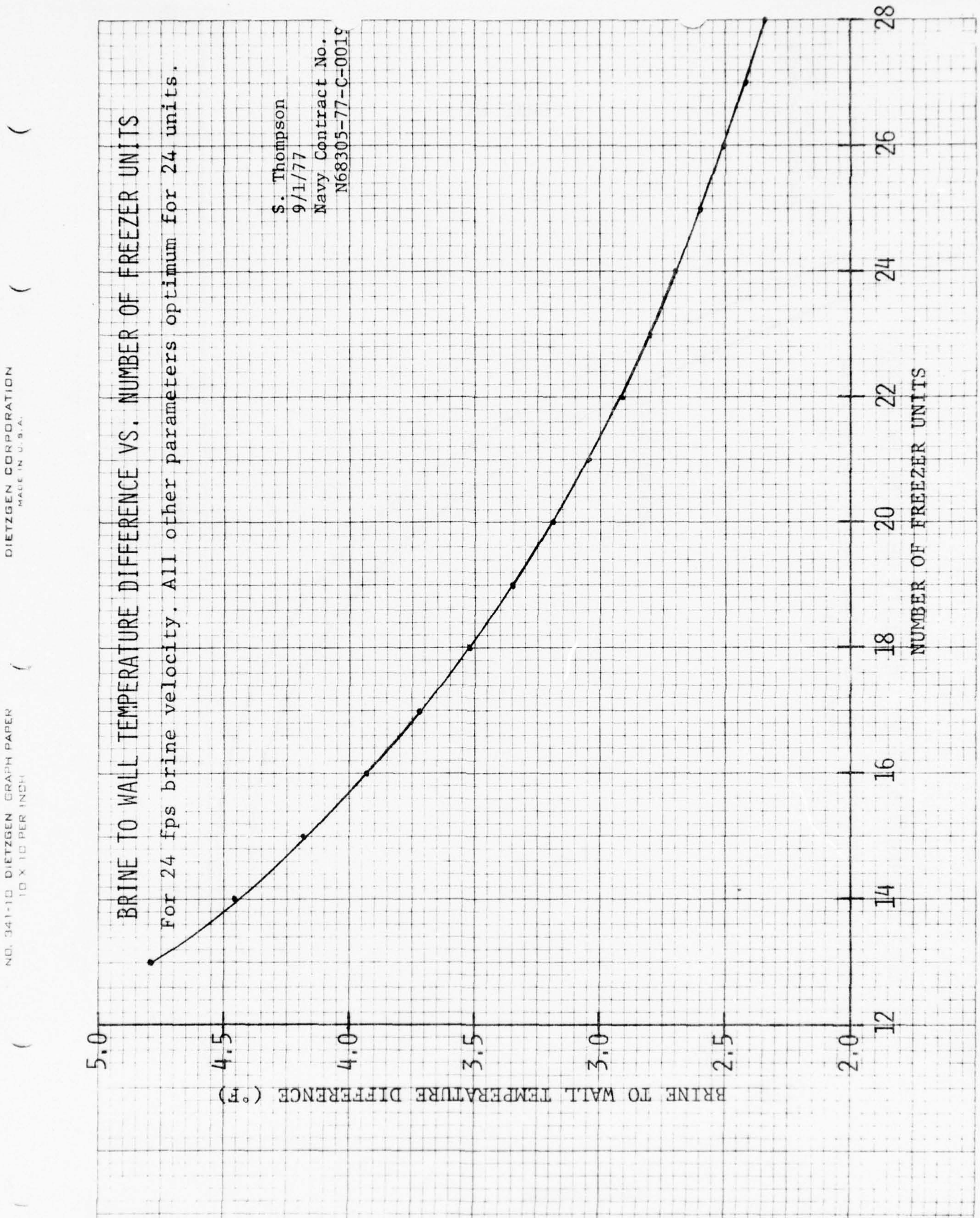

NO. 341-10 DIETZGEN GRAPH PAPER
10 X 10 PER INCH
DIETZGEN CORPORATION
MADE IN U.S.A.
BRINE TO WALL TEMPERATURE DIFFERENCE VS. NUMBER OF FREEZER UNITS
For 24 fps brine velocity. All other parameters optimum for 24 units.
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019
BRINE TO WALL TEMPERATURE DIFFERENCE (°F)
5.0
4.5
4.0
3.5
3.0
2.5
2.0
NUMBER OF FREEZER UNITS
12
14
16
18
20
22
24
26
28

