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Advanced Power Systems
Aerojet-General Corporation

EXPLORATORY RESEARCH ON JET-CONDENSER PERFORMANCE WITH
ORGANIC RANKINE-CYCLE POWER SYSTEMS

Prepared Under
Contract AT(04-3)-728
for the
San Francisco Operations Office
U. S. Atomic Energy Commission

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MASTER

Mid-Term Report

EXPLORATORY RESEARCH ON JET-CONDENSER PERFORMANCE WITH
ORGANIC RANKINE-CYCLE POWER SYSTEMS

19 December 1967

by

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CONTRACT FULFILLMENT STATEMENT

This Midterm Report is submitted in partial fulfillment of U. S. Atomic Energy Commission Contract AT(04-3)-728, and describes the work completed during Tasks I, II, and III as set forth in the Contract Work Statement.

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APPENDIX - WORK STATEMENT

EXPLORATORY RESEARCH ON JET-CONDENSER PERFORMANCE
WITH ORGANIC RANKINE-CYCLE POWER SYSTEMS*

1. INTRODUCTION

In the past the Aerojet-General Corporation, Von Karman Center, Power Systems Division, has been concerned with the design and development of an organic, Rankine-cycle power conversion system, in the 1-150 kwe range, for use in space and hydrospace applications. Figure 1 is a schematic diagram of this system for 12 kwe output.

A major component of the system is the jet-condenser which allows expansion of the working fluid through the turbine to extremely low back-pressures, heretofore unobtainable in Rankine-cycle systems. Such operation results in increased system efficiency. The jet-condenser condenses the saturated vapor with subcooled liquid, which insures single-phase flow through the heat rejection equipment (radiator). Furthermore, the jet-condenser is designed to allow considerable fluid pressure recovery because of momentum interchange between the high-velocity gas stream and the subcooled liquid stream, and because of conversion of the dynamic head of the condensate to static pressure in a suitable diffuser.

The contract work involves exploratory development to determine the design and performance of a jet-condenser applied to 1-15 kwe organic Rankine-cycle power conversion systems for turbine exhaust pressures ranging from 0.5 psia downward toward an ultimate goal of 0.1 psia. Initially, several jet-condenser designs will be prepared covering a range of turbine back-pressures, various organic working fluids, various flow rates, and various degrees of subcooling. A previously developed jet-condenser evaluation computer program will be employed in this analysis. All required jet-condenser designs will then be evaluated and the most promising units chosen for detailed mechanical design, fabrication, and testing. Both hydraulic testing (using water as the working fluid) and hot gas testing (using the eutectic mixture of biphenyl and phenyl ether as the working fluid) will be performed in the Aerojet-General full scale (12 kwe) Organic Rankine Cycle (ORACLE) Development Facility (Figure 2).

The objectives of this research program are to develop both the condensing and pressure recovery potential of the jet condenser into a range of operating conditions compatible with a zero-gravity space environment beyond that achievable with conventional system components. In addition to zero-gravity operating capability which the jet-condenser can provide for organic Rankine-cycle power conversion systems, the unit also provides a means by which the conversion efficiency of this cycle can be increased. This increase results from the ability to operate at extremely low turbine back pressures.

*The work described in this report was supported by U. S. Atomic Energy Commission.

Confirmation of the jet condenser as a component that will permit an organic Rankine-cycle system to operate in space, uninfluenced by zero-g, capable of tolerating station keeping thrust, and with a high overall conversion efficiency, will be a significant advance in available technology for satisfying multikilowatt space power requirements.

2. MATHEMATICAL DESIGN ANALYSIS - TASK I

2.1 SUMMARY OF CASES CONSIDERED

The work statement for Task I is included in the Appendix. A previously developed jet-condenser computer program was used to evaluate the physical dimensions and theoretical performance to be anticipated for jet condensers operating under a variety of conditions. Several fluids, flow rates (power levels), inlet pressures, and degrees of subcooling were considered in the analysis.

2.1.1 Working Fluids

Three organic working fluids were considered: the eutectic mixture of biphenyl and phenyl ether (Dowtherm A), biphenyl, and mono-isopropyl biphenyl (MIPB). All of these fluids exhibit the rather anomalous T-S diagram characteristic shown in Figure 3. Thus, any expansion of the fluid through a turbine is accompanied with the drying out process when the turbine inlet condition corresponds to saturated vapor.

2.1.2 Vapor Flow Rates

The vapor flow rates of interest were chosen to be:

Dowtherm A: 0.0273 lb/sec
 0.136 lb/sec
 0.327 lb/sec
 0.410 lb/sec

Biphenyl: 0.327 lb/sec

MIPB: 0.327 lb/sec

The corresponding electrical output of the power conversion system as a function of vapor flow rate is given as follows for the eutectic mixture of biphenyl and phenyl ether:

<u>Flow Rate (lb/sec)</u>	<u>Electrical Output (kwe)</u>
0.0273	1
0.136	5
0.327	12
0.410	15

2.1.3 Inlet Jet-Condenser Pressures

The inlet pressure to the jet condenser was chosen as follows:

Dowtherm A: 0.1, 0.3, 0.5 psia

Biphenyl: 0.5 psia

MIPB: 0.1, 0.5 psia

In all cases the analysis provides for entry of saturated vapor (100% quality) into the jet condenser.

2.1.4 Degree of Liquid Subcooling

The degree of subcooling of the liquid introduced into the jet condenser for purposes of condensation was varied as follows:

Dowtherm A:	25, 50, 75°F
Biphenyl:	50°F
MIPB:	50°F

In all cases the liquid is subcooled the required amount below the saturation temperature corresponding to the jet-condenser inlet vapor pressure.

2.1.5 Summary Charts

The 39 cases analyzed utilizing the jet condenser computer program in Task I are summarized in Table 1. Table 2 summarizes the vapor and subcooled liquid jet-condenser inlet conditions for the three organic fluids considered.

2.2 DETERMINATION OF MINIMUM α (MASS FLOW RATE-LIQUID/MASS FLOW RATE-VAPOR) FOR CONDENSATION

2.2.1 General

Before the 39 cases discussed above can be analyzed using the Aerojet-General jet-condenser computer program, it will be necessary to determine the ratio of mass flow rate of the liquid to mass flow rate of the vapor (α). The mass flow rate of the vapor is fixed by the Work Statement for Task I and corresponds to a specific electrical power output of the power conversion device of which the jet condenser is a single component. However, considerable latitude is available in the selection of α . Obviously, a minimum α is required in each case for complete condensation of the gas. However, other considerations will dictate a selection of α somewhat greater than the minimum necessary for complete condensation of the gas.

The vapor is expanded through a nozzle from its original state to the state existing in the mixing chamber. This latter state is determined primarily by the vapor pressure (saturation temperature) of the condensate when both the vapor and the coolant are two different phases of the same fluid. In the absence of noncondensibles, the vapor pressure of the condensate can be chosen sufficiently low (low temperature) so that the entry-to-exit pressure ratio of the vapor becomes supersonic.

The supersonic (choked) flow offers several advantages. First, the entry state is not affected by the possible pressure fluctuations in the mixing chamber. This is especially important if it is desirable that

the entry state of the vapor be kept constant as in the case where the entry state corresponds to the exit state from the turbine. Another advantage of the supersonic entry of the vapor into the mixing chamber results from the shock (normal) transition from the supersonic to subsonic vapor flow, which certainly must occur upstream of the vapor-condensate interface. Provided that the vapor is already in the saturated state and is in thermal equilibrium with the coolant, this transition is accompanied by a discontinuous pressure rise and results in "shock" condensation of the vapor instantaneously. Furthermore, the high velocity vapor will transfer momentum to the subcooled liquid, thereby increasing the theoretical isentropic pressure recovery possible with the jet condenser and a suitable diffuser. Thus, we will determine the minimum α required for complete condensation of the saturated vapor.

Consider the schematic jet condenser shown below:

m_v = mass flow rate of the saturated vapor, lb/sec

H_g = enthalpy of the vapor, Btu/lb

m_L = mass flow rate of the subcooled liquid, lb/sec

H_L = enthalpy of the liquid, Btu/lb

H_{mix} = enthalpy of the condensate, Btu/lb

$\alpha = m_L/m_v$

Applying conservation of thermal energy to the system, we have:

$$m_v H_g + m_L H_L = (m_L + m_v) H_{mix} \quad (1)$$

$$m_v H_g + m_L H_L = m_L H_{mix} + m_v H_{mix} \quad (2)$$

$$m_L (H_L - H_{mix}) = m_v (H_{mix} - H_g) \quad (3)$$

$$\alpha = m_L/m_v = (H_{mix} - H_g)/(H_L - H_{mix}) \quad (4)$$

or:

$$\alpha = (H_g - H_{mix})/(H_{mix} - H_L) \quad (5)$$

It is clear that the minimum α required to condense all the vapor occurs when T_{mix} (temperature of the condensate) = T_g (temperature of the vapor). This value of α would be sufficient to raise the

temperature of the subcooled liquid to the temperature of the incoming vapor by removing all the latent heat of vaporization from the vapor.

