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CONVERSION

DESIGN STUDY OF A TWO-PHASE TURBINE BOTTOMING CYCLE

Final Report

By
Walter R. Studhalter

MASTER

June 15, 1979

Work Performed Under Contract No. EY-76-C-03-1207

Biphase Energy Systems
Research-Cottrell
Santa Monica, California

U. S. DEPARTMENT OF ENERGY

Division of Industrial Energy Conservation

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Biphase Energy Systems

Research-Cottrell

FINAL REPORT, CONTRACT EY-76-C-03-1207

FOR

DESIGN STUDY OF A TWO-PHASE TURBINE
BOTTOMING CYCLE

Prepared for the
UNITED STATES DEPARTMENT OF ENERGY

June 15, 1979

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Frontispiece. Biphasic Nozzle, Separator and Turbine

1. FOREWORD

This final report describes the effort under the Department of Energy Contract EY-76-C-03-1207, "Design Study of a Two-Phase Turbine Bottoming Cycle". Effort on this subject was initiated by the U.S. Energy Research and Development Administration (Contract E(04-3)-1207) in 1976, and a final report on the ERDA-sponsored work was published on February 15, 1977, "A Biphasic Turbine Bottoming Cycle for a Diesel Engine". Under Department of Energy sponsorship, the effort has been augmented by six modifications which have added to the scope of the work. Specific application of the reported results will continue under the new D.O.E. Contract DE-AC-03-79ET-15350. The D.O.E. effort has been directed by Mr. John Neal, Mr. Marvin Gunn, and Mr. Michael Perlsweig.

2. ABSTRACT

The use of a Biphase turbine system to recover waste heat from diesel engines was examined and found to have many favorable attributes. Among these were low rpm, high torque, low heat exchanger cost and simplicity. Several candidate working fluid combinations were tested at temperatures of interest. The contact heat exchanger concept was substantiated by large scale experiment. The program includes subscale tests of key hardware components of a Biphase turbine bottoming system. These are the two-phase nozzle, two-phase turbine and direct contact heat exchanger. A comprehensive cost analysis was completed. A three-year program leading to a full-size system field demonstration has been planned. The first year of this program is complete and reported herein, and effort has started on the second year program.

SECTION 3

INTRODUCTION

The initial ERDA-sponsored studies of a Biphase Turbine Bottoming Cycle for a Diesel Engine (Contract E(04-3)-1207) defined the potential and advantages of the innovative Biphase turbine system. This effort was summarized in Reference 1*, which enumerated advantages over other methods of energy recovery in terms of cycle efficiency, cost and reliability. As a result of this favorable prognosis, a three year system demonstration program plan was developed and published (Reference 1). This stepwise plan begins with sub-scale component development and demonstration, and ends with a full scale field demonstration of the system used with a stationary diesel engine, gas turbine engine, or industrial furnace.

The three year hardware demonstration plan is summarized in Table 3-1. Hardware tasks in the first year included the Biphase nozzles, the separator/turbine, and the direct contact heat exchanger. This effort is complete and is reported herein. The favorable conclusion of the Year 1 effort led to a current proposal for the Year 2 continuation.

The second year tasks include design of full sized components for the defined reference system. The Biphase turbine delivers 276 HP from the exhaust of an 1800 HP diesel engine. The reference system heats a heat-transfer liquid (initially Therminol 66) in the diesel exhaust, then mixes the hot liquid with water at the entrance to the Biphase nozzle. Expansion of the two-phase fluid in the nozzle imparts kinetic energy to the liquid which operates a simple liquid turbine to produce 156 HP. Remaining kinetic energy in the steam is utilized in a radial inflow turbine to increase the output to 276 HP. The water and Therminol 66 are separated and recycled.

*References are found in Section 11.

Table 3-1. Biphase Diesel Bottoming - 3-Year Program Plan Summary

YEAR 1	YEAR 2	YEAR 3
<p>REFERENCE SYSTEM DESIGN - 1800 HP DIESEL ENGINE BOTTOMING</p> <p>COST ANALYSES AND COMPARISONS</p> <p>HEAT EXCHANGER FLUID COMPATIBILITY STUDY & LAB TEST</p> <p>KEY COMPONENT DESIGN, FABRICATION AND TEST:</p> <p>A. CONTACT HEAT EXCHANGER</p> <p>B. NOZZLES (3)</p> <p>C. SEPARATOR/TURBINE</p>	<p>DESIGN OF FULL SIZE COMPONENTS, 1800 HP DIESEL REFERENCE SYSTEM</p> <p>FABRICATE AND TEST IN A RECIRCULATING SYSTEM, ALL FULL SIZED COMPONENTS EXCEPT HEAT EXCHANGER</p> <p>DESIGN ANALYSIS OF EXTENDED SURFACE EXCHANGER</p> <p>CONTINUATION OF TESTS WITH EXISTING CONTACT EXCHANGER (FULL HEIGHT, 1/5 DIAMETER)</p> <p>A. VERIFY DESIGN CORRELATIONS FOR VAPORIZING/CONDENSING FLUID</p> <p>B. HEAT TRANSFER FLUID COM- PATIBILITY & ENDURANCE</p> <p>INTEGRATE RESULTS FROM COMPANY- SPONSORED RESEARCH ON MIXERS AND NOZZLES.</p>	<p>ENDURANCE TESTS, HARDWARE AND FLUID, FULL SIZE LOOP FROM YEAR 2</p> <p>FABRICATE AND TEST GAS BLADING FOR BIPHASE TURBINE</p> <p>DETERMINE TYPE OF HEAT EXCHANGER, DETAIL DESIGN AND FABRICATE</p> <p>DESIGN FIELD SYSTEM BASED ON DEVELOPED COMPONENTS</p> <p>- DESIGN REVIEW</p> <p>- SECURE AGREEMENT WITH OPER- ATOR OF 1800 HP DIESEL ENGINE</p> <p>FABRICATE, INSTALL AND TEST THE COMPLETE BIPHASE BOTTOMING SYSTEM ON A STATIONARY DIESEL IN THE FIELD</p> <p>- ANALYZE PERFORMANCE DATA</p>

The third year summarized in Table 3-1 proceeds with endurance and reliability tests using the Year 2 test hardware and the fabrication of the heat exchanger. This equipment is taken to the field and installed on a 1800 HP diesel engine for a completely applicable demonstration.

Although the above program is based on a specific reference system, it should be emphasized that options remain in the cycle upper liquid temperature (may be limited by fluid properties), selection of heat transfer fluid, placement of the pump on the hot or cold side of the heat exchanger, and condensing temperature level. The valid system studies also include application to gas turbine engines, and to industrial furnaces and kilns. During the Year 2 and 3 programs, favorable results are expected to suggest extensions to these other sources of waste heat.

This final report summarizes the contracted effort since the publication of Reference 1, with principal emphasis on the test programs for the contact heat exchanger and the Biphase nozzles and separator/turbine.

SECTION 4

BASELINE SYSTEM REVIEW

The reference bottoming system developed in Reference 1 was based upon an 1800 HP Colt-Pielstick PC-2 diesel engine. This engine is representative of many of interest for waste heat recovery. The Biphase cycle is two component, water and the heat transfer fluid Dowtherm A. Dowtherm A was selected on the basis of familiarity and lack-of-toxicity, although its specific heat (approx. 0.51 BTU/lb-F) is higher than desirable. This value of specific heat, plus the desire to limit turbine speed, resulted in the decision to keep the nozzle back pressure at 14.7 PSIA. The steam is further expanded to 1.2 PSIA after separation from the Dowtherm A.

This baseline system is illustrated in Figure 4-1, which also includes state point conditions. Diesel exhaust gas corresponding to the 1800 HP engine gives up its heat (cools from 719F to 242F) to the Biphase working fluid in a tubular heat exchanger. The heated fluid (660F) goes to the Biphase nozzle. The fluid in the heater is at a pressure of 2365 PSIA; it is 2.31 lb/sec Dowtherm A and 0.66 lb/sec water. In the expanding section of the nozzle, the water flashes to steam. On the separator wheel of the turbine, the Dowtherm A liquid is separated from the steam, and then used to operate the liquid turbine. This Biphase turbine includes radial inflow steam blades to extract power from expansion to 1.2 PSIA. The steam is then condensed. Condensed steam (water) and Dowtherm A are mixed together, pumped to the high working pressure, and returned to the heater. In this arrangement, the pump is operating on the cold side of the heater (190F), which is the

Figure 4-1. Reference Bottoming System, Tubular Exchanger
(Reference 1), State Point Conditions

least expensive pump location. Mixture of the Dowtherm A and water takes place in the pump suction, so the fluid mixture flows through the pump and the heat exchanger to the nozzle. Whether a good mixture can be maintained under these conditions has not been demonstrated yet; the alternate arrangement would place the fluid mixer just before the nozzle. Reference 1 includes a schematic which enlarges Figure 4-1 to include a start tank in which the fluid inventory is preheated by exhaust gas in the tank jacket.

During the course of the 1978-79 effort, the direct contact heat exchanger test program yielded encouraging results, as reported in Reference 5. Accordingly, a revised "baseline" system employing the direct contact exchanger was defined for use in the economic analyses (Reference 2). This system is illustrated in Figure 4-2. The heat transfer fluid Therminol 66 (Monsanto Company) was also substituted for Dowtherm A. Since the direct contact heat exchanger must operate at essentially atmospheric pressure, the main pump must operate on the hot side (Therminol at 660F). The consequences of this on equipment cost are detailed in Reference 2. Examination of Figure 4-2 also reveals two additional pumps: a low-head recirculating pump for the fluid, and a high-head water pump. The water circuit is separate from the heat exchanger, and mixing takes place just before the nozzle.

The system in Figure 4-2 is illustrated at steady state operation. Addition of the start equipment is shown in Figure 4-3. The fluid inventory is kept in the "start" tank, which is so located that it can be preheated by the diesel exhaust. The start tank inventory of preheated fluid is sufficient to bring the turbine and pumps up to operating speed and start operation of the contact exchanger. Figure 4-3 shows the start tank heated by diesel exhaust ducted to a jacket around the tank. This exhaust duct could also be used as a bypass when the Biphasic system is not in operation.

Figure 4-2. Reference Bottoming System With Direct Contact Heat Exchanger (Reference 2), State Point Conditions, Therminol 66

Figure 4-3. Reference Bottoming System, Direct Contact Heat Exchanger

In making the proposal for continuation of the Biphasic development work (Reference 6), flexibility was maintained by proposing a system similar to Figure 4-2, but with the feature of alternate heat exchangers, tubular or direct contact. When used with a tubular heat exchanger, this system can operate with the fluid pump on the cold side. It is also assumed that the heat transfer fluid may be Caloria HT43 (Exxon Co.), which could limit the upper temperature to a value below 660F. In this follow-on work, the details of the system selected for commercialization will be determined. Some consideration may also be given to a single-component system, which operates on steam and water.

SECTION 5

ECONOMICS OF BIPHASE TURBINES FOR WASTE HEAT RECOVERY

The Topical Report (Reference 2) was published on October 13, 1978, containing complete economic information. Following is a summary of the report.

The concept of a Biphase turbine, in which most energy is converted via a liquid turbine, results in significant cost advantages as compared to a gas turbine. In addition, the Biphase system has important potential advantages which are unique to waste heat recovery:

1. The available energy in the waste heat stream can be maximized.
2. An inexpensive primary heat exchanger can be used.

Other advantages of cost, reliability, and efficiency are common to the Biphase turbine application to waste heat and to other projects. Programs to demonstrate the advantages are currently undertaken in waste heat conservation, geothermal power, solar power, solar cooling, marine propulsion, and petroleum production. The technical principles and engineering definition of component parts of the system are sufficient to design and price specific complete systems, and to parameterize the system design to yield cost information on a wide range of system sizes and characteristics. This economic analysis is detailed in Reference 2.

The point of departure for the economic analyses is the reference diesel bottoming system under the subject contract. The reference diesel engine is an 1800 HP stationary engine which produces exhaust gas at 717F. The bottoming system utilizes this gas, without otherwise disturbing operation of the engine, to produce an additional 276 HP (206KW) of shaft power. Installed component costs were estimated using standard methods. The Biphas turbine cost is illustrated as an example in Figure 5-1. The installed cost of the bottoming system is estimated at \$72,526, or \$352 per KW. This baseline point is indicated on Figure 5-2, together with the indication of cost trends based on system size and on the temperature of the exhaust gas. These trends are applicable to bottoming systems applied to gas turbine engines and to industrial furnaces and kilns, as well as the baseline diesel engine. The cost of the baseline system including an electrical generator to deliver electrical power, is shown in Figure 5-3.