FIGURE 18

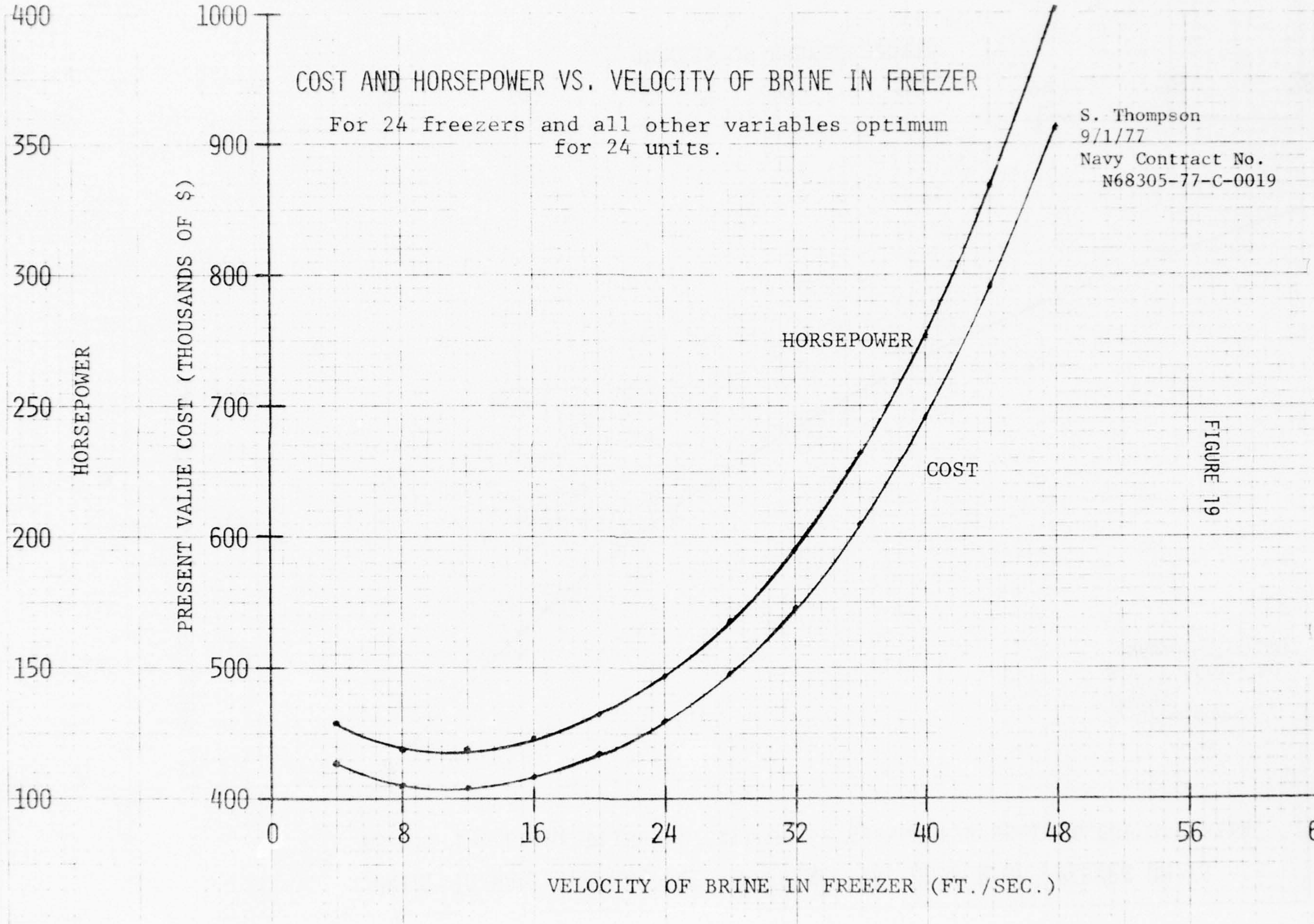
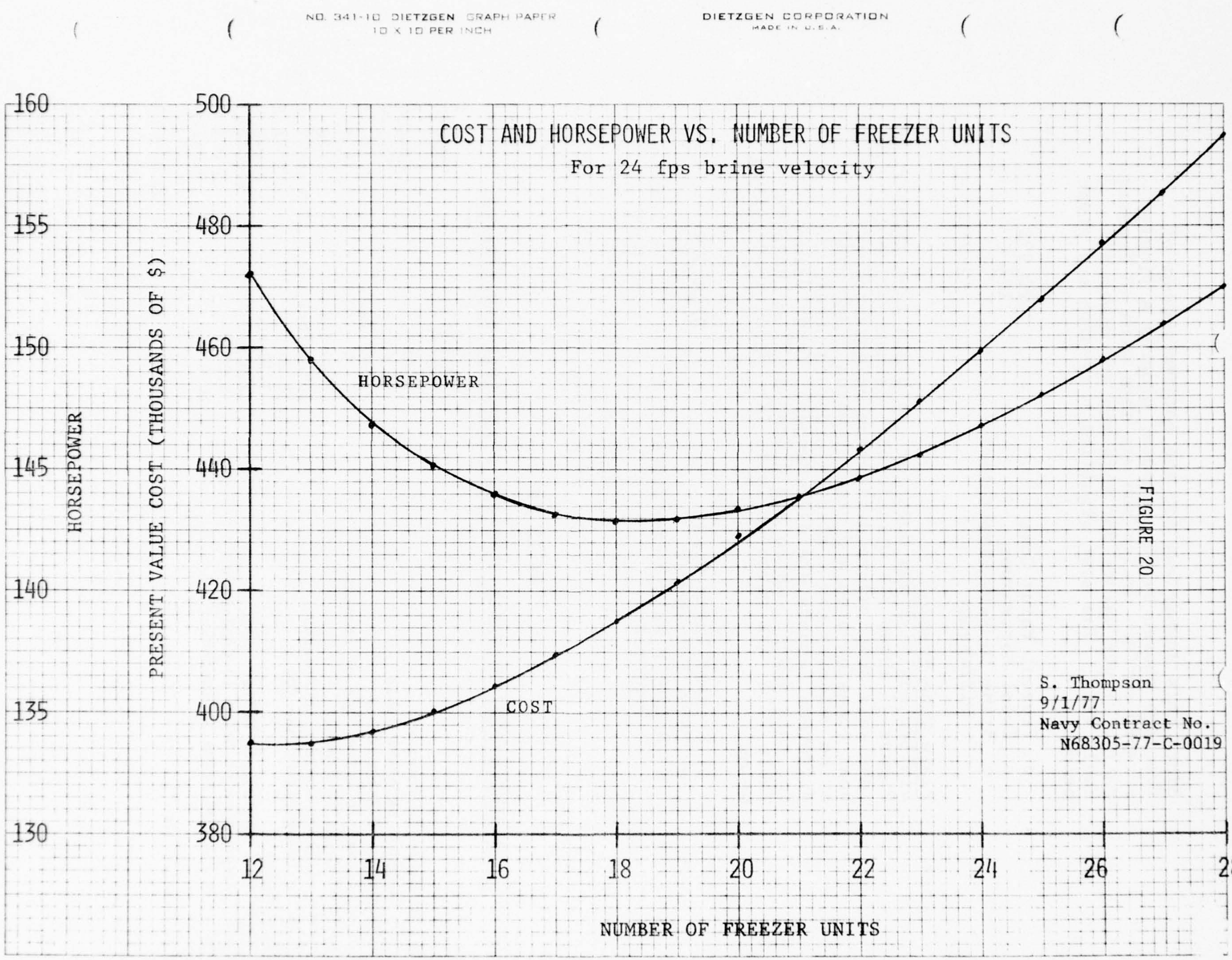

COST AND HORSEPOWER VS. VELOCITY OF BRINE IN FREEZER
For 24 freezers and all other variables optimum for 24 units.
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019
HORSEPOWER
COST
HORSEPOWER
PRESENT VALUE COST (THOUSANDS OF $)
VELOCITY OF BRINE IN FREEZER (FT./SEC.)
400
350
300
250
200
150
100
1000
900
800
700
600
500
400
0
8
16
24
32
40
48
56
64

FIGURE 19


NO. 341-10 DIETZGEN GRAPH PAPER
10 X 10 PER INCH
DIETZGEN CORPORATION
MADE IN U.S.A.
COST AND HORSEPOWER VS. NUMBER OF FREEZER UNITS
For 24 fps brine velocity
HORSEPOWER
PRESENT VALUE COST (THOUSANDS OF $)
160
155
150
145
140
135
130
500
480
460
440
420
400
380
HORSEPOWER
COST
12
14
16
18
20
22
24
26
28
NUMBER OF FREEZER UNITS
FIGURE 20
S. Thompson
9/1/77
Navy Contract No.
N68305-77-C-0019

Since this number was not the optimum number of freezers chosen by the computer program a final scraped surface design run was made with 18 freezer units. Input values for this run are given in Table VIII. The most significant results are given below:

o Present Value Cost	\$471,976
o Total Horsepower	174.6
o Brine to Wall Temperature Difference	42.7°F