Consider an energy balance similar to that formulated above, but in this case we will generate expressions in terms of the specific heat, latent heat of vaporization, and temperature, rather than enthalpy. Thus, for the case above, we have:

$$L m_v + m_v C_p (T_{\text{sat}} - T_{\text{mix}}) = m_L C_p (T_{\text{mix}} - T_L) \quad (6)$$

Where

- L = latent heat of vaporization of the fluid, Btu/lb
- m_v = mass flow rate of the vapor, lb/sec
- C_p = specific heat of the liquid, Btu/lb °F
- T_{sat} = temperature of the incoming vapor, °F
- T_{mix} = temperature of the condensate, °F
- m_L = mass flow rate of the liquid, lb/sec
- T_L = temperature of the subcooled liquid, °F

Then

$$\left[L + C_p (T_{\text{sat}} - T_{\text{mix}}) \right] m_v = m_L C_p (T_{\text{mix}} - T_L) \quad (7)$$

$$\alpha = m_L / m_v = \frac{L + C_p (T_{\text{sat}} - T_{\text{mix}})}{C_p (T_{\text{mix}} - T_L)} \quad (8)$$

α minimum occurs when $T_{\text{mix}} = T_{\text{sat}}$:

$$\alpha_{\text{min.}} = L / \left[C_p (T_{\text{sat}} - T_L) \right] \quad (9)$$

Note also that when $T_{\text{mix}} = T_L$, the value of α goes to infinity, as expected. Let us now consider actual values for the organic fluids of interest to this program.

2.2.2 Dowtherm A (Eutectic mixture of biphenyl and phenyl ether)

Consider the case shown below schematically. The vapor is introduced at 223°F corresponding to 0.1 psia. The liquid is subcooled 25, 50, and 75°F. In each case the condensate temperature corresponds to the inlet vapor temperature. $\alpha_{\text{min.}}$ is computed easily for each of these 3 cases using Eq. 9, above. A sample calculation follows:

$$I(223^{\circ}\text{F}) = 151.9 \text{ Btu/lb}$$

$$C_p = 0.43 \text{ Btu/lb } ^{\circ}\text{F}$$

$$T_{\text{sat}} = 223^{\circ}\text{F}$$

$$T_L = 198, 173, 148^{\circ}\text{F}$$

Fluid: Dowtherm A at 0.1 psia

$$\alpha_{\text{min.}} = (151.9)/(0.43) (223 - T_L)$$

The summary of the calculations is given in the table below

$\alpha_{\text{min.}}$	T_L	ΔT
14.1	198°F	25°F
7.0	173	50
4.7	148	75

When the vapor enters the jet condenser at 0.3 psia and 267°F, the corresponding results for Dowtherm A are as follows:

$\alpha_{\text{min.}}$	T_L	ΔT
13.8	242°F	25°F
6.9	217	50
4.6	192	75

When the vapor enters the jet condenser at 0.5 psia and 289°F, the corresponding results for Dowtherm A are as follows:

$\alpha_{\min.}$	T_L	ΔT
13.3	264°F	24°F
6.6	239	50
4.4	214	75

Figure 4 is a plot of $\alpha_{\min.}$ required for complete condensation versus ΔT , the degree of liquid subcooling, for the 3 values of jet-condenser inlet pressure for the case of Dowtherm A as the working fluid. Figure 5 is a plot of $\alpha_{\min.}$ versus P_3 , the jet-condenser inlet pressure for the 3 values of liquid subcooling considered. For a given degree of liquid subcooling, more liquid (higher α) is required for complete condensation as the inlet pressure is decreased. This effect, however, is quite weak (Figure 5), and is due to the variation of L and C_p with temperature, and the variation of T_{sat} with pressure. Also, for constant inlet pressure, the amount of liquid required for complete condensation decreases as the degree of subcooling increases, as expected. This effect is pronounced (Figure 4).

If we were to choose $\alpha = \alpha_{\min.}$ for our analyses, this would result in a very poor design, since the Mach number of the vapor in such a case would be nearly zero. There would be no momentum interchange between the vapor and subcooled liquid. As indicated previously, we would like the Mach number to be ≥ 1 . The value of α corresponding to Mach number = 1 is determined in Section 2.3 of this report. We will first return to the determination of $\alpha_{\min.}$ required for complete condensation in the cases where the working fluid is biphenyl and mono-isopropyl biphenyl.

2.2.3 Biphenyl

Eq. 9 is still applicable. For biphenyl the vapor is introduced at 284°F, corresponding to 0.5 psia. The liquid is subcooled 50°F.

$$L(284^\circ\text{F}) = 156.6 \text{ Btu/lb}$$

$$C_p = 0.63 \text{ Btu/lb } ^\circ\text{F}$$

$$T_{\text{sat}} = 284^\circ\text{F}$$

$$T_L = 234^\circ\text{F}$$

Fluid: Biphenyl at 0.5 psia

$$\alpha_{\min.} = (156.6 / (0.63)) (284 - 234) = 5.0$$

2.2.4 Monoisopropyl Biphenyl (MIPB)

Eq. 9 is once again used to compute $\alpha_{\min.}$ required for complete condensation. For MIPB, the vapor is introduced at 0.1 and 0.5 psia. The corresponding saturation temperatures are 265°F and 340°F, respectively. The liquid is subcooled 50°F in both cases.

$$L(265^{\circ}\text{F}) = 137.3 \text{ Btu/lb}$$

$$C_p = 0.40 \text{ Btu/lb } ^{\circ}\text{F}$$

$$T_{\text{sat}} = 265^{\circ}\text{F}$$

$$T_L = 215^{\circ}\text{F}$$

Fluid: Monoisopropyl biphenyl at 0.1 psia

$$\alpha_{\min.} = (137.3)/(0.40) (265 - 215) = 6.9$$

$$L(340^{\circ}\text{F}) = 132.1 \text{ Btu/lb}$$

$$C_p = 0.43 \text{ Btu/lb } ^{\circ}\text{F}$$

$$T_{\text{sat}} = 340^{\circ}\text{F}$$

$$T_L = 290^{\circ}\text{F}$$

Fluid : Monoisopropyl biphenyl at 0.5 psia

$$\alpha_{\min.} = (132.1)/(0.43) (340 - 290) = 6.2$$

For a given degree of liquid subcooling of the MIPB (50°F), it is seen that more liquid (higher α) is required for complete condensation as the jet condenser inlet pressure is decreased. However, as in the case of Dowtherm A, this effect is rather weak.

2.2.5 Conclusions

In the table below, the α_{minimum} required for complete condensation of the Dowtherm A, biphenyl, and monoisopropyl biphenyl for various jet-condenser inlet pressures and one degree of liquid subcooling is summarized.

α_{minimum} Required for Complete Condensation of Dowtherm A, Biphenyl, and Monoisopropyl Biphenyl

<u>Fluid</u>	<u>P₃</u>	<u>ΔT</u>	<u>α_{minimum}</u>
Dowtherm A	0.1 psia	50°F	7.0
MIPB	0.1	50	6.9
Dowtherm A	0.5	50	6.6
MIPB	0.5	50	6.2
Biphenyl	0.5	50	5.0

At 0.1 psia and 50°F liquid subcooling the MIPB requires slightly less liquid for complete condensation than does the Dowtherm A. At 0.5 psia and 50°F liquid subcooling the biphenyl requires the least liquid for complete condensation and the Dowtherm A the most. As α_{minimum} increases, the pumping power requirements of the power conversion system would increase, and system efficiency would tend to decrease at a constant value of jet-condenser inlet pressure. However, this effect would not be pronounced until α_{minimum} (required for complete condensation) reached a value of approximately 50.

In the case of Dowtherm A and MIPB where more than one jet-condenser inlet pressure was considered, at a given degree of liquid subcooling, more liquid (higher α) is required for complete condensation as the inlet pressure is decreased. This effect is rather weak but would also tend to increase pumping power requirements and decrease overall system efficiency. However, the advantage of operating at the lower jet-condenser inlet pressure (and, hence, lower turbine back-pressure) would more than offset this effect, since more energy per pound of vapor would be extracted from the working fluid during the expansion in the turbine to the lower back-pressure.

2.3 DETERMINATION OF α FOR SUPERSONIC OPERATION

2.3.1 General

It is desirable that the velocity of the vapor entering the jet-condenser correspond to $M \geq 1$. This supersonic (choked) flow offers

several advantages. First, the entry state is not affected by the possible pressure fluctuations in the mixing chamber (if the pressure fluctuations do not cause sonic operation). This is especially important if it is desirable that the entry state of the vapor be kept constant, as in the case where the entry state corresponds to the exit state from the turbine. Secondly, the high-velocity vapor will transfer momentum to the subcooled liquid, thereby increasing the theoretical isentropic pressure recovery possible with the jet-condenser and a suitable diffuser. Let us investigate this effect in more detail here. Consider the schematic jet-condenser and diffuser section shown below:

$$\text{Assume } \alpha = m_L / m_v = 10.0$$

where:

- m_v = mass flow rate of the saturated vapor, lb/sec
- v_v = velocity of the saturated vapor, ft/sec
- m_L = mass flow rate of the subcooled liquid, lb/sec
- v_L = velocity of the subcooled liquid, ft/sec

Applying conservation of momentum to the system, we have:

$$m_v v_v + m_L v_L = (m_v + m_L) V_1 \quad (10)$$

where:

V_1 is the velocity of the condensate at station (1).

Corresponding to supersonic operation of the jet-condenser, typical values for v_v and v_L are as follows:

$$\begin{aligned} v_v &= 600 \text{ ft/sec} \\ v_L &= 150 \text{ ft/sec} \end{aligned}$$

Then, substituting into Eq. 10, we have:

$$600 m_v = 150 m_L = (m_v + m_L) V_1 \quad (11)$$

$$600 + 150 \alpha = (1 + \alpha) V_1 \quad (12)$$

$$V_1 = (600 + 1500)/11 = 2100/11 = 191 \text{ ft/sec} \quad (13)$$

Now $V_1 = V_{1[IS]}$ (maximum possible velocity - isentropic process). Note that the condensate velocity (V_1) is greater than the initial liquid velocity because of momentum interchange from the high-velocity gas stream to the subcooled liquid.