These data are enlarged to include systems using two different kinds of primary heat exchangers in Reference 2. The component cost breakdown for one system only, the reference system with a contact heat exchanger, delivering shaft power, is shown on Table 5-1.

PAYBACK PERIOD

A payback period analysis was defined for the waste heat conservation systems based on annual cost. The approach considers interest, operating costs, taxes and insurance, the operating life, and the value of the power produced. An example of this payback analysis, which relates the required first cost of the bottoming system to the payback period, is shown in Figure 5-4. For the example discussed above, first cost = \$352 per KW, the payback period is between one and two years. Other examples of the use of this method are included in Reference 2.

Figure 5-1. Biphase Turbine, Installed Cost

Figure 5-2. Installed Waste Heat Biphase System Cost Based on Shaft Power Delivered

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Figure 5-3. Installed Waste Heat Biphase System Cost,
Based on Delivery of Electricity

Table 5-1. Diesel Bottoming Reference System
Contact Heat Exchanger

Based on 206KW Delivered at Turbine Shaft

Components, Major

Biphase Turbine and Nozzle	\$23,500	
Condenser	1,632	
Contact Heat Exchanger (Packed Tower)	10,712	
Therminol Pump	8,360	
Water Pump	2,172	
Vacuum Pump	450	
	<hr/>	
Subtotal		\$46,826

Minor Components and Shop Assembly

Control Console	\$ 2,000	
Control Valve	1,200	
Valves, Manual and Vacuum (9)	1,200	
Therminol Storage Vessel	500	
Water Storage Vessel	250	
Therminol Inventory (30 gal.)	300	
Piping	2,000	
Electrical	1,000	
Heat Insulation	800	
Structural - Skid, Equipment Mounting	7,000	
Shipping, Shop to Field Site	800	
	<hr/>	
Subtotal		\$17,050

Field Connection - Start-Up

Field Supervision	\$ 2,500	
Concrete Pad for Skid-mounted Equipment	650	
Engine Exhaust Flexible Duct	400	
Field Electrical	500	
Licensing	600	
Start-up (Routine, not R&D)	4,000	
	<hr/>	
Subtotal		\$ 8,650
		<hr/>
GRAND TOTAL		\$72,526

206KW delivered ----- (\$352 per delivered KW)

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Figure 5-4. Required First Cost, Payback Analysis
Based on Annual Cost

MARKET PENETRATION

Some early cost estimates made by Biphas were used by the Electric Power Research Institute (EPRI) for a study of the role of advanced technologies in small utilities. The EPRI study, Reference 3, utilized Biphas system costs somewhat higher than those reported herein; nevertheless, the Biphas system is shown to have the potential for a substantial penetration of the small utilities market. Table 5-2, from the EPRI report, gives estimated market penetration numbers (in percent) for the period 1985 - 2000.

Table 5-2. Intermediate-Peaking Capacity Penetrations
for Optimum Intermediate-Peaking (Oil Fired)
Advanced Technology Expansion Plans

Reference System	Type of Advanced Technology Intermediate- Peaking Capacity	Advanced Technology Capacity Penetration By Year 2000 (% of Total Capacity)	
		Base Load Participation Scenario	Purchased Power Scenario
35-MW NC Municipal	—	—	—
	Organic Rankine Cycle	9	19
	Two-phase Engine	12	22
	Oil-fired Stirling	18	28
35-MW NE Municipal	—	—	—
	Organic Rankine Cycle	0	19
	Two-phase Engine	0	22
	Oil-fired Stirling	6	26
10-MW Municipal with generation	—	—	—
	Organic Rankine Cycle	0	19
	Two-phase Engine	0	21
	Oil-fired Stirling	18	28
10-MW Municipal without generation	—	—	—
	Organic Rankine Cycle	9	19
	Two-phase Engine	11	21
	Oil-fired Stirling	18	23
35-MW Distribution Cooperative	—	—	—
	Organic Rankine Cycle	0	10
	Two-phase Engine	4	14
	Oil-fired Stirling	10	20

SECTION 6

HEAT TRANSFER FLUIDS

The studies of two-component Biphasic systems reported in Reference 1 envisioned a high vapor pressure fluid, which would flash to a vapor when pressure is reduced in the expansion section of the nozzle, and a low vapor pressure fluid which would remain liquid throughout the cycle. In some of the examples studied, water was chosen as the low vapor fluid, and Dowtherm A as the high vapor pressure fluid. It was also apparent from the analyses presented that fluid physical properties had a significant effect on system operating limits and efficiencies; for example, a low sensible heat of the nonvaporizing fluid enables operation at favorably low exhaust pressure with high mass ratio. Other fluid properties are related to safety (i.e., flammability), economy of operation (i.e., stability, cost) and range of operation (i.e., freezing point). During the reporting period, the properties considered desirable for the heat transfer fluids were tabulated (Table 6-1) to assist in future planning. This information was also circulated to companies in the Chemical Process Industries to encourage closer approach to the ideal fluid. In the waste-heat portion of Table 6-1, water was assumed to be the high vapor pressure fluid; no other fluid appeared to offer reasonable benefit. The low vapor pressure fluid may be different in a closed cycle bottoming cycle employing a tubular heat exchanger in the gaseous heat source (diesel exhaust), than in the cycle using a direct contact heat exchanger. In the direct contact exchanger, any fluid evaporated at the top of the exchanger is lost to the environment, hence the importance

Table 6-1. Desirable Properties, Heat Transfer Fluids

Property \ Program	Waste Heat -- Bottoming		Solar Cooling	
	Direct Contact Heat Exchanger	Tubular Heat Exchanger	Low V.P.	High V.P.
Vapor Pressure	Low <0.5 psia @ 600F	Low <1.0 psia @ 600F	Low <10 mmHg @ 300F	High <300 psia @300F
Specific Gravity	High	High	High	High
Specific Heat	Low <0.4 BTU/lb-F	Low <0.4 BTU/lb-F	High	-
Flammable	Non-flammable or AIT >1200F	Non-flammable AIT >900F	Non-flammable High AIT	Non-flammable High AIT
Toxicity	Non-toxic Non-objectionable	Non-toxic Non-objectionable	Non-toxic -	Non-toxic -
Stability-Temperature In Presence of	To 700F Metals	To 700F Metals	To 300F Steel, Copper, Aluminum, H ₂	To 300F Metals
Oxidation	Very High Resistance	High Resistance	High Resistance	-
Sealing Materials	Conventional	Conventional	Conventional	Conventional
Thermal Conductivity	High*	High*	High	High
Molecular Weight	-	-	Low	Low
Surface Tension	Low	Low	Low	-
Freezing Point	Low*	Low*	-	Low (Sub-zero)
Pour Point	Low	Low	Low	-
Viscosity	Low*	Low*	Low	Low
Water Solubility	Low or Zero	Low	Low or Zero	-
Hydrolysis Rate	Low	Low	Low	-
Gamma	-	-	-	Low
Critical Temperature	-	-	-	High >>300F
Latent Heat	-	-	-	High
Cost	Low	Low	Low	Low
Availability	Readily Available	Readily Available	Readily available in USA	Readily Available
*Secondary importance				

of very low vapor pressure and non-toxicity. Also, in the direct contact exchanger the fluid is in contact with air, and fire is a possibility.

During the course of the contract, the heat transfer fluids were studied by:

1. Review of available materials, and contact with principal manufacturers.
2. Subscale testing of individual fluids and pairs of fluids.
3. Full scale heat exchanger, nozzle and turbine testing operations with selected fluids.

AVAILABLE FLUIDS

The extensive review of the industry conducted during the previous program phase resulted in selection of the following fluids for consideration. Vapor pressure, specific heat, density, viscosity and thermal conductivity are tabulated in Reference 1.

- (1) Therminol group of heat transfer liquids:
Therminol 88, Therminol 66, Therminol 60, Therminol 55,
Therminol 44
- (2) Dowtherm A and Dowtherm G
- (3) Mobiltherm 600, later called Caloria HT43
- (4) Chevron HT-1
- (5) Arco Rubilene
- (6) Dow Corning group of Silicone and FluoroSilicone
fluids: Dow Corning 550 and 710
- (7) Calgon Refrigeration Oil
- (8) Dupont's Freon group
- (9) Fish oil (animal oil) (Thermofluid)

Subsequently, the following were also considered:

- (10) Tin metal
- (11) Krytox 143AC (Dupont Company)
- (12) Polyphenyl ether (05-124) (Monsanto Company)
- (13) Brayco 880 (Bray Oil Company)

Initial screening based on published properties resulted in selection of Therminol 66, Dowtherm A and fish oil for initial analysis. Although fish oil appeared to meet the requirements, it was omitted because of supply and environmental problems. Therminol 66 was selected for initial contact heat exchanger tests; later in the program Caloria HT43 became a serious contender.

SUBSCALE TESTING

Initial capsule tests, for screening possible working fluids, were conducted in the latter part of 1977. In these tests, a stainless steel capsule, volume approximately 50 ml., was filled with the test fluid, or with both low and high vapor pressure fluids, then sealed and placed in a furnace at high temperature. Pressure was monitored by means of a strain gage type transducer connected to the capsule with capillary tubing. After exposure, the samples were examined.

In the first series, compatibility tests of 19 samples were completed with approximately 1000 hours elapsed time at temperatures of 500° and 600°F in stainless steel capsules. The sample fluids were then transferred to plastic bottles for storage and visual examination. Table 6-2 lists the fluids used, the capsule designation letters and the visual appearance of the tested samples. The test procedure consisted of cleaning the stainless steel capsules, filling with fluids A and B (fluid A being the thermodynamic or high vapor pressure working fluid, fluid B the liquid) and preheating with slightly open capsule until vapor of

Table 6-2. Fluid Compatibility Test Results, First Series

FLUID B (Low Vapor Pressure)	FLUID A (High Vapor Pressure)	Capsule Letter & Batch No.	Result	Capsule Letter & Batch No.	Result
DOWTHERM A (Dow Chemical) (26.5% Diphenyl $C_6H_5C_6H_5$ 73.5% Diphenyl Oxide $(C_6H_5)_2O$)	H ₂ O	R ₁	IV	G ₁	IV
DOW CORNING 510 FLUID (Dow Corning) (Phenylmethyl Polysiloxane)	DOWTHERM A	-	-	B ₁	I-II
KRYTOX 143 AC (DuPont) (Perfluoroalkyl Polyether)	H ₂ O	D ₂	I-IV	J ₂	I-IV
	DOWTHERM A	-	-	W	I
	DOWTHERM J (Alkylated Aromatic Hydrocarbon)	AB ₃	I	-	-
TEERMINOL 66 (Monsanto)	H ₂ O	M ₁	IV _(III)	Q ₁	III
	DOWTHERM A	-	-	N	III
	DOWTHERM J	Z ₃	I	-	-
ARCO RUBILENE S-215 (Arco)	H ₂ O	K ₁	I	H ₁	IV
	DOWTHERM A	-	-	A	III
	DOWTHERM J	X ₃	I	-	-
MOBILTHERM 603 (Mobil)	H ₂ O	P ₁	I-II	S	III-IV
CHEVRON HEAT TRANSFER OIL 1 (Chevron)	H ₂ O	L ₁	II	Y ₁	II-IV
Legend: I - Original Condition Test Periods: Batch 1 - 1004 Hours II - Discolored Slightly Batch 2 - 977 Hours III - Heavily Discolored Batch 3 - 833 Hours IV - Solid Residue					

the medium A became visible, at which time the capsule was sealed and placed in the test furnace. The purpose of the pre-heating was to remove the air (in particular oxygen) from the capsule. The test pressure in the capsule is primarily given by the vapor pressure of the medium A at the test temperature. Table 6-2 shows that while the thermodynamic medium A was mostly water, tests were included with Dowtherm A, which is an eutectic mixture of diphenyl and diphenyl oxide, and with Dowtherm J, an alkylated aromatic hydrocarbon of higher vapor pressure (174.5 psi at 600°F and 74.33 psi at 500°F) than Dowtherm A. The latter has a vapor pressure of 45.34 psi at 600°F and of 15.65 psi at 500°F. The boiling point for Dowtherm A is at 494.8°F = 257.1°C; for Dowtherm J it is at 358.4°F = 181.3°C. The liquid components B listed include the synthetics (Silicone, Krytox and Therminol) and the three petroleum base oils (Rubilene, Mobiltherm and Chevron HT-1). A preliminary indication of the tests is that mixtures of synthetic liquids with the Dowtherms show better compatibility than together with water. The petroleum based oils seem to be compatible with water as long as the temperature does not rise as high as 625°F. It should be noted that furnace temperature readings were taken every two hours starting at 8 a.m. and ending at 4 p.m. each day. In the 500°F furnace the temperature varied between 500° and 525°F, with the temperature rising during any one day. In the other furnace, the temperature held steady at 625°F all the time, except for the last day when it climbed to 650°F. The furnaces were factory calibrated before the test.