6.2 Plain Tube Freezer

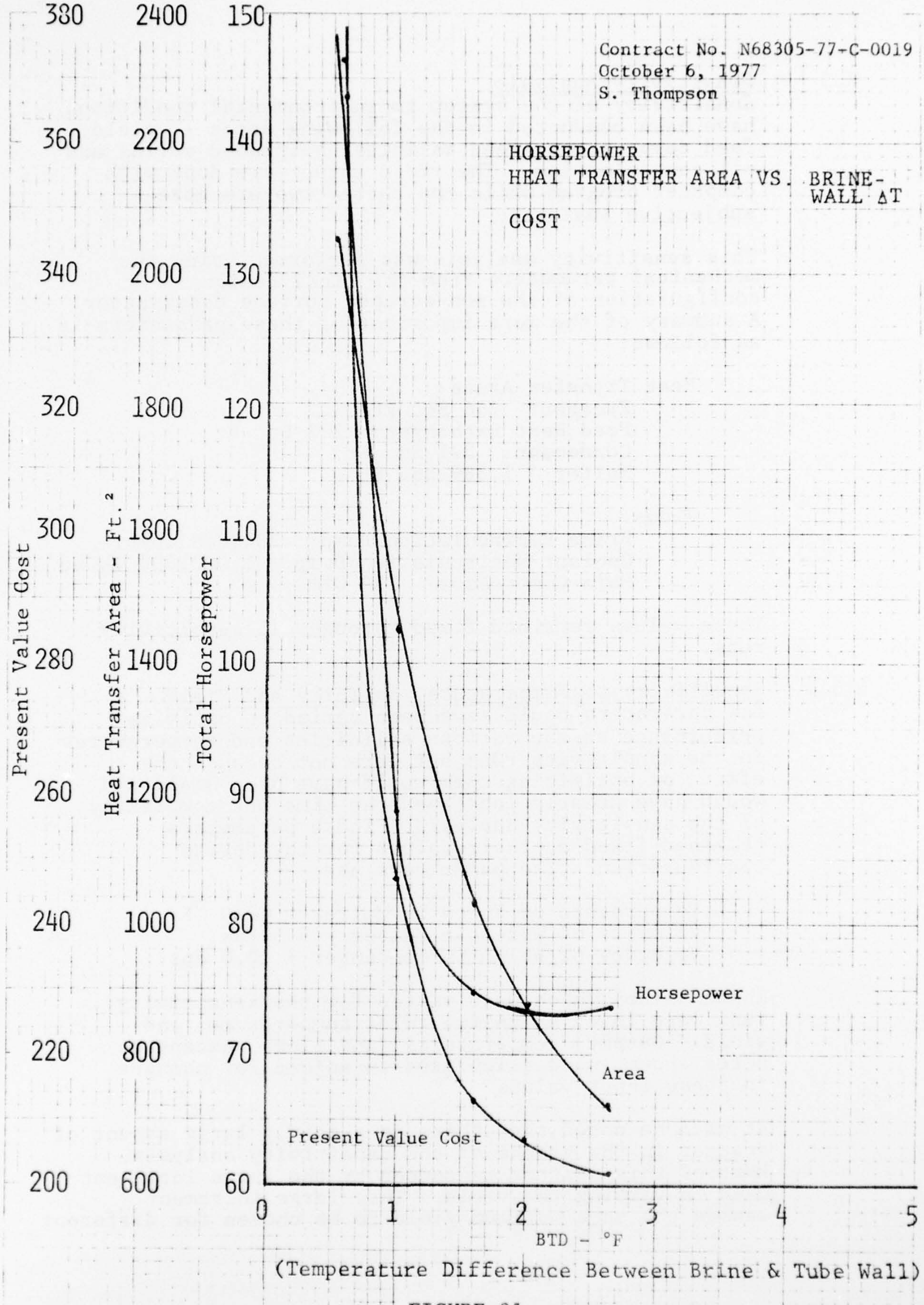
Similar analysis was done for the plain tube freezer. Figure 21 shows the relationship between brine to tube wall temperature difference on horsepower, heat transfer area and present value cost. The brine velocity was varied in order to obtain the different temperature differences. Input values for this design, which are different from those in Table VIII, are given in Table IX. Results are given below:

o Present Value Cost	\$208,329
o Total Horsepower	76.1
o Brine to Wall Temperature Difference	2.00°F

TABLE IX

MODIFICATIONS TO TABLE VIII FOR PLAIN TUBE DESALINATOR

	VALUE	RANGE
Number of Tubes in Freezer	370	350-480
Outside Diameter of Brine Tubes	1 in.	.75-1.50
Wall Thickness of Brine Tubes	.049 in.	
Number of Passes in Brine Side of Freezer	8	1-12
Velocity of Brine in Freezer	7.00 fps	6-8
Flow Rate of Freon Coolant in Freezer	10^6 lb./hr.	.7 * 10^6-1.3 * 10^6
Pitch Ratio (Tube Pitch/Tube OD)	1.20	1.05-1.30
Baffle Spacing	2.10 ft.	1.0-2.6
Temperature of Coolant in Melter	42.0°F	41.0-43.0
Approach Temperature of Brine in Condenser	5.20°F	5.0-5.4
Temperature Rise of Brine in Condenser	10.0°F	9.0-11.0
Freezer Cost (\$/ft.^{2})	30.00	30.00-50.00



Contract No. N68305-77-C-0019
October 6, 1977
S. Thompson
HORSEPOWER
HEAT TRANSFER AREA VS. BRINE-WALL ΔT
COST
Present Value Cost
Heat Transfer Area - Ft.^{2}
Total Horsepower
380 360 340 320 300 280 260 240 220 200
2400 2200 2000 1800 1800 1400 1200 1000 800 600
150 140 130 120 110 100 90 80 70 60
Horsepower
Area
Present Value Cost
0 1 2 3 4 5
BTD - °F
(Temperature Difference Between Brine & Tube Wall)

FIGURE 21

6.3 Sensitivity Analysis.

Sensitivity of the system to environmental conditions have been conducted in the following areas -- yield, feed temperature, feed salinity, suspended solids and oil contamination. The first three were done with computer program while the latter two are more subjective analyses.

This sensitivity analysis was performed using the mechanical parameters from the final design configuration of the non-scraped surface desalinator. A summary of the more important of these parameters is as follows:

Heat Transfer Areas:
- Freezer: 786 Sq. Ft.
- Feed Heat Exchanger: 232 Sq. Ft.
- Condenser: 271 Sq. Ft.
- Melter: 1,389 Sq. Ft.

Flows:
- Brine Recirculation Pump: 284,700 lb/hr
- Coolant Recirculation Pump: 1,000,000 lb/hr
- Main Compressor: 239 Cfm

These values remained fixed through all sensitivity runs.