We now would like to convert this velocity head to static pressure in a suitable diffuser section. For maximum conversion, we desire $V_2 \approx 0$, where V_2 is the velocity of the fluid stream at station (2). From the continuity equation ($w = \rho A v$), it is seen that we can get a hundred-fold decrease in the velocity at station (2) with respect to that at station (1), if we make $A_2/A_1 = 100$,

where:

A_1 = cross-sectional flow area at station (1)

A_2 = cross-sectional flow area at station (2)

Thus, for a circular cross section, we want $D_2/D_1 = 10$,

where:

D_1 = diameter at station (1)

D_2 = diameter at station (2)

This will ensure nearly complete conversion of the velocity head to static pressure. Applying Bernoulli's Principle to stations (1) and (2), we have:

$$P_1 + 1/2 \rho_1 V_1^2 = P_2 + 1/2 \rho_2 V_2^2 \quad (14)$$

where:

P_1 = pressure at station (1)

ρ_1 = density of fluid at station (1)

V_1 = velocity of fluid at station (1)

P_2 = pressure at station (2)

ρ_2 = density of fluid at station (2)

V_2 = velocity of fluid at station (2)

Now:

$$P_1 \approx 0 \text{ and } V_2 \ll V_1$$

Thus:

$$P_2 = 1/2 \rho_1 V_1^2 = (1/2) (63) (191)^2 / (32.2) (144) \quad (15)$$

$$P_2 = 248 \text{ psia } P_2(\text{IS}) \quad (16)$$

This is the maximum possible isentropic pressure recovery for this case. Note that $P_2 >$ inlet liquid pressure (151 psia). In such a case the jet condenser acts as a jet pump.

Now consider the case where the incoming vapor has very low velocity ($M \approx 0$). Here $v_v \rightarrow 0$, and Eq. 10 becomes

$$m_L v_L = (m_v + m_L) V_1 \quad (17)$$

Here, there is no momentum interchange from the gas to the liquid. Thus, we have:

$$V_1 = m_L v_L / (m_v + m_L) = \alpha / (1 + \alpha) v_L \quad (18)$$

$$V_1 = (10/11) (150) = 1500/11 = 137 \text{ ft/sec} \quad (19)$$

Again, we would like to convert this velocity head to static pressure in a suitable diffuser. Proceeding as before:

$$P_2 = 1/2 \rho_1 V_1^2 = (1/2) (63) (137)^2 / (32.2) (144) \quad (20)$$

$$P_2 = 128 \text{ psia} \quad (21)$$

Note that the liquid inlet pressure was 151 psia.

The overall pressure recovery in either of the above cases is defined by the equation:

$$P_2 = P_{\text{mix}} + \psi P_{\text{IS}} \quad (22)$$

where:

P_2 = actual pressure at station (2)

P_{mix} = pressure in the mixing chamber of the jet-condenser

P_{IS} = maximum (isentropic) pressure recovery

Ψ = pressure recovery factor

Now:

$$P_{mix} \approx 0$$

Thus:

$$\Psi = P_2/P_{IS} \quad (23)$$

For $P_2 = 248$ psia = P_{IS} , we have:

$$\Psi = 1.0 \text{ (complete momentum interchange and recovery)} \quad (24)$$

For $P_2 = 128$ psia, we have:

$$\Psi = 128/248 = 0.52 \text{ (no momentum interchange)} \quad (25)$$

It is clear that the momentum interchange between gas and liquid is an important factor in determining the pressure recovery. We also speak of "hydraulic" pressure recovery; i.e., recovery with no vapor present:

$$\Psi_{hydraulic} = P_2/P_{L_{entering}} \quad (26)$$

where:

P_2 = actual pressure at station (2)

$P_{L_{entering}}$ = pressure of liquid introduced into the jet-condenser after passing through the orifice nozzle assembly

For $P_2 = 128$ psia, we have:

$$\Psi_{hydraulic} = 128/151 = 0.85 \quad (27)$$

However, this is an ideal case. To date, values of $\Psi_{hydraulic}$ in the range 0.52-0.53 have been obtained experimentally.

Eq. 24 corresponds to supersonic gas flow while Eq. 25 corresponds to the case where the gas enters the jet-condenser at very low velocity and imparts negligible momentum to the condensate stream. Thus, the importance of supersonic operation of the jet-condenser is clearly indicated.

We will determine the necessary α required so that the inlet gas velocity corresponds to $M = 1.0$. Consider the schematic jet-condenser shown below:

For the general case, Eq. 5 applies:

$$\alpha = (H_g - H_{mix}) / (H_{mix} - H_L) \quad (5)$$

To ensure that the entering gas has velocity corresponding to $M = 1.0$, the pressure ratio, P_3/P_{mix} , must equal the critical pressure ratio. This ratio is a function of the working fluid. Since the pressure in the mixing chamber, P_{mix} , is chosen to be the saturation pressure corresponding to T_{mix} , it is a matter of supplying sufficient cold fluid so that T_{mix} (and hence P_{mix}) are reduced the proper amount. Since P_3 and the critical pressure ratio are known in any given case, it is possible to determine P_{mix} , T_{mix} , and H_{mix} . H_g and H_L are also known quantities. Consequently α corresponding to $M = 1.0$ can be easily computed from Eq. 5.

We will now consider actual values for the organic fluids and operating conditions of interest to this program.

2.3.2 Dowtherm A

Consider the case shown below schematically. The vapor is introduced at 223°F corresponding to 0.1 psia (100% quality). The liquid is subcooled 25, 50, and 75°F. In each case the condensate temperature, 210°F, is chosen such that the ratio of jet-condenser inlet pressure to the mixing chamber pressure P_3/P_{mix} , is equal to the critical pressure ratio, 1.67. This critical pressure ratio is easily computed for any fluid from the general expression for an isentropic expansion:

$$P_3/P_{mix} = \left(1 + \frac{\gamma-1}{2} M^2\right)^{\gamma/(\gamma-1)}$$

where:

γ = ratio of specific heats

M = Mach number

The critical pressure ratio occurs when $M = 1.0$:

$$P_3/P_{mix} = \left(1 + \frac{\gamma-1}{2}\right)^{\gamma/(\gamma-1)} = \left(\frac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)}$$

since

$\gamma = 1.03$ for Dowtherm A, we have:

$$\frac{P_3}{P_{mix}} = \left(\frac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)} = \left(\frac{2.03}{2}\right)^{\frac{1.03}{0.03}} = (1.015)^{34.4}$$

Thus:

$$P_3/P_{\text{mix}} = 1.67$$

The saturation pressure corresponding to the condensate temperature of 210°F is equal to 0.06 psia. Thus, the critical pressure ratio is $0.10/0.06 = 1.67$, as required. This will ensure introduction of the vapor at velocity corresponding to $M = 1.0$, as long as the gas nozzle is properly shaped. Since the inlet gas and liquid conditions and the outlet condensate conditions are now determined, the necessary enthalpies: H_g , H_L , and H_{mix} are easily arrived at. Then α corresponding to $M = 1.0$ is easily computed from Eq. 5 above for each of the 3 cases of interest. A sample calculation follows:

$$H_L (198^\circ\text{F}) = 57.5 \text{ Btu/lb}$$

$$H_L (173^\circ\text{F}) = 47.0 \text{ Btu/lb}$$

$$H_L (148^\circ\text{F}) = 36.8 \text{ Btu/lb}$$

$$H_g (223^\circ\text{F}) = 220.2 \text{ Btu/lb}$$

$$H_{\text{mix}} (210^\circ\text{F}) = 62.6 \text{ Btu/lb}$$

Fluid: Dowtherm A at 0.1 psia

Equation 5 becomes:

$$\alpha_{M=1.0} = (220.2 - 62.6)/(62.6 - H_L) = 157.6/(62.6 - H_L) \quad (28)$$

The summary of the calculations for the 3 cases at 0.1 psia jet-condenser inlet pressure is given in the following table.