The data from the first test series, and continued liaison with manufacturers, resulted in selection of ten low vapor fluids for test in a second series. The ten fluids are listed in Table 6-3. At this point, the high vapor pressure fluid was conceded to be water, and water was used as the second fluid in all further tests.

Table 6-3. Heat Transfer Fluid, Second Series Capsule Tests*

	Trade Name	Manufacturer	Composition	Auto Ignition Temp. F	Specific Heat @ 100C	Specific Gravity @ 100C	Vapor Pressure @ 500F psia	Price \$/lb
1	Therminol 66	Monsanto	Terphenyls	705	0.51	0.88	1.93	1.21
2	Dowtherm G	Dow Chemical	Di- and tri-aryl ethers	1030	0.40	1.10	5.7	0.94
3	C510	Dow Corning	Silicone- phenylmethyl polysiloxane		0.40	1.0		11.09
4	Tin Metal	--	--	Non-flammable	0.056	8.39	~0	2.0
5	FS 1265	Dow Corning	Fluorosilicone	720		1.25		52.00
6	Krytox 143AC	DuPont	Fluorinated oil- perfluoroalkyl polyethers	Non-flammable	0.25	1.9	0.003	152.0
7	05-124	Monsanto	Polyphenyl ether	1135	0.37	1.19	0.016	111.0
8	Caloria HT43	Exxon	Petroleum fraction plus inhibitors	759	0.51	0.86	0.30	0.22
9	Heat Transfer Oil No. 1	Chevron	Petroleum fraction plus inhibitors	~780	0.45	0.88	0.23	0.22
10	Dowtherm A	Dow Chemical	Diphenyl/diphenyl oxide	1150	0.43	1.0	15.6	0.89
--	Ideal Fluid	--	--	High or non- flammable	Low	High	Low	Low

*Each sample is the designated fluid plus water.

The second series of compatibility tests was run from September 19, 1977, to November 17, 1977. The second series placed samples of the fluid in contact with water in a sealed capsule, which included a pressure transducer. The purpose of the tests was to determine if the presence of water (which is added to the hot fluid in the two-phase cycle) would affect stability results. The capsules were heated to 400°F in an electric oven, and pressure was monitored for 500 hours.

Tabulated data, and detailed plots, are available in the Biphasic files. A summary of pressure data is presented in Figure 6-1.

In examining Figure 6-1, the four essentially horizontal curves are interpreted as satisfactory performance of the test equipment and also of the fluid in the presence of water. The pressure reached is approximately the vapor pressure of water at the temperature. The lack of pressure change with time is interpreted as a favorable measure of stability. Fortunately, this group included both fluids selected for the experimental program, Therminol 66 and Dowtherm A.

The tin/water sample exhibited a continuous pressure increase, reaching 1200 psi after 500 hours. This is interpreted as a continual hydrolysis (reaction with water to produce hydrogen). It should be noted that this slow reaction would not preclude the use of tin for a short time project or in an open system. The two-phase turbine, direct-contact exchanger system is essentially an open system.

Two other samples, Dowtherm G and Krytox 143AC, started out with satisfactory pressures for 25 and 50 hours respectively, and then dropped suddenly to a low value. This behavior is attributed to a leak in the capsule system, and so no valid data was obtained for these two samples. Also, Caloria HT43 apparently had a leak in the capsule from the beginning, because no pressure rise was registered.

Figure 6-1. Capsule Tests - 500 Hours at 400F,
Heat Transfer Fluids Plus Water

also contain traces of lubricating oil and particulates originating from carbon, fuel additives or mechanical wear. Accordingly, samples of Therminol 66 and Dowtherm A were exposed to water, water acidified with HNO_3 , water acidified with H_2SO_4 , and water plus used motor oil.

The ten sample capsules were connected to pressure transducers by means of a three-way valve (see Figures 6-2 and 6-3), so that the sample would not be continually connected to the transducer and so that the transducer could be recalibrated during the runs. Unfortunately, the three-way valve seals did not withstand the environment, and leaks were detected after 226 hours of heating (48 at 300F, 48 at 400F, and 130 at 500F). However, it is apparent that the reactions that we wished to observe would not result in marked pressure changes; accordingly, the visual observations recorded in Table 6-4 are indicative of the results expected. It is noted that fluid plus pure water (in the presence of a small amount of air) did not produce an observable change. This is consistent with previous capsule tests. Also the addition of 1 gram of used motor oil did not lead to an observable effect. On the other hand, small amounts of the mineral acids, nitric and sulfuric, did result in degradation of the fluid.

The conclusions of the third series of compatibility tests are:

1. Testing in sealed capsules at the 5 ml. level will not be expected to add further information. A program of exposing large samples to actual exhaust gas is indicated and will be recommended.
2. Some components of the diesel exhaust, including acidic components and oxidants, require further study. To protect the heat transfer fluid, consideration should be given to additives and/or purification procedures.

Figure 6-2. Ten Sample Capsules Mounted on Furnace

Figure 6-3. Pressure Measurement from Capsules

Table 6-4. Capsule Samples-Results of Temperature Exposure

Sample	Composition	Result
1	Therminol 66 5 ml.	No change
2	Therminol 66, 2.5 ml. Water 2.5 ml.	No apparent change
3	Therminol 66, 2.5 ml. 1% nitric acid 2.5 ml.	Dark Brown Color in aqueous portion
4	Therminol 66, 2.5 ml. 1% sulfuric acid, 2.5 ml.	Dark color in Therminol portion
5	Therminol 66, 2.5 ml. Water 2.5 ml. Used Motor Oil 1 gm	No apparent change
6	Dowtherm A, 5 ml.	No change
7	Dowtherm A, 2.5 ml. Water 2.5 ml.	Milky cast to aqueous portion
8	Dowtherm A, 2.5 ml. 1% nitric acid, 2.5 ml.	Dark Brown color in aqueous portion
9	Dowtherm A 2.5 ml. 1% sulfuric acid, 2.5 ml.	Dark floc in Dowtherm portion
10	Dowtherm A 2.5 ml. Water 2.5 ml. Used motor oil 1gm.	No apparent change

The above survey and test procedures identified heat transfer fluids suitable for the prototype test programs; however, comparison with desired properties, Table 6-1, indicates that further fluid development is indicated. The Dupont fluid Krytox 143AC most nearly meets the specifications, but was not considered further because of a current prohibitively high cost. Fluid manufacturers in the Chemical Process Industries are invited to help meet the specification requirements, and to call the author's attention to improvements.

TEST OPERATIONS AT FULL SCALE

Operating experience was obtained with Therminol 66 and with Caloria HT43 during the contact heat exchanger, nozzle and turbine test programs reported elsewhere in this report. Initial tests in the contact heat exchanger were conducted with heated Therminol 66 contacting nitrogen gas. Some water was initially present in the apparatus, and contact with air at the open top of the exchanger was possible. The first test program, covering a period of fewer than three hours and liquid temperatures less than 400F, has led to some observable ageing of the fluid. Traces of a black sludge-like material were deposited on the exchanger internals. Plume issuing from the exchanger top into the atmosphere was assumed to result from distillation of light ends of the fluid as part of the initial ageing process. Therminol 66 was used for the remainder of the tests, for a total operating time over 8 hours. The fluid was exposed to air at the top of the column, but only at low temperatures. The Therminol was exposed to water from a companion program which used the gas furnace on several occasions. This small amount of water contamination was tolerated for these tests because Therminol and water were contacted naturally as part of the Biphasic turbine cycle. In these tests, however, it was found that the Therminol and water formed an emulsion type mixture from which the two components were not easily recovered. The resulting

fluid cavitated readily in the pumps and had to be heated to above 300F at atmospheric pressure to separate the components.

It was also found that if the Therminol 66 were heated to 440F or more in the exchanger a fine vapor cloud was emitted from the top of the tower. Chemical analysis (Table 6-5) showed this cloud to be condensed Therminol 66 vapor. The latter observation indicated a future concern with adequacy of the demister section at the top of the tower to prevent loss of the fluid.

Unused, liquid Therminol 66 has a transparent light yellow color. After several heating cycles, however, the fluid became opaque and dark in color, and a small amount of black sludge was deposited in the water-cooled heat exchanger. Samples of the new and used Therminol were returned to the manufacturer for analysis and comment. Table 6-6 includes results of this analysis. The Monsanto Company reported that the used fluid showed "obvious degradation".

The initial portion of the nozzle test program was conducted with Therminol 66 (Monsanto Co.), in accordance with the baseline system definition. The Therminol is mixed with water in the nozzle, then discharged into the receiving tank, where the water should separate from the hot fluid. In practice, it was found that Therminol 66 and water formed some kind of a combination which was difficult to separate at temperatures below approximately 400F. This material caused test difficulties by lowering pump NPSH and by increasing in pressure during the heating cycle in the closed heating system. It was found that an extra blowdown of the fluid only through the nozzle was required to eliminate the water, before each data run could be essayed. In addition, the Therminol/water was apparently a poor lubricant; several instances of piston/liner seizure in the positive-displacement pump were recorded. It should be noted that all the difficulties with Therminol 66 discussed above are related

Table 6-5. Analysis of Vapor Samples from
Top of Contact Heat Exchanger

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Biphase Engines

ARLI Lab Log 68067

The Therminol 66 consists of several components of which the four major and two minor G.C. peaks were selected to represent the compound. Since Therminol is readily soluble in acetone, this solvent was selected as the diluent. Each sample bottle was rinsed with 20 ml acetone. There was sufficient Therminol recovered in each rinse to make further concentration unnecessary. The ratios of the six identifying peaks remained relatively unchanged in samples 2 and 3 when compared to the Therminol standard (basically there was a slight increase in the lower boiling constituents to the detriment of the higher boiling ones). In sample 1, the two minor constituents chosen for identification (one on the lower boiling side and one on the higher boiling side of the group of four major constituents) were both increased 20 fold over the standard concentration (to 20% each vs. 1% in the original) with the ratios of the four "major" constituents to each other remaining relatively constant.

The recovery from each bottle is listed below. For ppm determination, an average molecular weight of 240 was assigned to the Therminol, and for the mg/liter value, sampling volume was assumed to be 1 liter each for samples 1 and 2, and 100 in³ for sample 3.

<u>Sample Number</u>	<u>ppm (v/v)</u>	<u>mg/l</u>
1	180.	1.85
2	256.	2.64
3	175.	1.80

Table 6-6. Therminol 66, Analysis Reported by
the Monsanto Company

Property	Sample Used 8 Hours	Unused Sample
Appearance	Dark Brown	Hazy Light Yellow
Viscosity at 100F, cs	46.48	29.96
ph	4.8	7.5
SAN (Strong Acid Number)	0	0
TAN (Total Acid Number)	3.82	0.01
Water, ppm	208.0	332.0
Acetone Insoluble, mg/100 ml	147.0	9.0
Gas Chromatograph:		
Low Boilers, percent	0.1	0.7
High Boilers, percent	16.3	11.2

to the test equipment (pumps, separating tank) and may not be significant to the baseline Biphase cycle.

Nozzle testing continued with the alternate heat transfer fluid, Caloria HT43 (Exxon Co.). Caloria is petroleum based, and separated easily from water. The oil is also apparently a better lubricant in the pump. With the test equipment in use, Caloria resulted in a higher test frequency. All turbine tests were made with Caloria HT43.

At the present state-of-the-art, the best test results on large equipment have been obtained with Caloria HT43 heat transfer oil. Further testing involving a direct contact heat exchanger will start with Caloria.

SECTION 7

DIRECT CONTACT HEAT EXCHANGER TEST RESULTS-SUMMARY

The Topical Report (Reference 5) contains a complete description of this direct contact heat exchanger program. Following is a summary of the topical report.

INTRODUCTION

The Biphase turbine cycle application to diesel engine exhaust bottoming requires heat exchange between a hot diesel exhaust and a thermal fluid. Systems have been studied which use both tubular and direct contact exchangers, but because the heat exchanger cost has a large effect on system cost the less costly direct contact exchanger has been favorably considered (Reference 2).