6.3.1 Sensitivity to Temperature, Salinity and Yield. A few parameters could have been varied to optimize performance at the various salinities and temperatures in the sensitivity runs but were not because the effect of optimizing them would have been small and would have greatly increased the time and complexity of the sensitivity analysis. These parameters remained fixed at their values for the design configuration. The parameters are:

Temperature of Freon in Melter = 42.0 ^{o}F
Velocity of Water in Melter = 5.5 Fps.
Velocity of Brine in Condenser = 10.0 Fps.

Input values which were varied for the sensitivity runs were inlet salinity, inlet temperature, and yield. Output, temperatures, and flows, except as noted above, were calculated to adjust for changes in these input values.

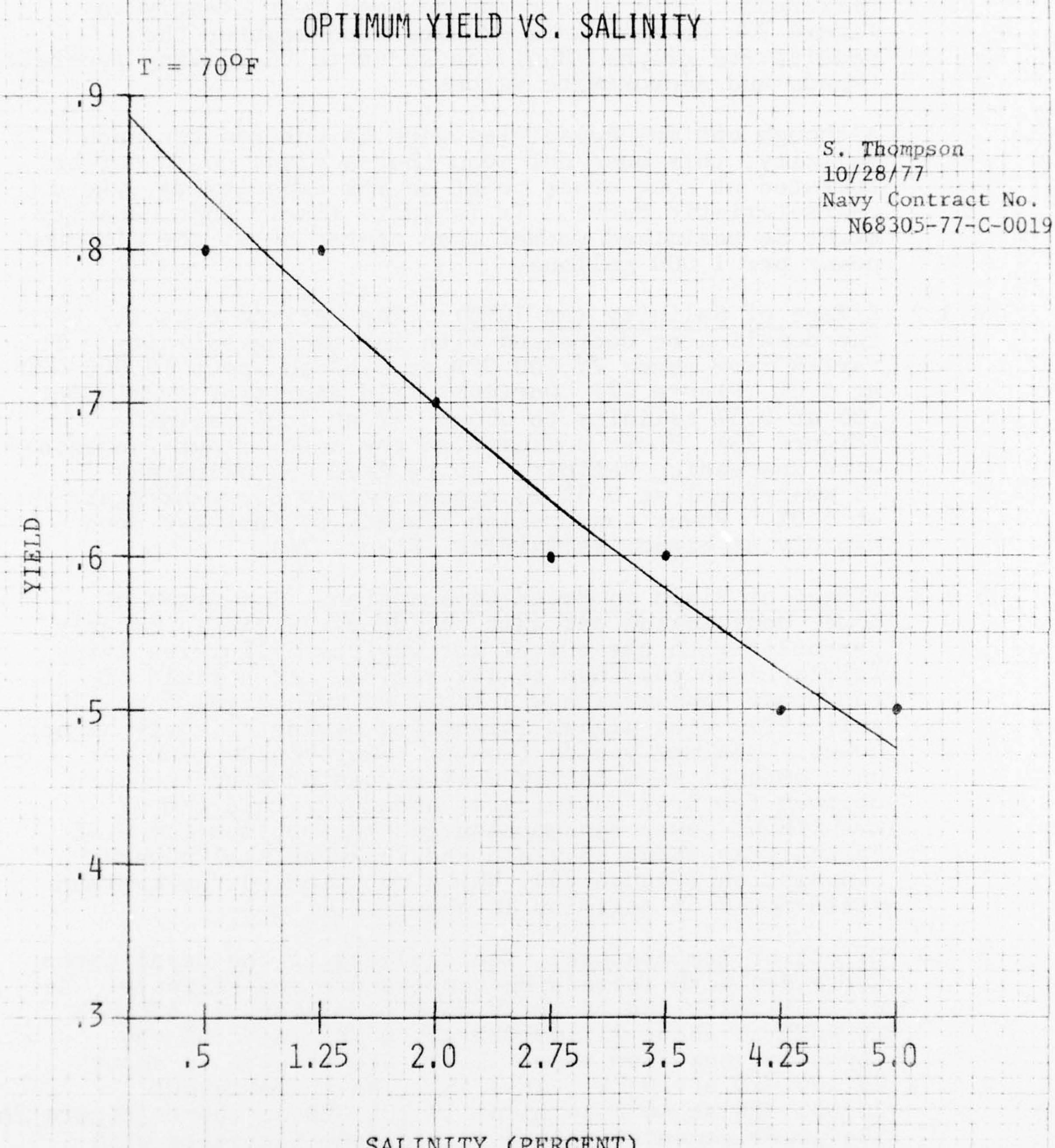
It will be noted that there is a fairly large amount of scatter in the graphs of the sensitivity analysis. Part of this scatter is caused by the large increment used in varying the yield. This large increment causes the same optimum yield to be chosen for different sets of conditions, as can be seen in Figure 22. Another cause for the scatter in these graphs is the convergence factor used in calculating desalinator capacity This results in the same value for capacity occurring in cases where the output value should be slightly different. This effect can be seen most clearly in Figure 23. Despite this problem with scatter the sensitivity graphs should be accurate in showing general trends and approximate values.

Although optimum capacities were calculated for each salinity, this does not mean that a plant could not be operated at some other point on its capacity vs. yield characteristic curve. It is quite likely that capacity might be maximized rather than operating at the minimum power per 1,000 gallons.

6.3.2 Effect of Salinity (at 70°F). The optimum capacity of the desalinator decreases with increasing salinity. It varies from about 22,000 GPD at 0.5% to about 19,000 GPD at 5.0% (Figure 22). Optimum yield decreases from about .85 at 0.5% salinity to about .47 at 5.0% salinity (Figure 23). Power consumption per unit output increases with increasing salinity, going from approximately 73 kw-hr/kgal at 0.5% to approximately 81 kw-hr/kgal at 5.0%. Power consumption appears to increase more rapidly at higher salinities (Figure 24).

6.3.3 Effect of Yield on Power Consumption. The graph of power consumption per unit output vs. yield is roughly parabolic with the curves for various salinities lying fairly close together at low yields. At a yield of .3 the power consumption was about 90 kw-hr/kgal for all salinities with higher salinities having slightly higher power. Optimum yields (lowest power consumption per unit output) were fairly well separated with the optimum for 3.5% salinity at about .6 yield and 77 kw-hr/kgal power consumption and the optimum for 0.5% salinity at about .8 yield and 73 kw-hr/kgal power consumption (Figure 25). While not plotted the optimum yield for 2.0% salinity is about .7.