α required for $M = 1.0$, Dowtherm A at 0.1 psia

$\alpha_M = 1.0$	T_L	ΔT
31.0	198°F	25°F
10.1	173	50
6.1	148	75

Consider the case shown below schematically. The vapor is introduced at 267°F corresponding to 0.3 psia (100% quality). The liquid is subcooled 25, 50, and 75°F, as before. The condensate temperature is chosen to be 246°F, which corresponds to a P_{mix} of 0.18 psia. This ensures the correct critical pressure ratio of 1.67 ($0.30/0.18 = 1.67$). Eq. 5 is once again employed, and the results at a jet-condenser inlet pressure of 0.3 psia are summarized in the following table.

$$\begin{aligned}
H_L (242^\circ\text{F}) &= 76.5 \text{ Btu/lb} \\
H_L (217^\circ\text{F}) &= 65.6 \text{ Btu/lb} \\
H_L (192^\circ\text{F}) &= 54.9 \text{ Btu/lb} \\
H_g (267^\circ\text{F}) &= 235.9 \text{ Btu/lb} \\
H_{\text{mix}} (246^\circ\text{F}) &= 78.2 \text{ Btu/lb}
\end{aligned}$$

$$\alpha_M = 1.0 = (235.9 - 78.2)/(78.2 - H_L) = 157.7/(78.2 - H_L) \quad (29)$$

α Required for $M = 1.0$, Dowtherm A at 0.3 psia

$\alpha_M = 1.0$	T_L	ΔT
93.0	242°F	25°F
12.5	217	50
6.8	192	75

Consider the case shown schematically below. The vapor is introduced at 289°F corresponding to 0.5 psia (100% quality). The liquid is subcooled 25, 50, and 75°F, as in previous cases for Dowtherm A. The condensate temperature is chosen to be 267°F, which corresponds to a P_{mix} of 0.30 psia. This ensures the correct critical pressure ratio of 1.67 ($0.50/0.30 = 1.67$).

$$\begin{aligned}
H_L (264^\circ\text{F}) &= 86.2 \text{ Btu/lb} \\
H_L (239^\circ\text{F}) &= 75.2 \text{ Btu/lb} \\
H_L (214^\circ\text{F}) &= 64.3 \text{ Btu/lb} \\
H_g (289^\circ\text{F}) &= 244.1 \text{ Btu/lb} \\
H_{\text{mix}} (267^\circ\text{F}) &= 87.5 \text{ Btu/lb} \\
\text{Fluid: Dowtherm A at 0.5 psia}
\end{aligned}$$

Eq. 5 becomes:

$$\alpha_M = 1.0 = (244.1 - 87.5)/(87.5 - H_L) = 156.6/(87.5 - H_L) \quad (30)$$

The summary of the calculations for the 3 cases at 0.5 psia jet-condenser inlet pressure is given in the following table.

α Required for $M = 1.0$, Dowtherm A at 0.5 psia

$\alpha_{M = 1.0}$	T_L	ΔT
120.0	264°F	25°F
12.7	239	50
6.8	214	75

Figure 6 is a plot of α required for $M = 1.0$ versus ΔT , the degree of liquid subcooling, for the three values of jet-condenser inlet pressure for the case of Dowtherm A as the working fluid. For constant inlet pressure the amount of liquid required to achieve supersonic vapor velocity decreases as the degree of liquid subcooling increases, as expected. Note that each curve has a well-defined vertical asymptote.

From Eq. 5 it is clear that when $H_L = H_{mix}$, α approaches ∞ . This would correspond to the physical case where the liquid inlet temperature to the jet condenser equalled the condensate outlet temperature and could only be achieved with infinite α . At 0.1 psia jet-condenser inlet pressure, T_{mix} was taken to be 210°F so as to achieve the necessary mixing chamber pressure. Therefore, for a ΔT of 13°F (223 - 210) the liquid would be introduced at 210°F. This is also the condensate temperature and is achieved only at infinite α . Consequently, at 0.1 psia the vertical asymptote in Figure 6 corresponds to $\Delta T = 13^\circ\text{F}$. Similarly, the vertical asymptote at 0.3 psia and 0.5 psia occur at values of $\Delta T = 21^\circ\text{F}$ and 22°F , respectively. As the degree of liquid subcooling is decreased at a constant value of jet-condenser inlet pressure, the value of α required for supersonic operation increases sharply. This would result in an increase in system pumping power requirements and decrease in overall system efficiency.

Figure 7 is a plot of α required for $M = 1.0$ versus P_3 , the jet-condenser inlet pressure for the three values of liquid subcooling considered. For a given degree of liquid subcooling, more liquid (higher α) is required for supersonic operation as the inlet pressure is increased. Therefore, the higher inlet pressures would result in greater pumping power requirements and reduced system efficiency.

Operation of the jet-condenser in the supersonic range results in high inlet vapor velocity, momentum interchange between this high-velocity gas stream and the subcooled liquid, and a potentially greater pressure recovery in the diffuser section. We now turn our attention to the cases where the working fluid is biphenyl and monoisopropyl biphenyl.

2.3.3 Biphenyl

Eq. 5 is still applicable. For biphenyl the vapor is introduced at 287°F , corresponding to 0.5 psia. The liquid is sub-cooled 50°F . Refer to the schematic below.

For the biphenyl the critical pressure ratio is 1.67. Thus, we desire that $P_3/P_{\text{mix}} = 1.67$ for $M = 1.0$. Hence, $P_{\text{mix}} = 0.3$ psia, and $T_{\text{mix}} = 260^{\circ}\text{F}$.

$$\begin{aligned} H_L (234^{\circ}\text{F}) &= 31.6 \text{ Btu/lb} \\ H_g (284^{\circ}\text{F}) &= 209.8 \text{ Btu/lb} \\ H_{\text{mix}} (260^{\circ}\text{F}) &= 42.3 \text{ Btu/lb} \\ \text{Fluid: biphenyl at 0.5 psia} \end{aligned}$$

Eq. 5 becomes:

$$\alpha_M = 1.0 = (209.8 - 42.3)/(42.3 - 31.6) - 167.5/10.7 = 15.7$$

2.3.4 Monoisopropyl Biphenyl

Eq. 5 above, is once again used to compute α corresponding to $M = 1.0$. For MIPB the vapor is introduced at 0.1 and 0.5 psia. The corresponding saturation temperatures are 265°F and 340°F . the liquid is subcooled 50°F in both cases. At 0.1 psia the situation is depicted schematically below.

For MIPB the critical pressure ratio is 1.67. Thus, we desire that $P_3/P_{\text{mix}} = 1.67$, for $M = 1.0$. Hence, $P_{\text{mix}} = 0.06$ psia, and $T_{\text{mix}} = 245^{\circ}\text{F}$,

$$\begin{aligned} H_L (215^{\circ}\text{F}) &= -90.2 \text{ Btu/lb} \\ H_g (265^{\circ}\text{F}) &= 68.8 \text{ Btu/lb} \\ H_{\text{mix}} (245^{\circ}\text{F}) &= -78.0 \text{ Btu/lb} \end{aligned}$$

Fluid: monoisopropyl biphenyl at 0.1 psia

$$\alpha_M = 1.0 = \left[68.8 - (-78.0) \right] / \left[-78.0 - (-90.2) \right] = 146.8/12.2 = 12.0$$

At 0.5 psia the corresponding schematic is shown below.

P_{mix} is taken to be 0.3 psia so that $P_3/P_{mix} = 0.5/0.3 = 1.67$, as required. T_{mix} is then 315°F.

$$H_L (290^\circ\text{F}) = 059.0 \text{ Btu/lb}$$

$$H_g (340^\circ\text{F}) = 94.5 \text{ Btu/lb}$$

$$H_{mix} (315^\circ\text{F}) = -48.0 \text{ Btu/lb}$$

Fluid: monoisopropyl biphenyl at 0.5 psia

$$\alpha_M = 1.0 = \left[94.5 - (-48.0) \right] / \left[-48.0 - (-59.0) \right] = 142.5/11.0 = 13.0$$

For a given degree of liquid subcooling of the MIPB (50°F), it is seen that more liquid (higher α) is required for supersonic operation as the jet-condenser inlet pressure is increased. However, as in the case of Dowtherm A, this effect is rather weak.

2.3.5 Conclusions

α Required for $M = 1.0$; Dowtherm A, Biphenyl, and MIPB

Fluid	P_3	ΔT	$\alpha_M = 1.0$
Dowtherm A	0.1 psia	50°F	10.1
MIPB	0.1	50	12.0
Dowtherm A	0.5	50	12.7
MIPB	0.5	50	13.0
Biphenyl	0.5	50	15.7

In this table, α required for $M = 1.0$ for Dowtherm A, biphenyl, and monoisopropyl biphenyl at various jet-condenser inlet pressures and degrees of liquid subcooling is summarized. At 0.1 psia and 50°F liquid subcooling the Dowtherm A requires less liquid for supersonic operation than does the MIPB. At 0.5 psia and 50°F liquid subcooling the Dowtherm A requires the least liquid for supersonic operation and the biphenyl the most. As α for supersonic operation increases, the pumping power requirements of the power conversion system would increase, and system efficiency would tend to decrease at a constant value of jet-condenser inlet pressure. However, this effect would not be pronounced unless α corresponding to $M = 1.0$ varied over a wide range.

In the case of Dowtherm A and MIPB where more than one jet-condenser inlet pressure was considered, at a given degree of liquid subcooling, more liquid (higher α) is required for supersonic operation as the inlet pressure is increased. This effect is rather weak but would also tend to increase pumping power requirements and decrease overall system efficiency. Thus, selection of Dowtherm A as the working fluid (supersonic operation) would maximize system efficiency. Operation at the lower jet-condenser inlet pressure (and, hence, lower turbine back-pressure) would further increase the overall system efficiency.

2.4 RECOMMENDATIONS FOR SELECTION OF α

The Work Statement for Task I specifies the organic working fluids, the vapor flow rates, the jet-condenser inlet pressures, and the degree of liquid subcooling to be considered in the mathematical design analysis. However, the ratio of liquid to vapor flow rate, α , is not specified. The detailed discussion and investigation presented in Sections 2.2 and 2.3, above, will now form the basis for meaningful selections of α corresponding to the several cases to be investigated. It is clear that introduction of the vapor at supersonic velocity is highly desirable for reasons stated previously, i.e., high momentum transfer to the subcooled liquid and choked flow operation. Therefore, values of α corresponding to $M \geq 1.0$ were chosen, where applicable. Only when unusually high values of α (≥ 50) were required to achieve the critical pressure ratio was this criterion abandoned. The following general rules were adopted:

(a) The temperature (mixing pressure) of the condensate was determined from the following considerations:

(1) Given the initial state of the vapor and the degree of subcooling of the liquid, the minimum liquid-to-vapor ratio (α) necessary for complete condensation is established, and the upper limit of the temperature of the condensate is determined.