Data for the design of such a direct contact heat exchanger are almost totally lacking in engineering literature, especially if evaporation and condensation of the heat transfer fluid is considered. Accordingly, the Biphase diesel bottoming project included an experimental program to obtain data pertinent to the contract baseline system. The scope of this program was expanded by a later add-on to include additional data which might be used to generalize the design approach to include other fluids and other conditions.

EQUIPMENT AND INSTRUMENTATION

The direct contact heat exchanger was sized to permit the experimental verification of the packing depth required to heat the

thermal fluid to the desired temperature (500°F) by direct contact with a diesel exhaust stream. The cross-sectional area of the exchanger was $\sim 1/5$ of that estimated as being necessary for the 1800 HP diesel engine reference system defined in Reference 1.

The test equipment is shown in the schematic, Figure 7-1. The contact heat exchanger consists of a 1-foot diameter packed bed of variable height. It has provisions below for the entry of hot gas and the exit of hot liquid, and provisions above to inject cold liquid and for cool gas exit. The liquid circuit, designed for a heat transfer fluid such as Therminol 66 (Monsanto Co.), includes a distributor tray above the packing, and a section below the packing where hot liquid passes through a water-cooled heat exchanger, and is pumped (via a filter) to the top of the exchanger to complete the circuit.

The packing of the heat exchanger was 1-inch Pall Rings made of carbon steel. The exchanger body was fabricated from a section of 12-inch pipe. Standard flanges were used at the top and bottom and in the center as a support. Commercial parts included a packing support, liquid distribution tray, and demister. View ports were provided just below the bed and above the distribution tray. The design included provision for experimental measurement of temperature, pressure, pressure distribution, and gas composition.

Compressed nitrogen was selected as the test gas. It was supplied from a tube trailer, heated inside tubes in a gas fired furnace, and injected into the exchanger below the packed bed. The cooled gas exited to the atmosphere at the top of the exchanger after passing through a demisting pad. Ancillary equipment included a cooling tower for the heat exchanger water and a steam boiler used for preheating the thermal fluid.

Figure 7-1. Instrumentation, Direct Contact Heat Exchanger Test

Instrumentation was provided consistent with the R&D role of the equipment. Figure 7-1 shows locations of temperature, pressure, and flow instruments. Temperatures were measured by type K thermocouples, except for several direct reading thermometers (not shown) used to adjust heat exchanger flows. Pressure drops were read by both vane type (Magnehelic) and damped type gages. Gas flow was metered by an orifice meter which had been calibrated with a large rotameter. In the later runs these two instruments were used in series. Liquid flow rates were determined by measurements from an orifice meter calibrated with Therminol 66 at operating temperatures.

RESULTS

The principal results of this test program are summarized in Table 7-1. The subscale contact heat exchanger was designed, built, and successfully operated to obtain the design characteristics required for the reference bottoming system in which an 1800 HP diesel engine is used. The heat transfer coefficients and actual pressure drop data were obtained for the subscale exchanger so that an initial design estimate can be made if Therminol 66 is used as the thermal fluid. It was determined that liquid Therminol 66 could be heated to a maximum temperature of 450 F when exposed to hot gases at temperatures up to 575°F. In the reference cycle, this temperature limitation would reduce the cycle efficiency by approximately 3 percentage points.

The phenomenon of evaporation/condensation was illuminated by the test results. Figure 7-2 (left side) illustrates the much smaller change in liquid temperature which results in the evaporating section compared to that in the condensing section of the packing. Similarly, Figure 7-2 (right side) shows the large effect on pressure drop of the evaporating fluid. These two effects, if not considered, would lead to large errors in sizing direct contact

Table 7-1. Contact Heat Exchanger - Test Results

1. TEST EQUIPMENT:	Experimental packed column of significant size fabricated and demonstrated - Available for further tests.
2. HEAT TRANSFER:	<p>Requirements of work statement satisfactorily completed.</p> <p>U overall data obtained.</p> <p>U less than estimated assuming no evaporation.</p> <p>Evaporation and condensation regions distinct.</p> $\frac{\text{Rate of Liquid Heating in Condensing Zone}}{\text{Rate of Liquid Heating in Evaporating Zone}} \approx \frac{4}{1}$ <p>Practical $(T_L)_{MAX} \leq 600 \text{ F}$ (T66 only) for $T_{gas} = 720 \text{ F}$</p>
3. SYSTEM DESIGN CONSIDERATIONS:	<p>Test results show: (1) Practical upper liquid temperature limit may be controlled by hydraulic (Δp) limitations rather than by wet-bulb temperatures. (2) Maximum liquid temperature controlled by evaporation effects.</p> <p>Operating conditions not yet optimized.</p> <p>Ageing and possible thermal degradation of Therminol 66 observed.</p> <p>Ability to predict performance of packed tower using other fluids and other conditions requires additional extended-duration runs.</p>

INNOVATIVE RESULTS REGARDING EVAPORATION/CONDENSATION
IN DIRECT CONTACT HEAT TRANSFER

Figure 7-2. Contact Heat Exchanger Test Results

equipment if conventional design correlations were used. The calculation of overall heat transfer coefficient for the heat exchanger is illustrated in Table 7-2 and data from all runs are summarized in Table 7-3.

It is recommended that testing using the direct contact exchanger be continued using hot compressed air and diesel exhaust gas as heat sources. Durability runs with Therminol 66 and with other fluids of low vapor pressure are recommended. The longer runs would supply data for developing a general design correlation in the presence of evaporation/condensation. Durability running with several heat transfer fluids would supply information on the effect of diesel exhaust and water on the fluids.

Table 7-2. Calculation of U_{AVG} Based on
Test 2 Experimental Data

$$U_{AVG} = \frac{\dot{q}}{A \Delta t_{lm}} = \frac{\dot{q}_A}{\Delta t_{lm}} \quad (\text{Eq. 3})$$

$$\dot{q} = \frac{(\dot{m}C_p)_L \Delta t_L + (\dot{m}C_p)_G \Delta t_G}{2} = 19.0 \frac{\text{Btu}}{\text{sec}} = 69,400 \text{ Btu/hr.}$$

$$\dot{q}_V = 69,400 / 6.0 \text{ ft}^3 = 11,580 \text{ Btu/hr-ft}^3$$

$$\dot{q}_A = 11,580 / 63 \text{ ft}^2 / \text{ft}^3 = 180 \text{ Btu/hr-ft}^2 \text{ packing surface}$$

$$\Delta t_{lm} = \frac{\Delta t_{top} - \Delta t_{bot}}{\ln \frac{\Delta t_{top}}{\Delta t_{bot}}} = 110^\circ \text{F}$$

$$U_{AVG} = 180,110 = 1.64 \text{ Btu/hr-ft}^2\text{-}^\circ \text{F}$$

Table 7-3. Summary of Experimental Heat Duties and U_{AVG}

Test	$\dot{q}$ (Btu/hr)	$\dot{q}_v$ Btu/hr-ft ³	$\dot{q}_A$ (Btu/hr-ft ²)	$(\Delta t)_{lm}$ °F	U_{AVG} (Btu/hr-ft ² °F)	$U_{predicted}$ (Btu/hr-ft ² °F)	
						Biphase	McEligot
1		14,370	228	68	3.2	--	--
2		11,360	180	110	1.6	--	--
3		6,870	110	32	3.4	13.0	9.1 - 18.5
4	Steady-state not achieved						

SECTION 8
NOZZLE TEST RESULTS

The basic concept of a Biphasic nozzle consists of the delivery of a liquid or liquid/gas mixture to the throat of a nozzle, such that in the expansion section, as pressure is reduced, a gas and a liquid phase exist together; as the gas expands and increases in velocity, it drags the liquid drops with it such that the liquid drops attain nearly gas velocity. The nozzle transforms heat energy in the entering fluids to kinetic energy in the exiting liquid. The moving liquid is subsequently transformed to torque in the Biphasic separator/turbine.

The theoretical velocity from such a nozzle is a function of the thermodynamic fluid enthalpy change, Δh :

$$V_{\text{THEOR.}} = \sqrt{2gJ\Delta h}$$

in which J = mechanical equivalent of heat = 778 ft-lb/BTU. Actual velocity (experimental) is determined by measuring nozzle thrust, and flow rates:

$$V_{\text{EXPT.}} = \frac{Fg}{\dot{m}}$$

Where F = measured thrust, $\dot{m}$ = flow rate total. Then nozzle efficiency is defined as

$$\eta_N = \left(\frac{V_{\text{EXPT.}}}{V_{\text{THEOR.}}} \right)^2$$

The nozzle test program consisted of measurement of the experimental velocity via thrust and flow measurement, and provision for calculating theoretical velocity: temperature and pressure conditions. Comparisons could thus be made among nozzle contours and among liquid mixing devices.

Inefficiencies in the nozzle result from fluid friction, irreversible expansion processes, and from differences between gas and liquid velocity (drag losses). The baseline Biphase system anticipates a nozzle efficiency in the range 0.75 to 0.85. The drag losses are essentially a function of the size of the liquid drops; Figure 8-1 from Reference 4 illustrates the importance of small drops. Accordingly, the Biphase nozzle definition must include the mechanism by which sufficiently small drops are formed in the expanding gas past the nozzle throat. In the case of the two component diesel-bottoming nozzle, fine droplets of oil in water, or water in oil, are produced just ahead of the nozzle contraction section by a mixing device. In accordance with the above, the nozzle test program considered different nozzle contours first, and then considered different upstream mixing devices. The experimental equipment to make the required measurements was developed during the program, then basic comparisons were made. Directions for fruitful additional R & D activity were identified during the program, resulting in an extensive company-funded research program on nozzle mixing and shape determination.

Losses discussed above (fluid friction) suggest a nozzle as short as possible. When aimed at the separator wheel in the Biphase turbine, the nozzle must be compact to minimize geometrical losses to turbine efficiency, and the "footprint" or nozzle exit pattern must be small to minimize separator size.

Figure 8-1. Effect of Length and Initial Drop Diameter on Exit Velocity, With and Without Breakup

All of the above considerations enter into a two-phase nozzle computer program used at Biphase to size and contour the nozzle component.

NOZZLE CONTOUR TESTS

The nozzle test program started with a series of contour tests. For these, three nozzles were fabricated, each with a different contour. The nozzles were sized for the baseline subscale (50HP) turbine: each had a throat diameter of 0.200 inch, Figure 8-2. Mark III nozzle had a conventional rounded throat contour and a bell-shaped expansion section, as plotted by the design program mentioned above. Mark II nozzle had the same entrance, throat and exit dimensions, but had a sharp throat (intersection of two cones) and straight conical expansion section. This contour was suggested by a closed-form solution to the nozzle equations. Mark I was similar to Mark II but only 1/3 the length. The Mark II and I nozzles are clearly easier to fabricate, and the short Mark I may have advantages in packaging with the turbine.

The three test nozzles were fabricated from electroformed nickel, which was deposited on an aluminum mandrel. The mandrels, shown in Figures 8-3 and 8-4, were the precise size and finish of the desired interior of the nozzle. After electroforming the nickel, the aluminum mandrel was dissolved out.

These three nozzles were calibrated, and preliminary off design data were obtained, by flowing water and air in the simple facility shown in Figure 8-5. Continued operating experience and off-design data were obtained with water-steam mixtures in the calibration rig shown in Figure 8-6. Maximum pressure

MARK III - ROUNDED THROAT, BELL-SHAPE EXPANSION

MARK II - SHARP THROAT, CONICAL EXPANSION

MARK I - SHORT; SHARP THROAT, CONICAL EXPANSION

Figure 8-2. Experimental Nozzle Contours

Figure 8-3. Mandrels for Mark I, II, and III Biphase Nozzles Sized for 50 HP Test Turbine

Left:
Aluminum
Mandrel

Right:
Electroformed
Nozzle

Figure 8-4. Biphase Nozzle Mark I - Aluminum Mandrel and Electroformed Nozzle

Design Conditions:

Dowtherm A, 1.5 lb/sec @ 660 F, 1200 psia inlet
Water, 0.3 lb/sec @ 560 F, 1200 psia inlet

Nozzle Diameters:

Inlet	0.539 in
Throat	0.200 in
Exit	1.043 in

Figure 8-5. Biphase Nozzle Mark I (Short, Sharp Throat)
Flowing Water and Air

- ① Test Biphase Nozzle
- ② Hinged Thrust Platform
- ③ Load Cell (Thrust)
- ④ Water/Steam Separator
- ⑤ Control and Measurement Trailer

Figure 8-6. Water/Steam Nozzle Calibration Rig

available in this facility was 100 psig, compared to nozzle design pressure of 800 psia. Flow capacity was up to 2 lb/sec. Useful calibration and operating information were obtained; however. all the data obtained were at off-design conditions.