6.3.4 Effect of Temperature. The capacity of the desalinator decreases with increasing temperature and salinity. At 0.5% salinity the capacity goes from about 22,600 GPD at 40°F to about 21,700 GPD at 100°F. At 2.0% the capacity goes from about 21,600 GPD at 40°F to about 20,400 GPD at 100°F. At 3.5% it goes from about 20,600 GPD at 40°F to about 19,100 GPD at 100°F (Figure 26). The power consumption per unit output increases with both temperature and salinity, but it is a stronger function of temperature than of salinity. At 0.5%


OPTIMUM YIELD VS. SALINITY
T = 70°F
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N68305-77-C-0019
YIELD
.9
.8
.7
.6
.5
.4
.3
.5
1.25
2.0
2.75
3.5
4.25
5.0
SALINITY (PERCENT)

FIGURE 22

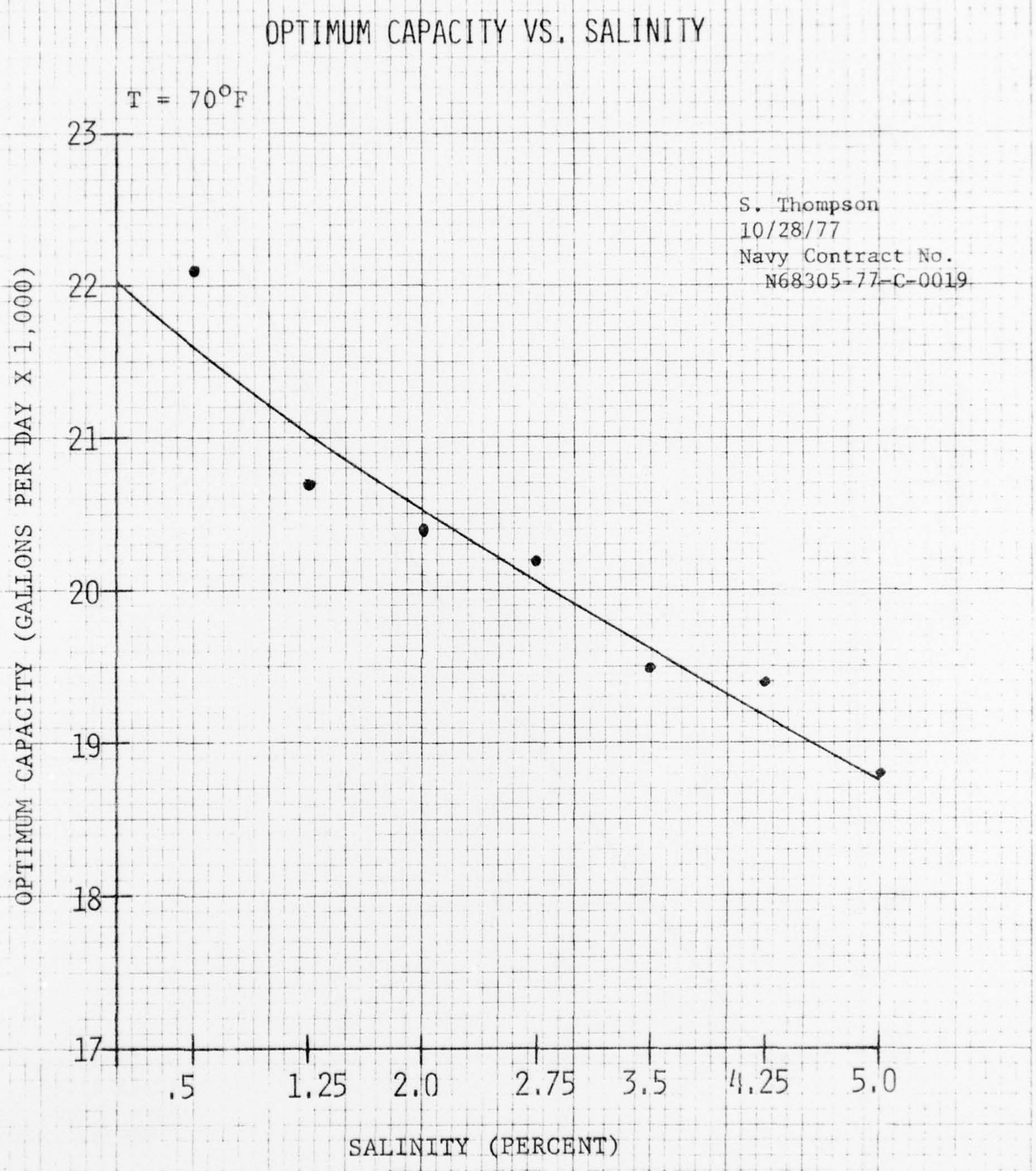
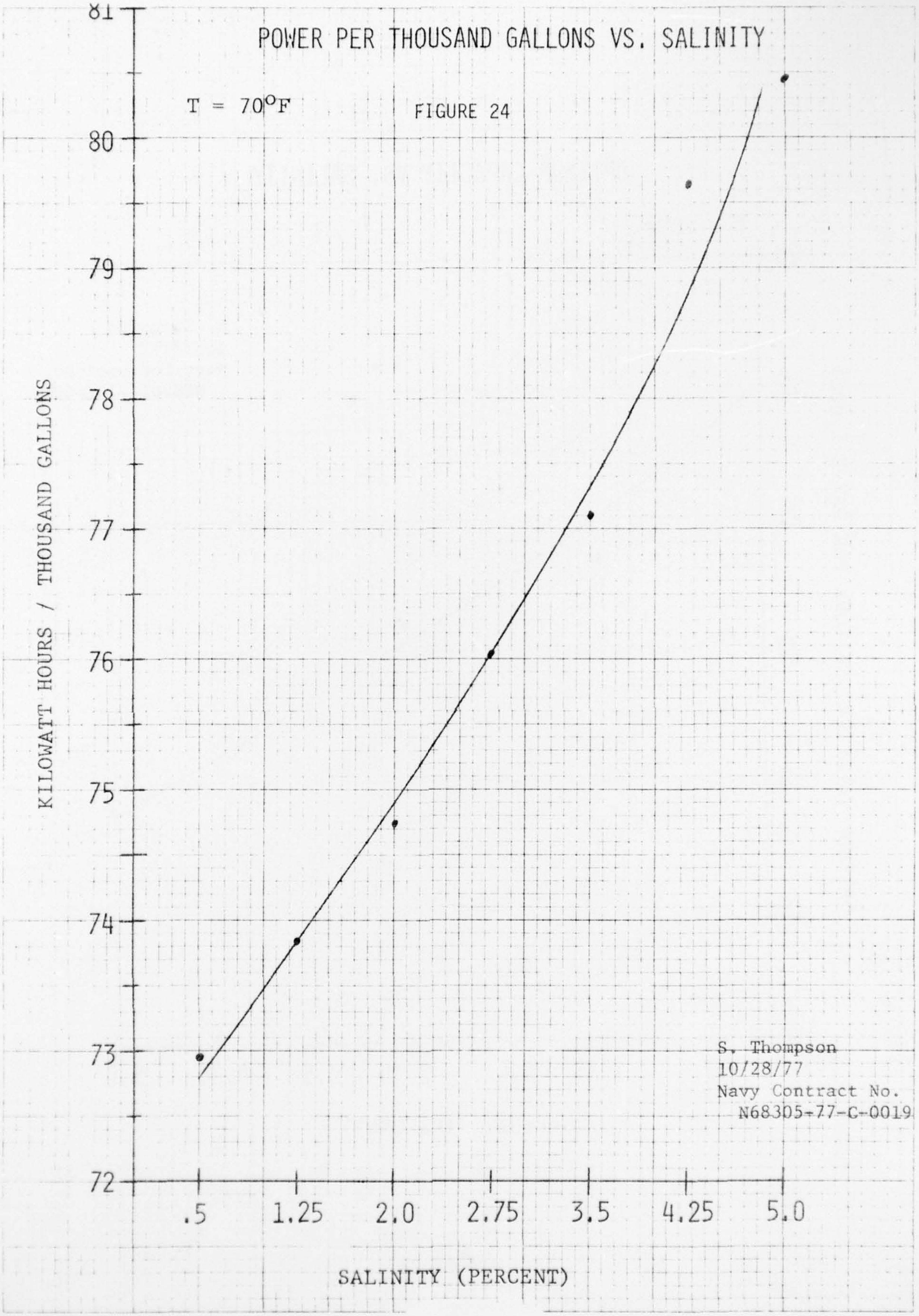