(2) The ratio of the initial pressure of the vapor to the pressure in the mixing chamber (taken equal to the saturation pressure of the condensate) determines the entry velocity of the vapor. It is desirable that the expansion ratio be high, preferably above the critical.

At 25°F subcooling, the critical pressure ratio can only be achieved at very high α (≈ 100). Such magnitude of α is not practical for a power conversion system. The pumping power becomes excessive, especially if the pressure recovery from the jet-condenser is poor. At subcooling $\geq 50^\circ\text{F}$, the critical pressure ratios are reached at moderate values of α . These results lead to the selection of the temperature of the condensate of Dowtherm A, biphenyl, and monoisopropyl biphenyl in the following manner:

(a) For $\Delta T = 25^\circ\text{F}$, the condensate temperature, T_{mix} , was taken to be equal to the inlet vapor temperature.

(b) For $\Delta T \geq 50^\circ\text{F}$, T_{mix} (P_{mix}) corresponds to the Mach Number $M \geq 1.0$.

(b) The value of liquid-to-vapor flow rate, α , was easily computed from the following expression (conservation of thermal energy):

$$\alpha = (H_g - H_{\text{mix}})/(H_{\text{mix}} - H_L) \quad (31)$$

where:

H_g = enthalpy of the hot gas (saturated vapor) entering the jet-condenser, Btu/lb

H_{mix} = enthalpy of the condensate (liquid) leaving the jet-condenser, Btu/lb

H_L = enthalpy of the subcooled liquid entering the jet-condenser, Btu/lb

The values of α given by Eq. 31 result in jet-condensers of unusually large dimensions. Past experience has shown that these values should be increased so as to reduce physical dimensions and assure complete condensation of the vapor. As a result, the values of α computed from Eq. 31 have been doubled. This overflow situation will result in greater condensation rates and a shorter jet-condenser assembly.

Table 3 summarizes the values of T_{mix} , P_3/P_{mix} , M , V , P_{mix} , and α used in the computer analysis of the 39 cases considered with the three working fluids.

2.5 INVESTIGATION OF OPTIMUM DEGREE OF LIQUID SUBCOOLING - RADIATOR ANALYSIS

In the 39 cases to be analyzed in Task I the degree of liquid subcooling has been specified as 25, 50, or 75°F. The greater the degree of subcooling, the lower the temperature of the condensate exiting from the jet-condenser. The radiator area varies as $1/T^4$, where

T = heat rejection temperature (absolute temperature scale). Thus, for a fixed quantity of heat to be rejected, the radiator area and weight will increase as the rejection temperature decreases. However, the lower rejection temperature is usually associated with a lower jet-condenser inlet pressure, and hence, higher overall system efficiency.

Detailed analysis and computer calculation have shown that the system efficiency of a reference organic Rankine-cycle power conversion system increases significantly as the turbine back-pressure (and jet-condenser inlet pressure) is decreased. Consequently, as system efficiency increases, the amount of heat to be rejected decreases, for constant electrical power output. So it is not all clear what set of operating conditions (jet-condenser inlet pressure, degree of liquid subcooling, and radiator inlet temperature) would result in minimum radiator area and weight. This problem was posed by Mr. G. Leighton, AEC Project Manager, in a meeting at Von Karman Center, on September 27, 1967. We shall investigate this situation in some detail.

Consider the schematic diagram below:

The condensate (liquid) exits from the jet-condenser at temperature T_{mix} , which equals the inlet temperature to the radiator, T_{in_R} .

Heat is rejected to the surroundings by radiation where the ambient temperature is T_a . The liquid leaves the radiator at temperature T_{out_R} and enters the jet condenser at this temperature. The degree of liquid subcooling is $T_{in_{JC}} - T_{out_R}$. The basic radiation heat transfer law for the case above can be stated as follows:

$$Q = \sigma \epsilon A (T_w^4 - T_a^4) \quad (32)$$

where:

- Q = rate of heat transfer by radiation, Btu/hr
- σ = Stefan-Boltzmann constant, 0.174×10^{-8} Btu/hr ft²°R⁴
- ϵ = emissivity of the surface, dimensionless
- A = radiating area, ft²
- T_w = wall temperature, °R
- T_a = ambient temperature, °R

In our case, the working fluid passes through the radiator as a liquid, so T_w is continuously changing, i.e., $T_w = T_w(x)$. See schematic below:

Eq. 32 would apply to a case of radiation heat transfer, where T_w = constant (corresponding to a change of phase). We are interested in an exact expression relating Q and A for our situation. Consider the following schematic:

where:

T_a = ambient temperature, °R

T_{f1} = fluid temperature at inlet to radiator, °R

T_{f2} = fluid temperature at outlet from radiator, °R

T_{w1} = wall temperature at radiator inlet, °R

T_{w2} = wall temperature at radiator outlet, °R

The fluid enters at temperature T_{f1} and exits at temperature T_{f2} .

The appropriate expression relating A and Q in this case is given as follows (Ref. 1):

$$A/Q = 1/(4 \sigma \epsilon \Delta T T_a^3) \ln \left[\frac{T_{w1} - T_a}{T_{w2} - T_a} \cdot \frac{T_{w2} + T_a}{T_{w1} + T_a} \right] \quad (33)$$

Knowing T_{w1} , T_{w2} , T_a , and the temperature drop across the radiator, ΔT , one could compute the ratio, A/Q, the specific radiator area (ft²/kw rejected).

For many applications, $T_a \ll T_{w1}$ and $T_a \ll T_{w2}$, and a more simple expression relating A and Q can be utilized. Consider the schematic below where we assume that $T_a \ll T_{w1}$ and $T_a \ll T_{w2}$. Also assume $T_{w1} \approx T_{f1} = T_{in}$ and $T_{w2} \approx T_{f2} = T_{out}$.

The appropriate expression relating A and Q in this case is given by (Ref. 2 and 3):

$$A/Q = \left[1/T_{out}^3 - 1/T_{in}^3 \right] / 3 \sigma \epsilon \Delta T \quad (34)$$

where A/Q is the specific radiator area (ft²/kw rejected). It is interesting to compare Eq. 34 with the more simple form of the radiation heat-transfer equation where the temperature of the radiating surface is constant:

$$Q = \sigma \epsilon A T^4 \quad (35)$$

T = radiating temperature = constant

Even in our present situation we could take the temperature, T, to be the arithmetic average of the inlet and outlet temperatures:

$$T = T_{avg} = \frac{T_{in} + T_{out}}{2} \quad (36)$$

This is only an approximation, but it is instructive to compare the results in a given situation when Eq. 34 (exact) and Eq. 35 (approximate) are used:

Sample calculation: $T_{in} = 200^\circ F = 660^\circ R$
 $T_{out} = 175^\circ F = 635^\circ R$

From Eq. 36:

$$T_{avg} = \frac{T_{in} + T_{out}}{2} = (200 + 175)/2 = 187.5^\circ F$$

$$\left[T_{avg} \right]^4 = (187.5 + 460)^4 = (647.5)^4 = 1.76 \times 10^{11} R^4$$

for use in Eq. 35:

$$Q = \sigma \epsilon A T^4 \quad (35)$$

Compare with equation:

$$Q = 3 \sigma \epsilon A \Delta T \left[1/T_{out}^3 - 1/T_{in}^3 \right] \quad (37)$$

We see from Eq. 37 and 35 that:

$$T_{avg}^4 = 3 \Delta T \left[1/T_{out}^3 - 1/T_{in}^3 \right] = T_{equiv.}^4 \quad (38)$$

Now:

$$T_{equiv.}^4 = 3 \Delta T \left[1/T_{out}^3 - 1/T_{in}^3 \right] = (3) (25) / \left[1/(635)^3 - 1/(660)^3 \right] = (75 \times 10^8) / (0.0422) = 1.780 \times 10^{11} R^4$$

Then:

$$T_{\text{equiv}} = 650^{\circ}\text{R} = 190^{\circ}\text{F}$$

while $T_{\text{avg}} = 187.5^{\circ}\text{F}$. As a result, in this case, the use of the arithmetic average of the radiator inlet and outlet temperature in Eq. 35 versus use of the more exact formulation represented by Eq. 37 would introduce about 1% error in the final result.

In summary, we will use the more exact expression relating A and Q for the case where the fluid enters the radiator at temperature T_{in} and exits at temperature T_{out} :

$$A/Q = \frac{1/T_{\text{out}}^3 - 1/T_{\text{in}}^3}{3 \sigma \epsilon \Delta T} \quad (39)$$

Note again that Q denotes rate of heat rejection.

Eq. 39 will be used to compute the specific radiator area for the several sets of operating conditions specified for Dowtherm A, corresponding to 12 kw electrical power output.