A facility for nozzle testing at more nearly design conditions was therefore fabricated. The facility, Figure 8-7, includes high pressure pumps for water and the heat transfer fluid (Therminol 66 was tested first). The Therminol is heated to the test temperature by recirculating through the Biphasic gas fired furnace. The facility capability is as follows:

Heat Transfer Fluid:	Maximum Temperature	600F
	Maximum Pressure	800 psig
	Maximum Flow	20 GPM
Water:	Maximum Pressure	800 psig
	Maximum Flow	4 GPM

The nozzle test set-up is illustrated by the schematic, Figure 8-8. The nozzle is mounted on a force table, so that thrust may be measured, and connected via flexible lines to the Therminol and water systems. The nozzle discharges into a receiving tank, which is large enough to contain the total Therminol charge from the pressure system. Water content of the nozzle discharge leaves the receiving tank at the top as steam, and is condensed for measurement. After the run, the fluid (Therminol) is pumped back to the Therminol tank.

The heat-up cycle involves pumping the fluid through the furnace and back to the tank. This operation is conducted under 50 to 75 psig pressure to assure pump suction. The water is not pre-heated. When Therminol is at test temperature, the recirculating valves are closed, and Therminol and water are pumped to the nozzle via the flexible lines. This system is capable of

Figure 8-7. Biphase Nozzle Test Facility

Figure 8-8. Nozzle Test Schematic - Initial Arrangement (7-31 to 8-23)

pressures up to 1000 psig. The Therminol and water are mixed upstream from the nozzle (as shown) in static mixers, such as Koch or Kenics, and also an injector type (Penberthy).

The initial portion of the test program, Table 8-1, was conducted with Therminol 66 (Monsanto Co.), in accordance with the baseline system definition. The Therminol is mixed with water in the nozzle, then discharged into the receiving tank, where the water should separate from the hot fluid. In practice, it was found that Therminol 66 and water formed some kind of a combination which was difficult to separate at temperatures below approximately 400F. This material caused test difficulties by lowering pump NPSH and by increasing in pressure during the heating cycle in the closed heating system. It was found that an extra blowdown of the fluid only through the nozzle was required to eliminate the water, before each data run could be essayed. In addition, the Therminol/water was apparently a poor lubricant; several instances of piston/liner seizure in the positive-displacement pump were recorded. It should be noted that all of the difficulties with Therminol 66 discussed above are related to the test equipment (pumps, separating tank) and may not be significant to the baseline Biphase cycle.

Testing continued with the alternate heat transfer fluid, Caloria HT43 (Exxon Co.), per Table 8-2. Caloria is petroleum based, and separated easily from water. The oil is also apparently a better lubricant in the pump. With the test equipment in use, Caloria resulted in a higher test frequency.

Theoretical nozzle analyses assume that (a) the two fluids are thoroughly mixed upstream of the nozzle, so that small liquid drops will be formed, and also (b) that the mixing devices will have some pressure drop, so that the fluid destined to be a gas (water) will start flashing before the nozzle entrance. The initial test results indicated that

Table 8-1. Nozzle Tests - Therminol 66/Water

NOZZLE	1978 DATE & RUN		THERMINOL			WATER			NOZZLE ENTRANCE		THRUST LB.	MASS RATIO THERMINOL/ WATER	REMARKS
			P ₁ PSIG	T ₁ F	FLOW GPM	P ₂ PSIG	T ₂ F	FLOW GPM	P ₆ PSIG	T ₆ F			
MARK 3	8-1	(1)	300	393	20	300	150	1.9	296	334	33.	9.2	
	8-4	(4)	230	398	18	230	146	1.8	230	355	9.3	8.6	
	8-7	(2)	420	508	20	420	242	1.15	420	460	28.	14.	
	8-8	(1)	475	518	20	475	235	1.0	475	478	30.	19.6	
	8-8	(2)	450	500	20	450	145	3.7	450	398	26.	4.4	
	8-8	(3)	200	362	15.5	-	-	-	200	362	8.	-	
	8-8	(4)	400	603	15.7	400	156	3.9	395	432	27.	3.1	Therminol pump sized-pressure oscillations
	8-16	(2)	-	-	-	-	-	-	-	-	-	-	
	8-16	(3)	470	465	22.5	470	190	2.2	470	410	26.	9.1	
MARK 2	8-1	(2)	180	401	14.6	180	159	2.2	180	356	12.	5.6	
	8-3	(1)	130	344	13.5	130	166	.5	130	339	4.	23.7	
	8-3	(2)	400	388	20.	400	240	1.0	400	362	16.	17.1	
	8-4	(1)	320	409	19.6	320	140	1.9	320	344	12.	8.5	
MARK 1	8-4	(2)	235	378	20.	-	-	-	235	364	13.	-	
	8-4	(3)	320	400	20.	320	137	1.8	320	354	17.	9.2	

Table 8-2. Nozzle Tests - Caloria/Water (Sheet 1 of 2)

NOZZLE	1978 DATE & RUN	THERMINOL			WATER			NOZZLE ENTRANCE		THROTTLE VALVE P PSI	THRUST LB.	MASS RATIO THERMINOL/ WATER	REMARKS
		P ₁ PSIG	T ₁ F	FLOW GPM	P ₂ PSIG	T ₂ F	FLOW GPM	P ₆ PSIG	T ₆ F				
MARK 3	8-21 (1)	395	411	22.0	395	132	1.7	390	374	-	13.	9.4	Instrument nozzle
	8-22 (1-3)	410	520	21.5	410	200	1.8	405	470	-	29.	8.7	
	8-22 (1-6)	430	518	21.9	430	144	1.95	425	489	-	24.	8.2	
	8-23	600	495	22.0	600	175	1.85	575	468	25	35.	8.7	Throttle valve installed
	8-24 (1)	460	502	21.5	465	105	3.3	400	400	60	27.	4.7	
	8-24 (2)	580	510	21.0	580	160	1.5	555	455	25	36.	10.2	
	8-25 (1)	605	520	20.5	610	140	1.3	580	446	25	46.	11.5	Transient
	8-25 (2) slice 1	700	528	22.0	700	201	1.2	655	475	45	50.5	13.4	
	8-25 (2) slice 2	500	512	22.0	530	140	1.6	430	442	100	30.	10.0	

Table 8-2. Nozzle Tests - Caloria/Water (Sheet 2 of 2)

NOZZLE	1978 DATE & RUN		THERMINOL			WATER			NOZZLE ENTRANCE		THROTTLE VALVE P PSI	THRUST LB.	MASS RATIO THERMINOL/ WATER	REMARKS
			P ₁ PSIG	T ₁ F	FLOW GPM	P ₂ PSIG	T ₂ F	FLOW GPM	P ₆ PSIG	T ₆ F				
MARK 3	9-8	(1)	515	500	20	515	135	1.95	475	430	40	25.0	7.5	
	9-8	(2)	550	495	19.5	550	132	1.40	520	450	30	28.0	10.1	
	9-8	(3)	550	493	19.5	550	142	1.06	520	460	30	27.0	13.4	
	9-11	(1)	410	508	20.5	410	170	2.1	370	420	40	21.0	7.1	
	9-11	(2)	850	520	21.0	850	127	2.8	400	422	450	34.0	5.5	
	9-11	(3)	800	510	19.5	800	120	2.3	420	404	380	28.0	6.2	
	9-11	(4)	400	513	20.5	400	245	1.1	340	502	60	13.0	13.6	
	9-11	(5)	340	511	19.5	340	225	1.0	220	504	120	11.0	14.2	
	9-11	(6)	330	520	19.0	330	215	1.0	220	515	110	18.0	13.8	
	9-11	(7)	565	510	20.5	565	130	2.1	430	424	135	28.0	7.1	
	9-11	(8)	535	504	20.5	535	118	2.1	430	426	105	29.0	7.1	
MARK 2	9-12	(1)	750	532	19.5	750	250	1.25	650	470	100	41.0	11.4	
	9-12	(2)	690	527	19.5	690	220	1.15	600	474	90	39.0	12.4	
MARK 1	9-12	(3)	650	523	20.0	650	218	1.51	560	455	90	38.0	9.6	
	9-12	(4)	630	522	19.5	630	190	1.40	530	454	100	36.0	10.1	

criterion (b) was not being realized; accordingly, the test set-up was altered by adding a throttle valve ahead of the nozzle, as shown in the revised schematic, Figure 8-9. This method of securing gas at the nozzle entrance was only partially successful.

A correlation of the data obtained with caloria/water is shown in Figure 8-10. The data show:

1. Correct power trend with mass ratio.
2. No discernible difference among the Mark I, II, and III nozzles.
3. Apparent lack of complete success in mixing (producing small drops), and in producing flashing before the nozzle by throttling.

The same correlation for Therminol 66/water, Figure 8-11, was not successful, reflecting the experimental difficulty experienced with this combination. The Therminol 66 tended to form combinations with water which interfered with heating and with separation in the gravity tank; the Therminol 66/water combination apparently was such a poor lubricant that pump seizure was frequently experienced. Nozzle and turbine testing with Therminol 66 were discontinued as a result of these experiences.

NOZZLE AND MIXER TESTS

To continue nozzle development and to initiate comparisons of mixing devices, it was determined to fabricate a new nozzle specifically designed for the available test conditions (previous tests had been run at severely off-design flow and pressure). Steam was used directly instead of water, as the high vapor pressure fluid, to minimize problems with liquid-liquid

Figure 8-9. Nozzle Test Schematic - Second Arrangement (8-25 to 9-12)

Figure 8-10. Experimental Nozzle Power - Caloria/Water

61-8

Figure 8-11. Experimental Nozzle Power - Therminol 66/Water

emulsions and to help in fluid mixing. Accordingly, the "standard" nozzle shown in Figure 8-12 was fabricated for:

Caloria Flow: 0.787 lb/sec @ 350F
Steam Flow: 0.121 lb/sec @ 105 psia saturated
Mass Ratio: 6.5

This "standard" nozzle was incorporated into the test equipment, Figures 8-13 and 8-14, and tests were continued with Caloria and steam to characterize this nozzle for turbine testing and to determine the performance of upstream mixing devices.

Four different static mixing devices were fabricated for test with the "standard" nozzle, Table 8-3.

1. The "Needle-Jet" mixer, Figure 8-15, was patterned after a mixer previously used successfully at the Jet Propulsion Laboratory. The mixer injector is made from 37 tubes, 3/32-inch diameter. Steam flows inside the tubes, and Caloria on the outside. The Caloria exits the injector through the sketched passage formed by the cusps between tubes. This mixer is designed to accomplish some heat transfer between the fluids as well as the mixing which takes place at the mixer exit. The mixing element is shown in Figure 8-16.
2. The Penberthy "steam ring" mixer is a commercial device which injects an annular stream of Caloria into the flowing steam, and completes the mixing by turbulence. Figure 8-17 illustrates this mixer.
3. and 4. The large and small "radial" mixers, Figure 8-18, are made from a free-flowing steam tube, with Caloria injection taking place through 1/16 holes from an annular plenum. The large radial mixer is a 1½-inch tube, with 16 holes each 1/16-inch diameter; the small radial mixer is a 1-inch tube, with 8 holes. The small radial mixer was not used except for calibration tests.

Figure 8-12. Test Nozzle Designed for Specific Turbine Test Conditions

Figure 8-13. Nozzle Test Equipment

Figure 8-14. Test Nozzle - Caloria/Steam Two-Phase Flow

Table 8-3. Experimental Mixers

Mixer Part	Needle Jet	Penberthy "Steam Ring"	Large Radial Mixer	Small Radial Mixer
Caloria Passage				
Shape	Cusp between tubes	Annular Orifice	16 - $\frac{1}{16}$ " holes	8 - $\frac{1}{16}$ holes
Area, in ²	0.013	0.24	0.049	0.0245
Steam Passage				
Shape	37 3/32 dia tubes	Center and annular outside of annular Caloria Orifice	Center "Straight through" flow	Center "Straight through" Flow
Area, in ²	0.126	1.70	1.77	0.80

Figure 8-15. Mixer/Injector for Caloria - Steam

Figure 8-16. Mixer/Nozzle for Caloria - Steam

CALORIA ORIFICE AREA = 0.24 in^2
 STEAM FLOW AREA = 1.70 in^2

Figure 8-17. Penberthy "Steam Ring" Mixer

	LARGE RADIAL MIXER	SMALL RADIAL MIXER
STEAM TUBE SIZE, IN.	1-1/2	1
NUMBER OF 1/16 HOLES	16	8

Figure 8-18. Radial Flow Mixers

Calibration tests were made with the needle jet mixer, to determine flow capability and to observe mixing. Figure 8-19 illustrates mixing performance, which was qualitatively judged to be good.