OPTIMUM CAPACITY VS. SALINITY
T = 70°F
S. Thompson
10/28/77
Navy Contract No.
N68305-77-C-0019
OPTIMUM CAPACITY (GALLONS PER DAY X 1,000)
23
22
21
20
19
18
17
.5
1.25
2.0
2.75
3.5
4.25
5.0
SALINITY (PERCENT)

FIGURE 23


POWER PER THOUSAND GALLONS VS. SALINITY
T = 70°F
FIGURE 24
KILOWATT HOURS / THOUSAND GALLONS
81
80
79
78
77
76
75
74
73
72
.5
1.25
2.0
2.75
3.5
4.25
5.0
SALINITY (PERCENT)
S. Thompson
10/28/77
Navy Contract No.
N68305-77-C-0019

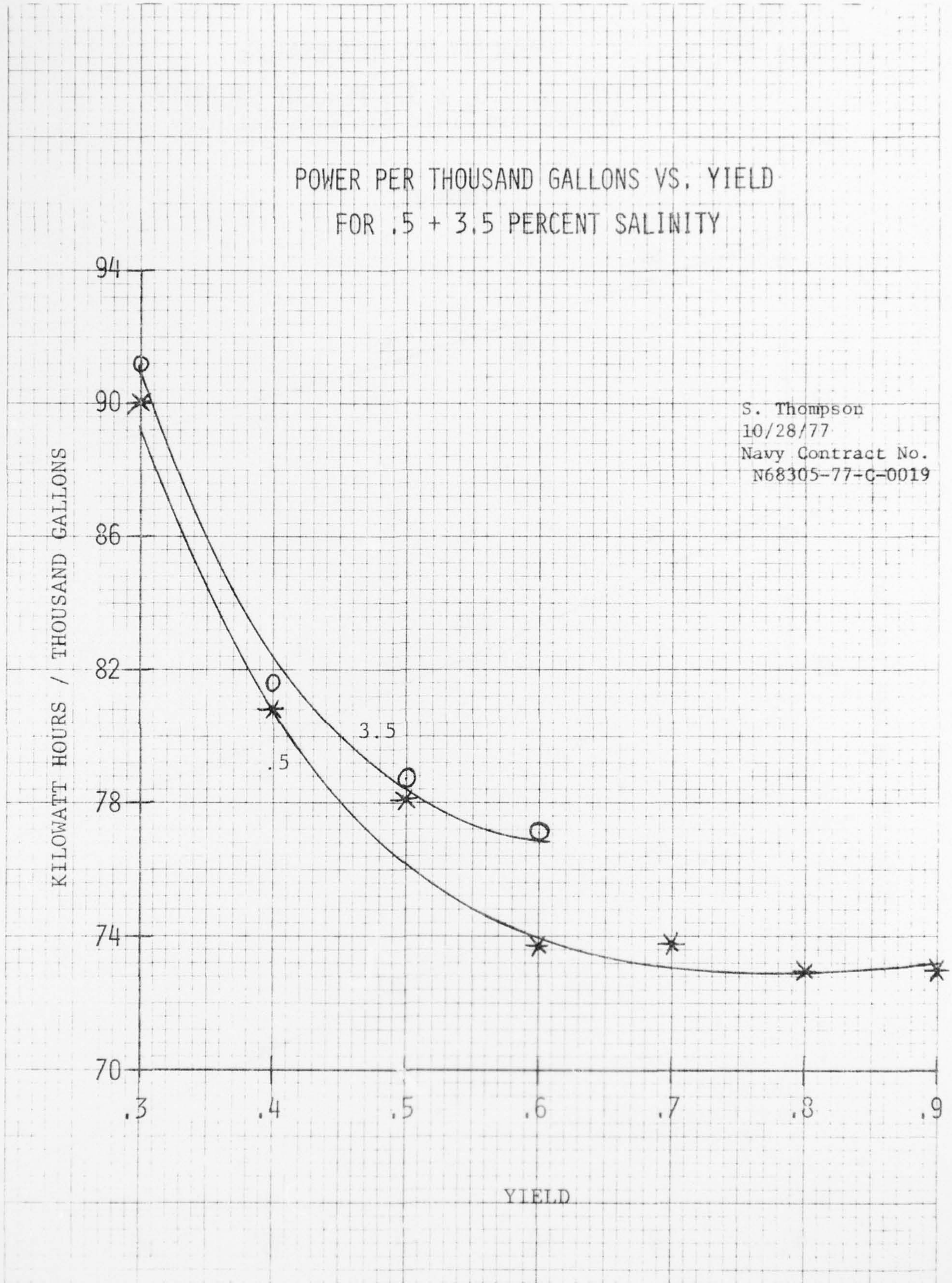
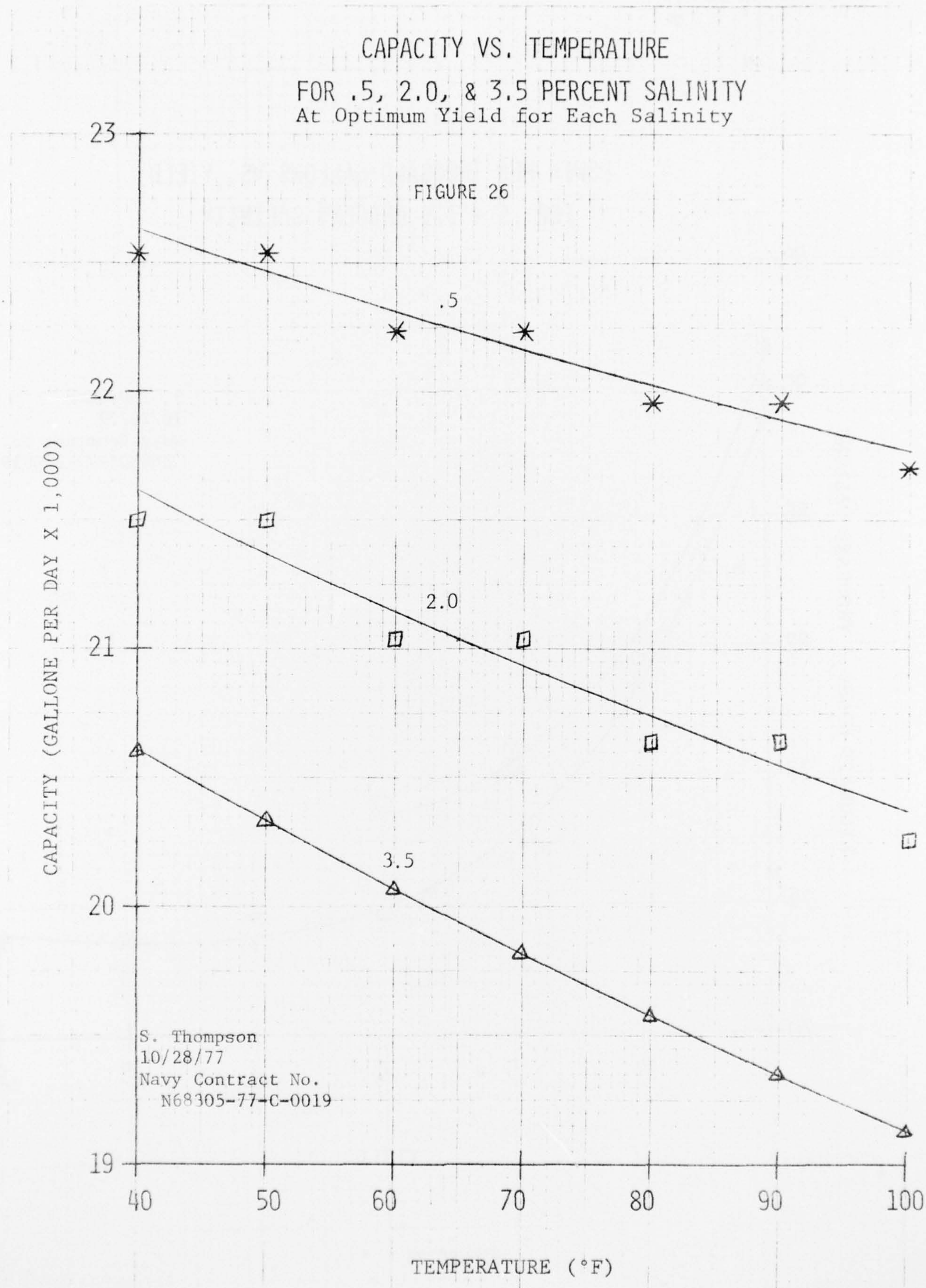