Choose:

- ϵ = 0.92, dimensionless (typical value)
- σ = 5.1×10^{-13} kw ft²°R⁴
- T_{in} = fluid temperature at radiator inlet, °R
- T_{out} = fluid temperature at radiator outlet, °R
- ΔT = $T_{\text{in}} - T_{\text{out}}$, °R
- A = radiator area, ft²
- Q = rate of heat rejection, kw
- A/Q = specific radiator area, ft²/kw

Table 4 summarizes the values of the specific radiator area, A/Q, computed from Eq. 39 for the several cases considered. In addition, the various values of P_3 and ΔT_{sub} specified in the Work Statement for Task I are listed. The degree of liquid subcooling, ΔT_{sub} , fixes $T_L = T_{\text{out}_R}$. The values of $T_{\text{mix}} = T_{\text{in}_R}$ were chosen to correspond to an inlet

gas velocity to the jet-condenser of $M = 1.0$. Then the temperature drop across the radiator, ΔT_{rad} , is determined. Figure 8 is a plot of specific radiator area versus ΔT , the degree of liquid subcooling, for the

three values of jet-condenser inlet pressure considered. It is clear from Figure 8 that the specific radiator area decreases as the jet-condenser inlet pressure increases (for constant degree of liquid subcooling). Also, at constant jet-condenser inlet pressure, the specific radiator area increases as the degree of liquid subcooling increases.

The preceding analysis and the information presented in Figure 8 are independent of the variation in overall system efficiency as a function of specific radiator area. It may be true that the total heat to be rejected in each case for fixed electrical power output also varies over a wide range. To investigate this effect, a detailed computer analysis of the overall system efficiency of a reference organic Rankine-cycle power conversion system was conducted. The nine cases listed in Table 4 formed the basis for this analysis. Figure 9 is a plot of the overall system efficiency, η , of the reference organic Rankine-cycle power conversion system versus $T_5 = T_L$ = liquid inlet temperature, for the three values of jet-condenser inlet pressure considered. The values of the liquid inlet temperature which result in maximum efficiency in each case are noted in Table 5.

Table 5 summarizes the results of the computer analysis. The jet-condenser inlet pressure, P_3 , is listed in addition to the degree of liquid subcooling, ΔT , which resulted in maximum system efficiency. The corresponding inlet temperature, T_5 , is also noted, along with the maximum system efficiency. The specific radiator area from Table 4 is listed along with the heat-rejection requirements calculated in the present analysis. Thus, the total required radiator area is easily computed and tabulated. Figure 10 is a plot of radiator area versus jet-condenser inlet pressure for a typical 12 kwe reference organic Rankine-cycle power conversion system using Dowtherm A as the working fluid. We conclude that the total radiator area is a minimum for 0.1 psia jet-condenser inlet pressure. Further, from Figure 9 and Table 5 it is noted that minimum radiator area at 0.1 psia jet-condenser inlet pressure occurs when the degree of liquid subcooling equals 38°F. At 0.3 and 0.5 psia, minimum radiator area occurs when the degree of liquid subcooling equals 57°F and 58°F, respectively.

2.6 COMPUTER EVALUATION OF JET CONDENSERS

2.6.1 Introduction

A computer program had been previously generated which predicts the rate of condensation of the hot vapor and the geometrical profile of the mixing chamber of the jet condenser.

The thirty-nine cases indicated in Table 1 were subjected to the jet-condenser computer analysis. The end result is a mixing chamber profile (radius versus length) corresponding to each set of operating conditions. These profiles are presented in Figures 11 through 22:

Figure 11, 12, and 13	Dowtherm A, 0.1 psia
Figures 14, 15, and 16	Dowtherm A, 0.3 psia
Figures 17, 18, and 19	Dowtherm A, 0.5 psia
Figure 20	Biphenyl, 0.5 psia
Figure 21	MIPB, 0.1 psia
Figure 22	MIPB, 0.5 psia

Two values of liquid droplet radius, 50μ and 100μ , were used in the analysis and both profiles are included in the figures. The 100μ droplet radius would result in a conservative design. The various mixing chamber profiles are discussed below.

The jet-condenser design parameters including the results of the computer-determined geometric profiles are summarized in Tables 6 through 10.

Table 6	Dowtherm A, 0.1 psia
Table 7	Dowtherm A, 0.3 psia
Table 8	Dowtherm A, 0.5 psia
Table 9	Biphenyl, 0.5 psia
Table 10	MIPB, 0.1 and 0.5 psia

2.6.2 Dowtherm A - Working Fluid

a. Effect of Vapor Flow Rate on Mixing Chamber Geometry and Performance

Consider Figures 11 through 19 which are plots of the mixing chamber profiles (radius versus length) for the Dowtherm A cases considered. Refer also to Tables 6, 7, and 8. Vapor flow rates of 0.027 (~ 1 kwe), 0.136 (~ 5 kwe), 0.327 (~ 12 kwe), and 0.410 (~ 15 kwe) lb/sec were considered. As the vapor flow rate increases, the diameter of the mixing chamber at the vapor inlet increases, as expected. This is evident in each of the Figures (11 through 19). Also, the final diameter (corresponding to 95% of the vapor condensed) increases as the vapor flow rate increases, since the volume of the vapor remaining determines almost completely this dimension. Figure 23 is a plot of jet-condenser mixing chamber inlet diameter versus vapor flow rate for $P_3 = 0.1$ psia, $\Delta T = 50^\circ\text{F}$, and $R_0 = 100\mu$. This figure is representative of all the cases considered.

It is also noted in each of Figures 11 through 19 that for a fixed initial droplet diameter (50μ or 100μ), the length of the mixing chamber is unchanged. This is true since the required length for

condensation is a function only of α , the ratio of liquid-to-vapor flow rates, and not of the vapor flow rate alone. α is constant in each figure. Thus, in Figure 12 (0.1 psia and 50°F subcooled liquid), curves 1 through 4 all indicate a mixing chamber length of ≈ 4.3 in. for initial droplet diameter of 50 μ , and ≈ 14.8 in. for initial droplet diameter of 100 μ . See also Table 6. Figure 24 is a plot of jet-condenser mixing chamber length (corresponding to 95% of the vapor condensed) versus vapor flow rate, indicating the independence of these two quantities.

A measure of the jet condenser mixing chamber performance would be the pressure recovery capability of the assembly. The computer program computes the isentropic velocity of the condensate stream at the mixing chamber exit and also the isentropic (maximum theoretical) pressure recovery in the diffuser. From Eq. 10 the isentropic velocity, V_{IS} , is given by:

$$V_{IS} = \frac{\alpha V_L + V_V}{1 + \alpha} \quad (40)$$

where:

α = ratio of liquid-to-vapor mass flow rate, m_L/M_V

V_L = inlet velocity of the subcooled liquid, ft/sec

V_V = inlet velocity of the vapor, ft/sec

The isentropic pressure, P_{IS} , is given by:

$$P_{IS} = 1/2 \rho V_{IS}^2 \quad (15)$$

where:

ρ = density of the condensate stream, lb/ft³

V_{IS} = isentropic velocity, ft/sec

Thus, the isentropic pressure is a function of α and independent of the vapor flow rate. Table 11 summarizes the values of the jet-condenser isentropic pressure recovery for the various Dowtherm A cases analyzed.

b. Effect of Inlet Pressure on Mixing Chamber Geometry and Performance

Consider Figures 11 through 19 and Tables 6 through 8. For fixed vapor flow rate, ΔT , and R_0 , the diameter of the mixing chamber at the vapor inlet increases as the jet-condenser inlet pressure decreases. This is evident in Figure 25 where the jet-condenser mixing chamber inlet diameter is plotted versus the jet condenser inlet pressure for

$m_v = 0.327$ lb/sec, $\Delta T = 50^\circ\text{F}$, and $R_0 = 100\mu$. As the pressure is decreased, the vapor density decreases, resulting in a greater (required) cross-sectional area and diameter.

Variation of the mixing chamber length with the jet-condenser inlet pressure is not so easily explained. Three cases are shown in Figure 26 where jet-condenser mixing chamber length is plotted versus jet-condenser inlet pressure. For the case of $m_v = 0.327$ lb/sec, $\Delta T = 25^\circ\text{F}$, and $R_0 = 100\mu$, the length is a minimum at $P_3 = 0.3$ psia. However, for the case of $m_v = 0.327$ lb/sec, $\Delta T = 50^\circ\text{F}$, and $R_0 = 100\mu$, the length increases as the jet condenser inlet pressure increases. When $m_v = 0.327$ lb/sec, $\Delta T = 75^\circ\text{F}$, and $R_0 = 100\mu$, the length is again a minimum at $P_3 = 0.3$ psia. Other values of m_v or R_0 do not change these trends. Thus, the lower inlet pressures require a mixing chamber with greater cross-sectional area and in some cases, minimum length.

The jet-condenser mixing chamber performance (pressure recovery capability) does vary with the jet-condenser mixing chamber inlet pressure as noted in Table 11. Figure 27 is a plot of jet-condenser pressure recovery ($\Psi = 1.0$) versus jet-condenser inlet pressure for $\Delta T = 25, 50$, and 75°F . It is clear that maximum pressure recovery occurs at 0.1 psia and $\Delta T = 75^\circ\text{F}$ (subcooled liquid) among the cases considered. For this case, then, pumping power requirements would be a minimum.

c. Effect of Degree of Liquid Subcooling on Mixing Chamber Geometry and Performance

Consider Figures 11 through 19 and Tables 6 through 8. At a given value of jet-condenser inlet pressure, one would expect that as the degree of liquid subcooling increases (colder fluid introduced), the jet-condenser mixing chamber inlet diameter would decrease. This is, indeed; the case as noted in the figures and tables. As the degree of liquid subcooling increases, the hot vapor stream is shocked initially with a colder liquid, resulting in a greater initial condensation rate. Figure 28 is a representative plot of jet-condenser mixing chamber inlet diameter (actually slightly downstream of the inlet, corresponding to the first printout of the computer analysis) versus the degree of liquid subcooling for three values of jet-condenser inlet pressure, P_3 . In each case it is seen that the initial diameter decreases as the degree of liquid subcooling increases.