The concluding nozzle test series utilized the test nozzle (Figure 8-12) which has a "sharp" throat, that is, the intersection of two cones. The four different mixers described above were tested with steam-Caloria. The test set up with the Penberthy mixer is shown in Figure 8-20.

Nozzle data are tabulated in Appendix A. The first series, Table A-1, featured the large radial mixer. Tests 2-2-79 through 2-7-79 were made with the mixer close-coupled to the nozzle. Starting with test 2-9-79 (1001), an 8-inch long spacer was added between the mixer and the nozzle, in order to determine the effect of longer contact time of the mixed fluids. The data do not show a measurable effect of this change. At test 2-9-79 (0001), the steam throttling was all accomplished at the boiler valve, to assist in steam flow measurement with dry steam. No effect on mixer performance was expected from this change. Data from the large radial mixer runs are plotted, Figure 8-21, as a function of mass ratio. At the design mass ratio (6.5), the nozzle efficiency appears to be in the range 50-60 percent. Considerable data scatter is evident; to some degree, this is caused by the blowdown operation of the Caloria facility, which limits run duration to approximately one minute. Under these conditions, some of the plotted points represent transient conditions. Those points near the start of a blowdown, listed in the tabular data, have been omitted from the plots for this reason.

MIXER
ELEMENT

AIR - WATER
LOW FLOW

AIR - WATER
HIGH FLOW

Figure 8-19. Mixer Calibration Tests with Air and Water

Figure 8-20. Nozzle Test Set-up on Force Table, with Penberthy Mixer

Figure 8-21. Test Results, Large Radial Mixer, Test Nozzle

The Penberthy mixer data are presented in Table A-2 in the Appendix. Data are plotted, Figure 8-22. Some scatter of the early points was caused by difficulty in controlling Caloria flow during these runs; this was corrected by rearrangement of the control valves. The trend appears to show a nozzle efficiency between 50 and 60 percent at design mass ratio of 6.5.

Tests with the needle-jet mixer are tabulated in Table A-3 in the Appendix, and data are plotted in Figure 8-23. As with the other two mixer types, the efficiency at mass ratio of 6.5 was between 50 and 60. Test difficulties were experienced with plugging of the small holes, and procedure differences were observed between the use of nitrogen or argon for pressurizing the Caloria tank. These will be discussed below.

The three static mixer types tested all yielded approximately the same performance, such that ranking is not possible at this time. The measured efficiencies agreed quite well with the expected values. Extrapolations to higher nozzle pressures anticipated in the prototype turbine indicate that the design goals of 75 to 85% nozzle efficiency can be met.

Some additional test nozzle runs were made with the radial mixers modified by including packs of 100 mesh copper screen, in an effort to secure finer liquid breakup. These tests were not successful, and are not reported, because the fine mesh screen rapidly plugged up with decomposition products from the Caloria heat transfer fluid. These products included a gummy sludge and scale particles. This plugging was observed at the mixer even though the facility was provided with an adequately-sized 200 mesh filter. The experience was that the facility

INLET PRESSURE >50 PSIG, OPEN SYMBOLS

Figure 8-22. Test Results, Penberthy Mixer, Test Nozzle

INLET PRESSURE >50 PSIG, OPEN SYMBOLS

Figure 8-23. Test Results, Needle Jet Mixer, Test Nozzle.

filter became plugged and damaged, even in the very short operating times. This same difficulty was experienced with the needle jet mixer: the Caloria passages, the cusps shown in Figure 8-15, became completely plugged during the first 11 runs (flow was maintained through damaged static seals in the mixer). For the remainder of the needle jet tests, the Caloria passage cusps were drilled out by a 1/64-inch diameter drill. It was concluded that with Caloria, and probably with other petroleum-based heat transfer fluids, that mixer types dependent on small passages or fine screens could not be considered.

At the end of the test program, several additional tests were made with the test nozzle sharp throat (0.580-diameter) rounded to resemble a conventional nozzle shape. The throat was first rounded to 0.584 diameter, then 0.589. There was no noticeable change in performance.

The nozzle test runs were made by pumping the Caloria from a 14 cubic foot supply pressure tank. The Caloria was brought up to test temperature by recirculating (before the test) through a gas-fired furnace, using the vane-type Blackmer pump. Observations were made of this facility set-up to determine the water content of the Caloria resulting from previous runs and contact with steam. Too great a water content resulted in pressure rise during the heating cycle, and difficulty with pump suction. The test experience was that nozzle test into the separating tank at temperatures above 375F resulted in Caloria sufficiently free of water that no test difficulties were caused. Below 375F it was usually necessary to blow the Caloria inventory down through the nozzle, with no steam added, to remove the bound water. It was also found that tank pressurization with argon gas resulted in much less pumping difficulty (presumably caused by dissolved gas) than pressurization by nitrogen.

Conclusions of the nozzle test program are:

1. Efficiencies in the range 50 to 60 percent were demonstrated, indicating that the design goal of 75 to 85% should be attainable in the prototype turbine.
2. The three types of static mixer tested all gave similar performance.
3. The petroleum-based fluid Caloria HT43 cannot be used in mixer types dependent on small orifices or fine mesh screens.
4. Mixer and nozzle development in a facility with longer operating times is desirable in order to obtain non-transient data.

SECTION 9

TURBINE TEST PROGRAM

INTRODUCTION

The test turbine design details were determined following the identification of a baseline nozzle from the companion nozzle test program. The nozzle, with a 0.58 diameter throat, was designed for a total flow of 0.9 lb/sec., 0.787 lb/sec of Caloria heat transfer fluid at 350F, and 0.121 lb/sec saturated steam at 105 psia. Turbine power under these conditions is in the range of 8 to 15 HP, depending upon allowable speed and turbine liquid pickup efficiency.

The test turbine utilized a simple liquid impulse element (called a "U-tube") to convert the kinetic energy of liquid on the separator rim into torque. The spent liquid was not delivered from the turbine under pressure, which is an alternate capability of this turbine when using a "pitot" element. The steam, once separated, was simply condensed; gas blading, which could produce additional shaft power, was not incorporated in this test setup.

Caloria HT43 (Exxon Co.) was selected as the low-vapor pressure heat transfer fluid on the basis of comparatively better performance (compared to Therminol 66) determined in the nozzle test program and the heat exchanger test program. Saturated steam from the Biphase boiler was the high-vapor pressure fluid.

The Biphase Energy Systems' facility for testing the diesel bottoming turbine included a trailer-mounted dynamometer and control subsystem, a gas-fired furnace to heat the Caloria fluid, Caloria pumps, an oil-fired steam boiler, a steam condenser, and

all necessary instrumentation and accessories. Turbine safety while running was monitored by proximity and vibration readout and by overpressure and overtemperature cutoffs.

Successful runs were made over a variety of conditions of temperature, pressure, mass ratio, and power level. These will be reported in the following sections. Figure 9-1 illustrates the turbine mounted for test in a Biphase trailer facility.

TEST TURBINE

The diesel bottoming test turbine was built up using an impulse turbine element sized to match the flow capability of the selected nozzle, and the nozzle developed during the nozzle test program. The turbine housing, separator wheel, and shaft mountings were adopted from previous programs. The turbine assembly is shown in Figure 9-2. When mounted for test, the shaft is vertical, with the nozzle above the separator and the turbine below. Figure 9-2 shows that the liquid turbine pickup is bolted to a shaft, mounted to the housing with three bearings, which transmits torque via the pulleys at the bottom. The separator wheel is mounted on three bearings, and is not restrained in any way. The rim speed of the separator may approach the velocity of the working fluid leaving the nozzle; this determines the design rim speed of the separator. The bearings are oil lubricated, and separated from the working fluid by the four Clipper Lip seals shown on the drawing. The turbine/pickup assembly and the separator disc/support assembly are statically and dynamically balanced. Figure 9-3 is a photograph of the rotating elements of the machine.

Working fluid passage through the turbine is as follows (referring to Figure 9-2): The two-phase fluid (Caloria liquid, steam) enters via the Biphase nozzle, to impinge on the inner rim of the separator. The gas and liquid are separated under high "g" forces. The gas (steam) exits upward through a duct which is connected to a condenser. The liquid (Caloria) forms a layer on the inner rim of the separator wheel. This liquid, injected with the exit velocity

Figure 9-1. Diesel Bottoming Turbine in Biphase Trailer Facility

Figure 9-3. Turbine Rotating Elements

of the nozzle, causes the free-wheeling separator wheel to rotate. The liquid flows through holes in the web of the separator wheel, and would eventually overflow the rim if not removed by the liquid turbine pickup. The turbine pickup used with the diesel bottoming turbine dips into the liquid layer on the separator rim and reverses the flow direction 180 degrees, producing torque on the turbine shaft. The reversed flow has small velocity relative to the housing (if the turbine were rotating at half separator speed, the relative velocity would be zero); this flow falls by gravity to the housing and drains into a sump. It is noted that the test turbine shaft has provision for a pitot-type pickup, which can direct the liquid to the center of the shaft for discharge under pressure. This feature was not used for the diesel bottoming setup.

The operation of the impulse turbine pickup was studied in a separate test rig using water on a mechanically driven separator and a stationary pickup. Figure 9-4 illustrates the pickup. When held stationary, exit velocity of the liquid stream is approximately the same as inlet velocity, but turned 180 degrees. The water test rig was used in a Biphase program to study losses in the wake and to help adjust wheel pickup geometry. The pickup shown in Figure 9-4 is an impulse type similar to the diesel bottoming part. Figure 9-5 illustrates the impulse pickup and also the pitot type; these are interchangeable in the test turbine. The impulse pickup used in the reported test program is shown in Figure 9-6. This pickup has a small wake deflector on the bottom. A deflector for the exit stream, shown in Figure 9-6, was added during the test program.

The turbine nozzle, Figure 9-7, was selected from the nozzle test activity, reported in another section of this report. The nozzle tests were made with several different static mixers upstream of the nozzle, to initiate the formation of small drops in the steam/Caloria mixture. For the turbine test program, the upstream

Figure 9-4. Liquid Impulse Turbine

"Pitot" Type
Pickup

Impulse or "Scoop"
Pickup

Figure 9-5. Turbine Pickups

Figure 9-6. Impulse Pickup,
Diesel Bottoming Program

Figure 9-7. Test Nozzle Designed for Specific Turbine Test Conditions

mixer used was a Koch static mixer. The nozzle design conditions are:

<u>Caloria HT 43 Fluid</u>	Flow	0.787 lb/sec
	Temperature In	350F
<u>Saturated Steam</u>	Flow	0.121 lb/sec
	Temperature In	331F
	Pressure to Mixer	105 PSIA
<u>Nozzle Inlet</u>	Total Flow	0.908 lb/sec
	Mass Ratio	0.787/0.121 = 6.5
	Temperature	347.5F
	Pressure	95 PSIA
<u>Nozzle Outlet</u>	Pressure	14.7 PSIA
	Ideal Velocity	1047 ft/sec
	Expected Velocity	621 ft/sec

The following are turbine design parameters:

Separator Wheel Diameter	31.5 inches
Nozzle Angle to Separator Rim Tangent	23 deg = α
Nozzle Angle to Plane of Separator Web	15 deg = β

$$\text{Separator Rim Velocity} = V_{\text{RIM}} = V_s (\cos \alpha)^2 (\cos \beta)^2 F_L$$

where

$$V_s = \text{Nozzle spouting velocity}$$

$$F_L = \text{Factor for windage and friction}$$

$$V_{\text{RIM}} = 621 (\cos 15)^2 (\cos 23)^2 (0.85) = 417 \text{ ft/sec}$$

$$\text{Separator RPM} = 3034 \text{ RPM}$$

For the turbine speed at half the separator speed, 1517 RPM, the expected torque at design conditons is 27.8 lb-ft, and power is 8.02 HP.

TEST FACILITY

The turbine test facility diagram, Figure 9-8, shows all plumbing and instrumentation. Steam is supplied to the mixer from the Biphasic boiler facility via flow control and metering devices. The Caloria heat transfer fluid is pumped through the tubes of a gas-fired furnace, metered, controlled and filtered, and delivered to the nozzle. Steam/oil separation and operation of the impulse turbine pickup are detailed elsewhere. The separated steam is ducted to a water-cooled condenser, and condensate is not reused. The caloria fluid exits the turbine casing by gravity to a sump, and is then pumped back around the circuit. An inventory of 15-20 gallons of Caloria is maintained in the sump. The pump is a Blackmer vane type, with a jet pump assisting the suction pressure. Caloria temperature up to 500F and steam pressure up to 150 psig are available with this setup.