POWER PER THOUSAND GALLONS VS. YIELD
FOR .5 + 3.5 PERCENT SALINITY
94
90
86
82
78
74
70
KILOWATT HOURS / THOUSAND GALLONS
3.5
.5
S. Thompson
10/28/77
Navy Contract No.
N68305-77-C-0019
.3
.4
.5
.6
.7
.8
.9
YIELD

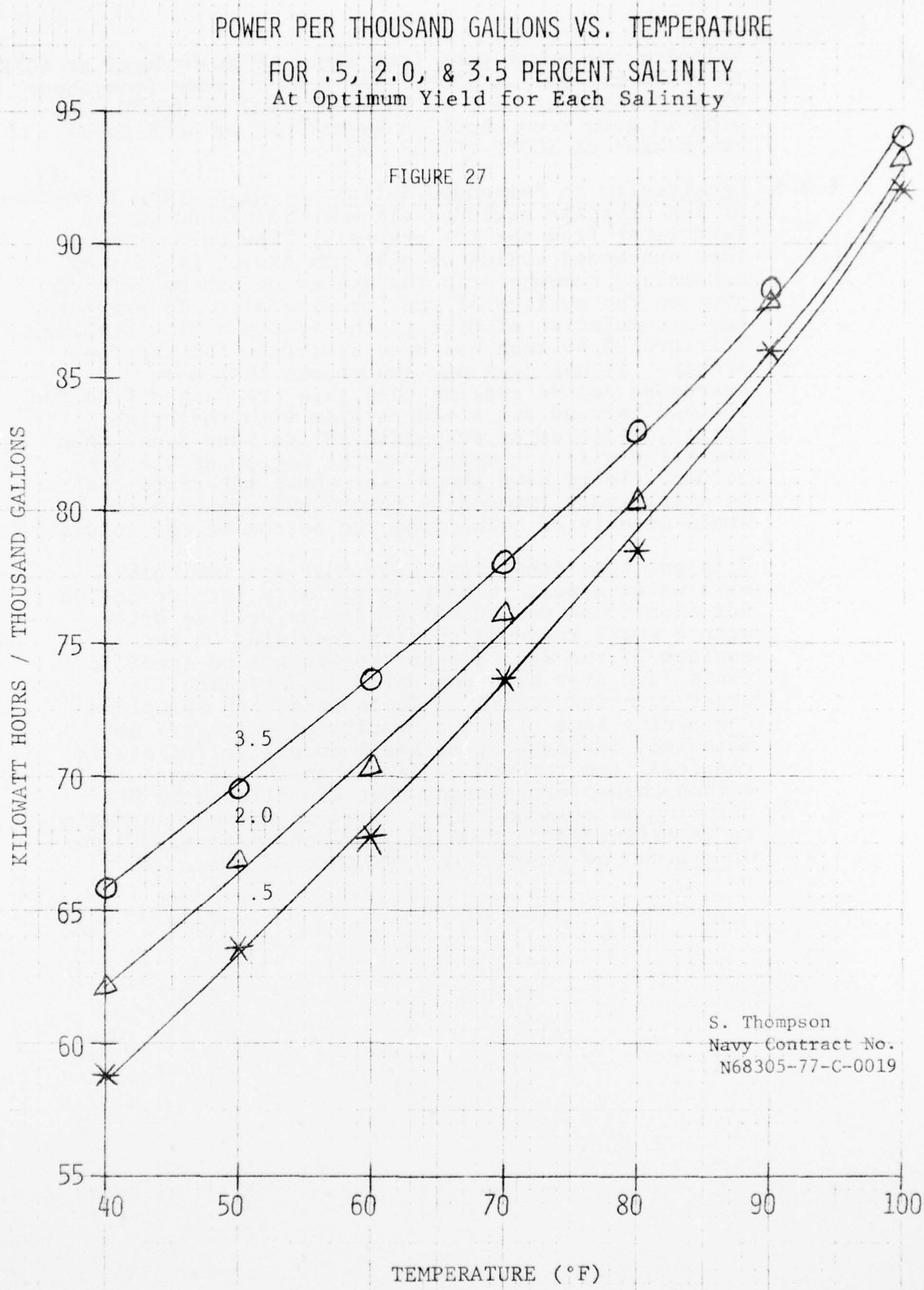
FIGURE 25


CAPACITY VS. TEMPERATURE
FOR .5, 2.0, & 3.5 PERCENT SALINITY
At Optimum Yield for Each Salinity
FIGURE 26
23
22
21
20
19
.5
2.0
3.5
CAPACITY (GALLONE PER DAY X 1,000)
S. Thompson
10/28/77
Navy Contract No.
N68305-77-C-0019
40
50
60
70
80
90
100
TEMPERATURE (°F)