Also, the degree of liquid subcooling should affect the overall jet-condenser mixing chamber length in a similar fashion, as the condensation rates should be greater for the colder liquids. Figure 29 is a plot of jet-condenser mixing chamber length versus degree of liquid subcooling for three values of jet-condenser inlet pressure, P_3 . In each case it is seen that the mixing chamber length (corresponding to 95% of the vapor condensed) decreases as the degree of liquid subcooling increases, as expected. In general, then, the colder the subcooled liquid, the smaller and more compact the jet-condenser mixing chamber. In Figure 29

it is noted that the three curves cross one another in a rather unordered manner. This is undoubtedly because the ratio of liquid-to-vapor flow rates, α , in each case is not constant. α was chosen so as to result in introduction of the hot vapor at a velocity corresponding to $M \geq 1.0$ when $\Delta T = 50^\circ\text{F}$ and 75°F .

The jet-condenser mixing chamber performance (pressure recovery capability) varies with the jet-condenser mixing chamber inlet pressure as noted in Table 11. Also, Figure 30 is a plot of jet-condenser pressure recovery ($\Psi = 1.0$) versus degree of liquid subcooling, for $P_3 = 0.1, 0.3, \text{ and } 0.5 \text{ psia}$. It is clear that, in each case, the pressure recovery capability increases as the degree of liquid subcooling increases. It should be noted that for $\Delta T = 25^\circ\text{F}$, the value of α corresponded to sub-sonic velocity of the hot vapor. Thus, for this case there would not be appreciable momentum transfer from the hot vapor to the subcooled liquid. Hence, the pressure recovery capability would definitely be hindered. In a like manner, the values of α corresponding to $\Delta T = 50^\circ\text{F}$ result in introduction of the hot vapor slightly supersonic. For $\Delta T = 75^\circ\text{F}$ the values of α chosen resulted in even higher (supersonic) velocity of the hot vapor. Thus, for this case the momentum transfer would be a maximum, and the pressure recovery the greatest.

d. Effect of Liquid Droplet Radius on Mixing Chamber Geometry and Performance

Consider Figures 11 through 19 and Tables 6 through 8. In the computer analysis of the 39 jet-condenser cases, two values of liquid droplet radius were considered, 50μ and 100μ . The 50μ radius was chosen, based on experimental correlations involving the interaction of high-velocity gas streams with liquid droplets, previously discussed. The 100μ radius would result in a more conservative design for the jet-condenser mixing chamber, as the total exposed surface area of the liquid droplets would be less, and, hence, the condensation rates would be lower. We would like to have the liquid droplet radius as small as possible so as to increase the total surface area exposed to the hot vapor, increase the condensation rates, and reduce the size of the mixing chamber.

Figure 31 is a plot of jet-condenser mixing chamber inlet diameter versus the liquid droplet radius. As expected, the inlet diameter increases as the liquid droplet radius increases. In some cases this effect is rather weak, but physically would be expected, as the hot vapor encounters a greater total surface area initially. Hence, the condensation rates would be greater initially. Figure 32 is a plot of jet-condenser mixing chamber length versus liquid droplet radius. Again, as expected, the mixing chamber length increases strongly as the liquid droplet radius increases. As an example, compare the mixing chamber length (corresponding to 95% of the vapor condensed) of 4.3 in. when the

liquid droplet radius is 50μ with a length of 14.8 in. when the liquid droplet radius is 100μ . These results were computed for the case where $P_3 = 0.1$ psia. Similar results are noted for the cases where $P_3 = 0.3$ and 0.5 psia.

The jet-condenser performance is independent of the liquid droplet radius at least with respect to the current mathematical model. The inlet velocity of the liquid and the hot vapor and the value of α were the same for cases involving liquid droplet radii of 50μ and 100μ .

2.6.3 Biphenyl - Working Fluid

Only two cases were analyzed with biphenyl as the working fluid. These cases corresponded to a vapor flow rate of 0.327 lb/sec, $P_3 = 0.5$ psia, $\Delta T = 50^\circ\text{F}$, and two values of the liquid droplet radius: 50μ and 100μ . The jet-condenser mixing chamber profiles are presented in Figure 20 for these cases, and the appropriate results are listed in Table 9. It is seen that the jet-condenser mixing chamber inlet diameter is relatively insensitive to the liquid droplet radius (compare 5.0 in. to 4.7 in.), but the mixing chamber length (corresponding to 99% of the vapor condensed) is a relatively strong function of the liquid droplet radius (compare 9.5 in. to 35.0 in.), as expected.

The pressure recovery capability of a jet-condenser operated with biphenyl as the working fluid at the conditions indicated in Table 9 would yield an isentropic pressure recovery of 139 psia ($\Psi = 1.0$) or a pressure recovery of 89 psia if a pressure recovery factor of 0.64 is assumed (see Table 11).

2.6.4 Monoisopropyl Biphenyl - Working Fluid

a. Effect of Inlet Pressure on Mixing Chamber Geometry and Performance

Four cases were analyzed with monoisopropyl biphenyl (MIPB) as the working fluid. These cases corresponded to a vapor flow rate of 0.327 lb/sec, $P_3 = 0.1$ and 0.5 psia, $\Delta T = 50^\circ\text{F}$, and two values of the liquid droplet radius: 50μ and 100μ . The jet-condenser mixing chamber profiles corresponding to these four cases are presented in Figures 21 ($P_3 = 0.1$ psia) and 22 ($P_3 = 0.5$ psia), and the appropriate results are listed in Table 10.

It is seen from Table 10 that the jet-condenser mixing chamber inlet diameter increases as the jet-condenser inlet pressure decreases. As the inlet pressure is decreased, the density of the vapor decreases. Hence, the required cross-sectional flow area and diameter are increased. Also, the length of the mixing chamber is relatively insensitive to changes in the jet-condenser inlet pressure. Compare a length of 18.6 in. for $P_3 = 0.1$ psia to a length of 16.1 in. when

$P_3 = 0.5$ psia. The pressure recovery of the jet-condenser is also relatively insensitive to changes in the jet-condenser inlet pressure when MIPB is employed as the working fluid. The pressure recovery capability varies by about 1% between operation at $P_3 = 0.1$ psia and 0.5 psia (Table 11).

b. Effect of Liquid Droplet Radius on Mixing Chamber Geometry and Performance

Two values of liquid droplet radius were considered: 50μ and 100μ . The results are summarized in Table 10. It is seen that the jet-condenser mixing chamber inlet diameter is quite insensitive to the liquid droplet radius both at $P_3 = 0.1$ and 0.5 psia. However, the length of the mixing chamber (corresponding to 99% of the vapor condensed) varies considerably as the liquid droplet radius varies. For $P_3 = 0.1$ psia the length is only 5.2 in. for a liquid droplet radius of 50μ , but increases to 18.6 in. when a value of 100μ is used. Similar results are noted when $P_3 = 0.5$ psia.

The jet-condenser performance is independent of the liquid droplet radius.

3. ANALYTICAL RESULT EVALUATION AND CONCEPT SELECTION - TASK II

3.1 A COMPARISON OF WORKING FLUIDS

3.1.1 Geometry

Table 12 summarizes the jet-condenser geometries obtained under identical conditions for the three working fluids. At $P_3 = 0.5$ psia the jet-condenser mixing chamber inlet diameter does not vary appreciably between the three fluids. The mixing chamber length (corresponding to 95% of the vapor condensed) is a minimum for MIPB and maximum for biphenyl. Differences in geometry are due to differences in thermal conductivity, density, viscosity, and coefficient of self-diffusion among the three fluids. These fluid properties control the rate of mass transfer and rate of condensation.

At $P_3 = 0.1$ psia only Dowtherm A and MIPB were considered as working fluids. The physical dimensions of the jet-condenser are similar, as noted in Table 12.

3.1.2 Performance

Table 13 compares the three working fluids with respect to jet-condenser pressure recovery potential. At $P_3 = 0.5$ psia, Dowtherm A exhibits the greatest pressure recovery, but the results are similar and are due to slight differences in inlet velocities and α . At $P_3 = 0.1$ psia, similar comments apply.

3.1.3 Additional Considerations

An important consideration in the choice of a working fluid for a power conversion system would be the energy available per lb of working fluid during the isentropic expansion in the turbine. The greater the energy available, the less is the required flow rate for a given electrical power output, and the smaller is the pumping power. System efficiency would increase as the pumping power requirements decreased. Table 14 summarizes the energy removal per lb of working fluid during an isentropic turbine expansion, when expanding to turbine backpressures of 0.1, 0.3, and 0.5 psia. These results are plotted in Figure 33 as Btu removed/lb of working fluid versus turbine backpressure, for the three working fluids considered. It is noted that the energy removal is a maximum for biphenyl, with Dowtherm A nearly as great. The corresponding values of MIPB are considerably less than for the other two fluids. Thus, fluid flow rates for systems operating with biphenyl and Dowtherm A would be less, pumping power less, and system efficiency would be greater.

Another important consideration in the choice of a working fluid for a power conversion system would be the fluid melting point. The melting

points for the three working fluids of interest in this study are listed in Table 14. MIPB has the lowest melting point and biphenyl the highest. At the melting point of MIPB and Dowtherm A, no trace or line heaters would be required for many applications to keep the fluid in the liquid state during periods of nonoperation with normal ambient temperature prevailing. However, with biphenyl as the working fluid it would be necessary to liquefy the fluid prior to system startup by use of appropriate trace or line heaters.