The turbine shaft is connected to a hydraulic pump, which provides torque measurement and also gives speed control during start and stop transients. An electrical generator was also attached to the turbine shaft for some runs. Figure 9-9 is an overall view of the test trailer, and Figure 9-10 shows the instrument panel.

Safety devices consist of turbine speed control (by means of the hydraulic system); the separator speed is adjusted by nozzle flow and mass ratio. Bentley proximity and vibration gages were mounted radially and tangentially with respect to the separator wheel. The separator speed was limited to 4000 RPM, at which point stress levels were still conservatively low.

The operating procedure for typical runs was as follows: The system was preheated by pumping Caloria through the furnace and through the nozzle and separator/turbine. The flow of the liquid alone did not cause separator rotation. Then the steam flow was initiated and the separator was brought up to speed with the turbine held (this gave a stall torque measurement). Finally, the turbine rotation was initiated, and power measurements were made.

Figure 9-8. Diesel Bottoming Turbine Test Facility

Figure 9-9. Turbine Test Facility

Figure 9-10. Instrument Panel, Turbine Test Facility

Variables from run to run included flow rates, mass ratios, Caloria temperature, and ratio of separator to turbine speed, u/c . The system was stopped by shutting down steam flow.

TEST RESULTS

Turbine runs were made on five days: March 2, 5, 7, 8, and 9, 1979. Data obtained is tabulated by test numbers for each day; all recorded test numbers are presented in Appendix B. The following ranges of the variables were observed:

Turbine Power, HP	0-10.74
Separator Speed, RPM	0-3805
Turbine Speed, RPM	0-1486
Turbine Speed/Separator Speed, u/c	0.22-0.64
Total Flow, lb/sec	0.337-1.239
Mass Ratio, Caloria Flow/Steam Flow, r	2.66-13.71
Caloria Temperature, F	350-480
Nozzle Inlet Pressure, PSIG	20-100

Operation of the test hardware and the test facility was, in general, successful and controllable. Maximum power measurements were made in the expected range. Neither hardware failure nor damage was experienced. Test results are presented as follows:

Figure 9-11. Turbine Power Measurements.

$$\begin{aligned}
 m_l &= \text{Liquid flow rate, lb/sec} \\
 V_{\text{RIM}} &= \text{Separator rim velocity, ft/sec} \\
 N_s &= \text{Separator speed, RPM} \\
 u/c &= \text{Speed ratio, turbine to separator} \\
 f\left(\frac{u}{c}\right)\left(1-\frac{u}{c}\right) &= \text{A correlation factor for } u/c \text{ variation. At maximum power, } \\
 &\quad u/c=0.5 \text{ and this factor equals 1.}
 \end{aligned}$$

$$\left(\frac{m_1 V_{RIM}^2}{2g} \right) 4 \left(\frac{u}{c} \right) \left(1 - \frac{u}{c} \right) = \frac{m_1 N_s^2 \left(\frac{u}{c} \right) \left(1 - \frac{u}{c} \right)}{468,518}$$

Figure 9-11. Diesel Bottoming Turbine Power Measurements

Figure 9-12. Turbine Power Measurements plotted against Nozzle Inlet Pressure. The parameter u/c , speed ratio, turbine to separator, is used to group the data in three ranges.

Figure 9-13. Turbine Power Measurements plotted against total flow, with the mass ratio r as a separating variable.

Figure 9-14. Observed Separator RPM. Typical points, with various values of u/c and r , are plotted against nozzle inlet pressure.

Figure 9-15. Calculated vs Observed Separator Speed.

It was observed that operation at the turbine speed expected to give maximum power, half separator speed, or $u/c=0.5$, was not possible with the available impulse turbine pickup. At a given separator speed, the turbine speed could be increased from stall up to a u/c of approximately 0.43, at which point further turbine speed increase would result in flooding the separator wheel. This flooding slowed the separator due to liquid friction with the housing. This condition caused no damage, but required restabilization of the run. It was also found that stable operation could be obtained at higher turbine speeds, u/c greater than approximately 0.58. Flooding occurred in the u/c range of 0.43 to 0.58. In this range, the relative velocity of the liquid jet issuing from the impulse pickup is low (at $u/c=0.5$, the relative velocity relative to the housing is zero). It is postulated that the exiting jet was not directed sufficiently away from the separator rim or away from the turbine disc, so that low jet velocities the flow interfered with these rotating members, causing flooding. The test turbine was not equipped with observation ports; accordingly, this phenomenon was not directly observed. However, the company-sponsored research and development program has built hardware and provided instrumentation to develop the pickup geometry, and pickup designs are expected to be available in the future, which

Figure 9-12. Turbine Power Measurements

Figure 9-13. Turbine Power Measurements

9-20

Figure 9-14. Observed Separator RPM

Figure 9-15. Calculated vs Observed Separator Speed

will operate at any u/c value. During the current test program, one effort was made to help direct the exit jet; the baffle shown in Figure 9-6 was added to the pickup for the tests on 3-9-79. This one modification did not prevent flooding at u/c values above 0.43.

CONCLUSIONS - TURBINE TEST

1. The basic feasibility of operating a Biphase separator/turbine with heat transfer oil and steam has been demonstrated.
2. Turbine performance at the test power levels substantiates the calculated diesel bottoming baseline system.
3. Turbine operation at off-design conditions is practicable.
4. The impulse turbine pickup requires development under conditions where it may be observed visually. Operation at u/c near 0.5 requires additional development. (This effort has started under Biphase sponsorship.)
5. The heat transfer fluid, Caloria HT43, is suitable for use with steam under the test condtions.
6. The design of the full scale baseline diesel bottoming turbine may proceed with confidence.

SECTION 10

CONCLUSIONS AND PROGRAM PLAN

The major conclusion evident from the Biphasic system analysis and development effort conducted to date is that the concept and hardware are ready to proceed to the defined second stage, which is pre-commercialization. Test programs of the system components, in hardware of full-size or significant size, were generally successful, as detailed elsewhere in this report. Areas requiring additional development were defined in some instances; the recommended action is being pursued under company-sponsored programs or is included in the Year 2 follow-on effort from D.O.E. These defined areas include:

1. A completely satisfactory static mixer has not yet been demonstrated, such that nozzle performance may be increased. The mixer for oil/water has been specified (small drops) and company-funded development is underway to achieve the requirements. If the goals are not achieved with static mixer elements, then dynamic and other approaches will be investigated. These research activities are scheduled prior to the diesel bottoming Year 2 program need for design information.
2. The low-vapor-pressure heat transfer fluid best characterized at present is Caloria HT43. This fluid will probably not meet the upper temperature limit of 660F in a closed system, and will definitely not meet 660F in the open system with a direct contact heat exchanger. Improvements in the heat transfer fluids are being sought in the industry. If these are not found, then Caloria can be used for Year 2 and Year 3

tests at reduced upper temperature, at the sacrifice of a few points of cycle efficiency. This problem will be addressed in the first month of Year 2.

3. Detailed design of the impulse-type turbine pickup will require test fixtures to visualize the flow patterns, especially at U/C ratios near 0.5. This work is scheduled in Year 2.
4. The non-steady state operating parameters of the Biphasic cycles have not been completely defined. These include start, stop, and turndown. This effort is scheduled in the follow-on Year 2.

The program plan for the Year 2 follow-on is illustrated, Figure 10-1. In the second year program, the separator/turbine is designed and fabricated for testing. The full-scale separator/turbine, condenser and pumps are operated closed loop during development testing. The second year also includes the design of two different kinds of primary heat exchangers, to transfer heat from the diesel exhaust to the heat transfer fluid. One is a relatively conventional extended surface exchanger; the other a direct contact exchanger of the type explored in Year 1. The second year program includes a recommended engineering evaluation of the existing contact exchanger, to determine design parameters in the presence of fluid evaporation and recondensation, and to determine fluid compatibility and endurance in this service. The decision as to which exchanger to build for the third year program will be made late in Year 2.

The third year proceeds with endurance and reliability tests using the Year 2 test hardware and augmented with the selected heat exchanger. This equipment is taken to the field and installed on a 1800 HP diesel engine for a completely applicable demonstration. Formation of a partnership agreement with the owner of the engine is a task requirement in Year 2.

Although the above program is based on a specific reference system, it should be emphasized that options remain in the cycle upper liquid temperature (may be limited by fluid properties), selection of heat transfer fluid, placement of the pump on the hot or cold side of the heat exchanger, and condensing temperature level. The valid system studies also include application to gas turbine engines, and to industrial furnaces and kilns. During the Year 2 and 3 programs, favorable results are expected to suggest extensions to these other sources of waste heat.

Figure 10-1. Program Plan, Demonstration of Biphase Diesel Bottoming Cycle Second Year

SECTION 11

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APPENDIX A

Nozzle Data

h = Enthalpy, BTU/lb

r = Mass Ratio, Caloria Flow/Steam Flow

V_{expt} = Measured exit velocity, ft/sec

$$V_{\text{expt}} = Fg/\dot{m}$$

where F = measured thrust, lb,

$\dot{m}$ = total flow rate, lb/sec

V_{theor} = Theoretical (isentropic) Velocity, ft/sec

$$\eta = \text{Nozzle Efficiency} = \left(\frac{V_{\text{expt}}}{V_{\text{theor}}} \right)^2$$

Table A-1. Diesel Bottoming - Nozzle Tests 1979 Large Radial Mixer (Sheet 1 of 2)

RUN NUMBER	THRUST LB.	CALORIA TO MIXER				STEAM TO MIXER				DOWNSTREAM MIXER UPSTREAM NOZZLE				r	V _{EXPT} FT/SEC	V _{THEOR} FT/SEC	η
		ΔP IN. H ₂ O	FLOW LB/SEC	TEMP. F	PRESS. PSIG	ΔP IN. H ₂ O	FLOW LB/SEC	PRESS. PSIG *	TEMP F *	PRESS. PSIG	TEMP. F	SP. VOL. -FT ³ /LB	h BTU/ LB.				
2-2-79																	
102	8.7	7.0	.576	438	77	22	.039	38	293	35	392	9.961	14.76	456	572	63	
103	10.8	6.5	.55	436	72	42	.056	42	284	40	409	9.233	9.83	574	758	57	
104	12.2	6.5	.55	435	75	60	.070	47	290	44	405	8.552	7.86	634	855	55	
2-5-79																	
106	10.0	15.0	.823	471	130	28	.045	41	296	40	461	9.823	18.3	371	590	50	
107	15.0	12.0	.736	471	112	57	.072	55	298	53	440	7.721	10.22	598	816	54	
#2	31.0	42.0	1.36	476	240	115	.129	100	335	70	471	6.378	10.56	670	875	59	
2-6-79																	
a	21.0	13.0	.78	445	140	125	.124	80	316	77	428	5.580	6.30	748	1087	47	
b	19.9	12.0	.746	452	135	101	.104	67	314	69	412	6.006	7.17	5754	995	57	
c	17.8	13.0	.78	457	132	78	.091	67	309	63	418	6.531	8.57	658	907	53	
d	23.0	13.0	.783	444	145	132	.128	81	320	73	398	5.619	6.11	4813	1068	58	
e	15.9	12.0	.752	436	145	53	.069	55	317	55	398	7.114	10.91	624	780	64	
2-7-79																	
A	18.0	13.5	.806	419	155	95	.099	64	306	62	411	6.56	8.14	640	920	48	
B	18.8	13.5	.79	447	157	114	.110	66	308	63	401	6.39	7.18	673	968	48	
C	17.0	13.5	.786	463	165	77	.087	60	305	59	393	6.675	9.04	627	859	53	
D	13.8	13.5	.801	435	170	47	.064	52	300	50	398	7.676	12.5	514	717	51	

Note: Nozzle - sharp throat, 0.58 inch diameter

*Downstream of Metering Orifice

Table A-1. Diesel Bottoming - Nozzle Tests 1979 Large Radial Mixer (Sheet 2 of 2)