salinity the power goes from about 59 kw-hr/kgal at 40°F to 92 kw-hr/kgal at 100°F. At 2.0% it goes from about 62 kw-hr/kgal at 40°F to 93 kw-hr/kgal at 100°F. At 3.5% it goes from about 66 kw-hr/kgal at 40°F to 94 kw-hr/kgal at 100°F (Figure 27).

6.3.5 Sensitivity to Suspended Solids and Oils. The formation of ice crystals excludes all material found in the feed water from the ice crystals. The only problem that suspended solids or oils can cause, is if they accumulate somewhere in the system or become carried over on the surface of the ice crystals. To prevent any accumulation of dirt in the system a fine strainer (filter) of 40 mesh has been specified for the feed system. It has been our experience that most suspended solids smaller than this are washed from the ice and carried out of the system with the brine. Large quantities of suspended solids more dense than the brine will accumulate in the bottom of the wash column. Tests have shown that these materials can be effectively removed by continuously purging a small quanity of brine from the bottom of the column.

Oils and some other materials that are immiscible with water appear to have an affinity for ice and do not seem to be washed off of ice as well as brine, with a small residual quanity remaining on the surface of the ice. To our knowledge, no specific tests have ever been conducted to determine the exact quantity removed. Tests made with solutions containing some oil, the quanity of which was not measured, indicated that small quanities (of oil in the feed) are removed to less than the .7 mg/l of carbon-chloroform extract that are allowed by EPA drinking water standards. Based on these observations, it is estimated that up to 5 ppm and probably 10 ppm of oil can be removed to less than .7 ppm.



POWER PER THOUSAND GALLONS VS. TEMPERATURE
FOR .5, 2.0, & 3.5 PERCENT SALINITY
At Optimum Yield for Each Salinity
FIGURE 27
KILOWATT HOURS / THOUSAND GALLONS
95
90
85
80
75
70
65
60
55
3.5
2.0
.5
40
50
60
70
80
90
100
TEMPERATURE (°F)
S. Thompson
Navy Contract No.
N68305-77-C-0019

FORCED CONVECTION HEAT TRANSFER (WITHOUT BOILING)

DITTUS - BOELTER EQUATION

$$Nu_d = .023\, Re_d^{\,.8}\, Pr^{\,n}$$

Evaluate Properties at Fluid Bulk Temperature

n = .4 for Heating
.3 for Cooling

$$Nu_d = \frac{h d_o}{k}$$

$$Re = \frac{d_o \rho V}{\mu}$$

$$Pr = \frac{C_p \mu}{k}$$

h = Heat Transfer Coefficient

d_o = Hydraulic Diameter

k = Thermal Conductivity

ρ = Density

V = Velocity

μ = Viscosity

C_p = Specific Heat

APPENDIX A

NUCLEATE POOL BOILING

ROHSENOW GIVES:

$$\frac{C_l \, \Delta T_x}{h_{fg} \, Pr_l^{1.7}} = C_{sf}\left[\frac{q/A}{\mu \, h_{fg}}\sqrt{\frac{g_c \, \sigma}{g\,(\rho_l - \rho_v)}}\right]^{.33}$$

$$Nu_d = \frac{hd}{k} \qquad Re_d = \frac{d\rho V}{\mu} \qquad Pr = \frac{C_p \mu}{k}$$

ROHSENOW'S EQUATION CAN BE SOLVED TO GIVE:

$$h = \mu \, h_{fg}\sqrt{\frac{g\,(\rho_l - \rho_v)}{g_c \, \sigma}}\left[\frac{C_p}{h_{fg} \, Pr_l^{1.7} \, C_{sf}}\right]^{\frac{1}{.33}} \Delta T_x^{\frac{.67}{.33}}$$

C_1 = Specific Heat of Saturated Liquid

DT_x = Temperature Excess = Tw - Tsat

h_{fg} = Enthalpy of Vaporization

Pr_1 = Prandtl No.of Sat. Liquid

q/A = Heat Flux/Unit Area

μ = Liquid Viscosity

σ = Surface Tension

g = Gravitational Acceleration

ρ_l = Density of Saturated Liquid

ρ_v = Density of Saturated Vapor

C_{sf} = Const. Det. from Exp. Data

g_c = 32.174 ft-lb_m/lb - Sec^2

APPENDIX A

BO PIERRE'S EQUATION FOR INCOMPLETE EVAPORATION

$N_u = A\left[R_e^{\,2}\,K_f\right]^n$ for $10 < R_e^{\,2}k_l < 0.7\times10^{12}$

$X_e < .9$

$N_u = \dfrac{hD}{k_1}$

A = Constant = 0.0009 for Std. Pierre Relationship

$R_e = \dfrac{GD}{\mu}$

$K_f = \dfrac{J\Delta x\,\lambda}{L} = \dfrac{778.16\,\Delta x\,h_{fg}}{L}$

The work of Altman, Norris, and Staub led to the conclusion that Bo Pierre's relationship in its original form should be used between

$$2 * 10^9 < R_e^{\,2}K_f < 1.5259 * 10^{11}$$

and a and n should be changed for the range

$$1.5259 * 10^{11} < R_e^{\,2}\,K_f < 1.5 * 10^{12}$$

the final relationships are as follows:

$$N_u = A\left[R_e^{\,2}\,K_f\right]^n$$

for $2 * 10^9 < R_e^{\,2}\,K < 1.5259 * 10^{11}$

$A = .0009$
$n = \frac{1}{2}$

for $1.5259 < R_e^{\,2}\,K < 1.5 * 10^{12}$

$A = .0225$
$n = 3/8$

for complete range:

$G < 2 * 10^5$
$.15 < X \quad , X_{ex} < .80$

APPENDIX A

BO PIERRE'S EQUATION FOR INCOMPLETE EVAPORATION
(Continued)

n	=	Exponent = 0.5 for Std. Pierre Relationship
h	=	Film Coefficient
D	=	Diameter
k_l	=	Thermal Conductivity of Liquid
G	=	Mass Velocity =
μ	=	Viscosity of Liquid
Δx	=	Charge in Vapor Quality
h_{fg}	=	Latent Heat of Vaporization
L	=	Tube Length

APPENDIX A