RUN NUMBER	THRUST LB.	CALORIA TO MIXER				STEAM TO MIXER				DOWNSTREAM MIXER UPSTREAM NOZZLE				r	V _{EXPT} FT/SEC	V _{THEOR} FT/SEC	η	
		ΔP IN. H ₂ O	FLOW LB/SEC	TEMP. F	PRESS. PSIG	ΔP PSI/IN.H ₂ O	FLOW LB/SEC	PRESS. PSIG *	TEMP F *	PRESS. PSIG	TEMP F	SP. VOL. FT ³ /LB	h BTU/ LB.					
2-9-79																		
0001	26.1	19.0	0.94	436	180	6.3	174	.147	81	328	88	424	4.941	1238.9	6.40	773	1108	49
2	29.7	20.0	0.963	452	270	8.5	235	.188	101	336	93	423	4.697	1237.9	5.12	831	1126	46
3	32.1	19.5	0.95	456	295	9.2	255	.201	108	340	103	422	4.277	1236.0	4.73	898	1290	48
4	16.1	19.0				3.9	108											
1001	17.4	12.0	0.772	397	170	4.2	116	.108	60	301	60	411	6.740	1236.2	7.15	637	965	44
1002	28.7	14.0	0.815	444	205	8.5	235	.182	92	330	90	422	4.830	1237.7	4.48	927	1284	52
1003	32.6	17.0	0.893	449	204	9.4	260	.197	99	334	92	422	4.7368	1237.5	4.53	963	1284	56
1004	19.0	19.0	0.945	448	160	3.9	108	.1062	65	312	62	423	6.660	1242.0	8.895	582	892	43
1005	15.0	21.0	0.985	445	205	2.4	66	.0773	55	308	54	398	7.220	1230.5	12.74	455	724	39
2-13-79																		
1	14.9	10.0	0.682	450	125	4.3	119	.107	56	297	52	420	7.651	1241.8	6.38	608	984	38
2	19.0	7.5	0.58	453	170	7.5	208	.156	71	308	66	394	6.09	1226.3	3.73	831	1268	43
3	25.0	16.0	0.86	458	200	5.5	152	.136	78	319	74	438	4.78	1248.1	6.32	808	1077	56
4	25.0	17.0	0.89	456	205	5.5	152	.138	80	319	74	438	4.78	1248.1	6.45	783	1068	54
5	21.0	16.0	0.86	451	190	4.6	127	.121	73	313	72	433	4.948	1245.7	7.11	689	1022	45
6	33.0	17.0	0.89	458	210	8.2	227	.192	110	333	93	421	4.685	1236.8	4.64	982	1274	60
7	24.0	19.0	0.94	462	210	5.0	138	.127	75	317	71	446	6.115	1252.5	7.40	724	1009	51
8	16.2	18.0	0.91	452	205	3.0	83	.0857	53	311	50	450	8.182	1256.9	10.62	524	796	43
9	11.0	17.0	0.89	462	210	1.7	47	.058	40	303	39	452	9.905	1259.1	15.34	374	632	35

Note: Large Radial type mixer, 0.58 A1 sharp-throat-nozzle-steam
throttled at boiler valve (start 2-9-79)
2-9-(1001). 8" spacer added between mixer and nozzle

*Downstream of Metering Orifice

Table A-2. Diesel Bottoming - Nozzle Tests 1979 Penberthy Mixer

RUN NUMBER	THRUST LB.	CALORIA TO MIXER				STEAM TO MIXER				DOWNSTREAM MIXER UPSTREAM NOZZLE				r	V _{EXPT} FT/SEC	V _{THEOR} FT/SEC	η	
		ΔP IN. H ₂ O	FLOW LB/SEC	TEMP. F	PRESS. PSIG	ΔP PSI/IN.H ₂ O	FLOW LB/SEC	PRESS. PSIG *	TEMP F *	PRESS. PSIG	TEMP F	SP. VOL. FT ³ /LB	h BTU/ LB.					
2-14-79																		
1	22.5	34	1.233	480	82	2.9	80	0.100	84	322	79	454	5.645	1255.5	12.33	543	831	43
2	28.3	19	.927	477	99	5.3	147	0.150	106	336	93	438	4.788	1245.7	6.18	846	1147	54
3	31.8	14	.803	473	100	7.9	219	0.192	117	339	99	433	4.498	1242.2	4.18	1029	1353	58
4	11.8	24	1.06	444	65	2.7	75	0.092	71	300	60	437	6.960	1249.3	11.54	330	796	17
5	1.0	9	.66	436	11	2.2	61	0.052	17	267	11	402	13.56	1243.6	12.7	45	445	
1001	17.9	17	.87	469	64	3.0	83	0.0932	66	308	61	426	6.775		9.36	598	870	47
1002	30.0	20	.947	476	99	5.0	138	0.147	110	339	90	444	4.968		6.44	883	1124	62
1003	27.9	16	.85	472	98	6.0	166	0.162	108	332	91	433	4.851		5.25	888	1217	53
1004	17.0	18	.91	468	76	3.9	108	0.115	80	313	67	434	6.328		7.92	534	961	31

Note: Penberthy mixer with Caloria in the inner annulus 0.58 sharp-edged throat aluminum nozzle

No trouble in heating sphere - end pressure 80-90 psig. It was later determined that the

sphere was pressurized with argon

Runs 1-5 had varying Caloria flow and should not be considered

*Downstream of Metering Orifice

Table A-3. Diesel Bottoming - Nozzle Test 1979 Need e-Jet Mixer (Sheet 1 of 2)

RUN NUMBER	THRUST LB.	CALGRIA TO MIXER				STEAM TO MIXER				DOWNSTREAM MIXER UPSTREAM NOZZLE				r	V _{EXPT} FT/SEC	V _{THEOR} FT/SEC	η	
		LP	FLOW	TEMP.	PFESS.	LP	FLOW	PRESS.	TEMP	PRESS.	TEMP	SP.	h					
		IN. H ₂ O	LE/SEC	F	PSIG	PSI/IN.H ₂ O	LB/SEC	PSIG *	F *	PSIG	F	VOL.	BTU/					
		T ₂	P ₂	P ₁	T ₁	P ₅	T ₅	FT ³ /LB	LB.									
2-16-79																		
0001	11.9	6.5	.568	383	248	4.6	127.0	.129	88	325	50	361	7.312	1212.5	4.40	550	1091	25
0002	8.9	6.5	.568	389	248	3.8	105.0	.109	72	315	41	374	8.670	1220.5	5.21	423	979	19
0003	4.4	7.0	.589	391	268	2.4	66.4	.073	47	306	26	382	12.038	1226.8	8.07	214	719	9
0004	12.9	4.8	.490	388	202	6.2	172.0	.081	11	339	57	351	6.487	1205.9	6.06	727	962	55
2-16-79																		
1005	2.7	3.1	.386	446	245	1.1	30.4	.039	21	264	11	436	20.357	1255.5	9.90	205	508	16
1006	5.7	3.6	.412	456	248	1.9	52.6	.061	37	278	21	439	14.709	1254.8	6.75	388	752	27
1007	8.3	4.1	.44	459	253	2.7	74.7	.080	50	295	29	432	11.909	1249.6	5.50	514	901	33
1008	3.4	4.1	.44	458	240	1.1	30.4	.039	21	278	12	449	19.891	1260.4	11.3	229	497	21
1009	2.9	4.2	.445	456	242	0.7	19.4	.029	16	259	9	447	22.35	1260.7	15.3	197	389	26
1010	9.2	3.9	.429	451	213	3.3	91.3	.094	60	303	32	411	10.862	1239.6	4.56	566	988	33
1011	3.7	2.9	.373	441	155	1.2	33.2	.041	21	268	8	429	22.85	1252.0	9.11	288	465	38

Note: *Needle-jet mixer, steam in tubes, Caloria outside. 0.58 aluminum nozzle.
 0001-4 pressurized by N₂, reached 150 psig at 325F, blew system down, again
 reached 170 psig at 400F, so run was made at the lower tempure.
 1005-1011 pressurized with Ar no problem in heating or pumping. Pressure in
 tank 90 psig at 500F. Difference between pressurizing with N₂ and Ar is
 pronounced.
 Caloria flow was controllable on first series, not on second. Caloria flow
 appeared choked on second series (at higher temperature).

*Downstream of Metering Orifice

Table A-3. Diesel Bottoming - Nozzle Test 1979 Needle-Jet Mixer (Sheet 2 of 2)

RUN NUMBER	THRUST LB.	CALORIA TO MIXER				STEAM TO MIXER						DOWNSTREAM MIXER UPSTREAM NOZZLE					r	V _{EXPT} FT/SEC	V _{THEOR} FT/SEC	η	BTU SEC	
		LP IN. H ₂ O	FLOW LB/SEC	TEMP. F	PRESS. PSIG	LP PSI/IN. H ₂ O	FLCW LB/SEC	PRESS. PSIG*	TEMP. F*	CUAL	PRESS. PSIG	TEMP. F	SP. VOL. FT ³ /LB	h BTU/ LB.	$\frac{\dot{m}v}{pV \Delta T_c}$							
2-22-79																						
0001	22.7	17.0	0.89	457	134	3.4	94.0	.118	105	338	-3	58	430	7.095	1246.0	.0221	7.54	725	947	59	10.6	
0002	23.5	18.5	0.93	458	143	3.7	120.4	.128	115	340	-7	62	438	6.784	1249.5	.0290	7.27	715	982	53	10.8	
0003	23.5	18.5	0.93	454	130	3.5	97.0	.122	109	341	-3	59	430	6.997	1245.9	.0286	7.62	719	946	58	10.9	
0004	20.9	13.0	0.78	450	115	3.7	97.0	.120	104	336	-5	53	425	7.582	1244.2	.0306	6.50	748	984	58	10.1	
0005	20.2	9.1	0.65	450	91	4.1	113.5	.131	107	337	-5	50	425	7.940	1244.6	.0350	4.96	833	1084	59	10.8	
A-6	Needle jet mixer, cuspr drilled out. Both Caloria and Steam flow rates per recalibration Caloria filter (200 mesh 5.5) completely plugged and ruptured. Some large, hard particler (.1 inch.) and much fine. Replaced with 100 mesh copper screen																					
	2-23-79																					
	0001	15.2	9.5	0.67	475	93	2.8	77.5	.097	82	323	-3	41	444	9.456	1254.0	.0305	6.87	637	910	49	
	0002	15.1	9.0	0.65	476	91	3.2	88.6	.107	87	325	-4	43	445	9.135	1255.3	.0325	6.07	642	972	44	
	0003	23.8	17.0	0.89	477	130	4.3	119.0	.143	125	345	-8	66	445	6.495	1252.5	.021	6.21	742	1068	48	
	0004	21.2	13.5	0.80	474	120	3.9	108.0	.131	114	341	-6	58	444	7.215	1253.0	.0214	6.09	733	1045	49	
0005	13.6	11.0	0.71	470	99	1.9	52.6	.077	73	329	-11	35	458	10.784	1262.4	.0274	9.23	556	773	52		
0006	12.5	11.0	0.71	466	89	1.9	52.6	.073	64	307	-4	31	449	11.613	1259.0	.0281	9.72	514	726	50		

*Downstream of Metering Orifice

APPENDIX B

DATA FROM TURBINE TEST RUNS

P_7 = Pressure, Nozzle Inlet, PSIG
 T_3 = Steam Temperature downstream of Metering Orifice, F
 P_{21} = Steam Pressure, Upstream of Metering Orifice, PSIG
 T_3 = Caloria Temperature into Mixer/Nozzle, F
 r = Mass Ratio, Caloria Flow/Steam Flow
 u/c = Speed ratio, turbine RPM/Separator RPM
HP = Measured Turbine power output

APPENDIX C

PROPERTIES OF HEAT TRANSFER FLUID EXXON CALORIA HT43

NOTE: Exxon states: "The values shown here are representative of current production. Some are controlled by manufacturing specifications, while others are not. All of them may vary within modest ranges".

CALORIA HT43 (EXXON) HEAT TRANSFER FLUID

Property	Temperature, F				
	77	200	400	500	600
Density, lb/ft ³	53.5	50.5	46.0	43.9	41.6
lb/gal.	7.15	6.75	6.15	5.87	5.56
Specific Heat, Btu/lb-F	0.50	0.55	0.65	0.70	0.75
Thermal Conductivity, Btu/ft-hr-F	0.0776	0.0743	0.0699	0.068	0.0661
Viscosity, lbm/hr-ft	100.0	12.0	2.6	1.7	1.3

Composition: petroleum fraction (similar to light lube oil)

Maximum Recommended Bulk Temperature: 600 F

Maximum Recommended Film Temperature: 680 F

Pour Point: 15 F

Flash Point (COC): 420 F

AIT: 759 F

Vapor Pressure (600 F): 1.1 psia

Molecular Weight: 170-550

Figure C-1. Properties of Caloria HT43