Both the energy available per lb of working fluid and the melting point are important considerations in the choice of a working fluid for a power conversion system.

3.2 RECOMMENDED CONFIGURATIONS AND OPERATING CONDITIONS FOR EXPERIMENTAL EVALUATION

The availability of the Aerojet-General reference organic Rankine-cycle power conversion system for hot gas testing has resulted in the selection of Dowtherm A as the working fluid for all testing phases of this program. Since overall system efficiency is enhanced at lower turbine backpressures, only jet-condenser assemblies operating at 0.1 psia will be considered for testing. The ORACLE power conversion system, Phase I, is sized for 12 kwe operation. Hence, the jet-condenser units chosen for fabrication correspond to a vapor flow rate of 0.327 lb/sec (12 kwe). The jet-condenser configurations selected for detailed mechanical design, fabrication, hydraulic testing, and hot gas testing are summarized below:

Working Fluid: Dowtherm A

Vapor Flow Rate: 0.327 lb/sec (12 kwe)

$P_3 = 0.1$ psia

Liquid Droplet Radius: 100 μ

Configuration (a): $\Delta T = 50^\circ\text{F}$

Configuration (b): $\Delta T = 75^\circ\text{F}$

Configuration (c): $\Delta T = 25^\circ\text{F}$

Thus, these three configurations have been reduced to a set of working drawings.

4. MECHANICAL DESIGN AND FABRICATION OF EXPERIMENTAL UNITS TASK III

4.1 MECHANICAL DESIGN OF UNITS

4.1.1 Introduction

The components required to fabricate a complete jet-condenser assembly are noted below schematically:

Design considerations for each of the components are discussed below.

4.1.2 Gas (Vapor) Nozzle

As previously discussed, we want to introduce the vapor into the jet-condenser at $M \geq 1.0$. Thus, the throat of the vapor nozzle must be properly sized to correspond to $M \geq 1.0$ for configurations (a) and (b). For configuration (c) (subsonic operation) the inlet area corresponds to the inlet (subsonic) velocity. From the continuity equation:

$$m_v = \rho_v A_v V_v$$

where:

m_v = mass flow rate of the vapor, lb/sec

ρ_v = density of the vapor, lb/ft³

A_v = cross-sectional flow area, ft²

V_v = velocity of the vapor, ft/sec

Since m_v , ρ_v , and V_v are known for each of the three configurations, we can compute the required cross-sectional area, A_v . Since configurations (a) and (b) will use supersonic gas, the throat dimension will correspond to velocity $M = 1.0$. In configuration (c) the inlet area and diameter will correspond to the inlet (subsonic) velocity. Care must be taken to shape the vapor nozzle properly as the position of the liquid nozzle further complicates the vapor flow path.

4.1.3 Liquid Nozzle

The liquid is introduced through an appropriately-sized orifice plate which results in a liquid velocity $\approx 125 - 150$ ft/sec. Applying Bernoulli's Equation to stations (1) and (2) indicated schematically below, we have:

$$P_1 + 1/2 \rho V_1^2 = P_2 + 1/2 \rho V_2^2$$

$$P_1 \gg P_2$$

$$V_2 \gg V_1$$

thus:

$$P_1 \cong 1/2 \rho V_2^2$$

$$V_2 = \sqrt{\frac{2 g P_1}{\rho}}$$

In general, the above expression is not correct due to contraction (vena contracta) of the fluid stream as it exits from the orifice. A correct expression would be of the form:

$$V_2 = C_v \sqrt{\frac{2 g P_1}{\rho}}$$

where C_v is an appropriate velocity coefficient < 1 . For a sharp orifice, $C_v \approx 0.61$. For a contoured orifice, $C_v \approx 0.97$. Knowing P_1 , ρ , and estimating C_v , one can compute V_2 . Then, the total required flow area of the orifice can be computed from the continuity equation:

$$m_L = \rho_L A_L V_L$$

Then, an orifice hole diameter can be selected and the required number of openings computed. Both sharp orifices and contoured orifices have been designed for use in this program.

4.1.4 Mixing Chamber

The profile of the mixing chamber and overall dimensions are a by-product of the jet-condenser computer analysis conducted for 39 sets of operating conditions. The inlet to the mixing chamber is appropriately modified so as to accommodate both the vapor nozzle and the liquid nozzle. The outlet from the mixing chamber is contoured so as to mate smoothly with the mixing chamber throat.

4.1.5 Mixing Chamber Throat

It is assumed that all the vapor has been condensed at the time the flow reaches the throat region. The total flow rate, $m_v + m_L$, is known. Hence, the throat cross-sectional area and diameter can be easily computed from the continuity equation:

$$m_{\text{total}} = m_L + m_v = \rho_L A_L V_L$$

Thus, the throat dimension is determined.

4.1.6 Diffuser

In the diffuser region we are interested in converting the velocity head of the condensate to static pressure. The greater the pressure recovery, the less will be the pumping power requirements, and the greater the system efficiency. Consider the schematic of the diffuser shown below:

where:

P_1 = pressure at the diffuser inlet
 D_1 = diameter at the diffuser inlet
 V_1 = velocity at the diffuser inlet
 P_2 = pressure at the diffuser outlet
 D_2 = diameter at the diffuser outlet
 V_2 = velocity at the diffuser outlet

It has been found experimentally that the total included diffuser angle, Θ , should be in the range of 5 to 7° to avoid flow separation at the walls. We would like $V_2 \approx 0$ (i.e., convert the velocity head ($1/2 \rho V_1^2$) completely to static pressure (P_2)). In order to achieve this, consider the continuity equation:

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2 = \text{constant}$$

$$\rho_1 \approx \rho_2$$

Thus:

$$A_2/A_1 = V_1/V_2$$

Take:

$$V_1/V_2 = 100$$

Then:

$$A_2/A_1 = 100 \text{ or } A_2 = 100 A_1$$

Thus:

$$D_2 = 10 D_1$$

Hence, knowing D_2 and Θ , we can size the diffuser.

In order to compute the pressure recovery, apply Bernoulli's Equation at station (1) and (2):

$$P_1 + 1/2 \rho_1 V_1^2 = P_2 + 1/2 \rho_2 V_2^2$$

$$\rho_1 \approx \rho_2$$

$$P_1 \ll P_2$$

$$V_2 \ll V_1$$

Thus:

$$P_2 \cong 1/2 \rho_1 V_1^2$$

The jet condenser assembly drawing corresponding to configuration (a) - (50°F subcooled liquid) is shown in Figure 34, while the assembly drawing for configuration (b) - (75°F subcooled liquid) is presented in Figure 35.

Configuration (c) corresponded to 25°F subcooled liquid and subsonic operation. The mixing chamber was found to be extremely large physically (Table 6) and would have resulted in excessive fabrication costs. Also, the subsonic operation results in a rather poor design as the momentum interchange between the hot vapor and the subcooled liquid is poor. In addition, any pressure fluctuations in the mixing chamber will be reflected in a pressure fluctuation at the turbine outlet. To make this configuration operable at a supersonic vapor velocity, it would be necessary to increase the ratio of liquid-to-vapor flow rate, α , to an extremely large value. This mode of operation is unsuitable for power conversion systems, as it would result in excessive pumping power requirements and reduced system efficiency. Thus, agreement was reached with Mr. G Leighton, AEC Project Manager, that configuration (c) would be reduced to a layout drawing only and would not be fabricated for testing.

4.2 FABRICATION OF UNITS

The fabrication of jet-condenser assemblies corresponding to configurations (a) and (b) was completed on November 22, 1967, and prepared for hydraulic testing (Task IV) and hot gas testing (Task V). All necessary machining of the required hardware was accomplished at Aerojet-General. All parts are machined of aluminum for minimum cost and minimum weight. Because of the size and the complexity of the jet-condenser vapor nozzle housing, the jet-condenser mixing chambers, and the jet-condenser mixing chamber throat, these parts were initially cast of aluminum by an outside vendor, and delivered to Aerojet for final machining. This procedure worked very well and resulted in a reasonable schedule and absolute minimum cost. Direct machining of these parts from large aluminum billets would have resulted in exorbitant costs.

Photographs of the completely assembled jet-condensers are shown in Figure 36 (configuration (a)) and Figure 37 (configuration (b)).

5. SUMMARY AND CONCLUSIONS

Tasks I, II, and III of AEC Contract No. AT(04-3)-728 have been completed. Expenditures for these tasks have been as planned previously. Hydraulic and hot-vapor testing which make up Tasks IV and V of the subject contract will be given full attention in the coming months. Completion of all work outlined in the Work Statement is anticipated by the scheduled completion date.

The major conclusions gleaned from Tasks I, II, and III are enumerated below:

1. A mathematical design analysis (Task I) was conducted on 39 sets of operating conditions involving three working fluids. Various flow rates, inlet pressures, and degrees of liquid subcooling were considered. For each set of operating conditions specified in the Work Statement, a computer analysis yielded a corresponding jet-condenser geometric profile. The effect of vapor flow rate, inlet pressure, degree of liquid subcooling, and liquid droplet radius on the geometric profile and the jet-condenser performance was studied in detail. The following trends were noted:

(a) The diameter of the jet-condenser mixing chamber at the vapor inlet increases as the vapor flow rate increases.

(b) The length of the jet-condenser mixing chamber is independent of the vapor flow rate (constant α).

(c) The diameter of the jet-condenser mixing chamber at the vapor inlet decreases as the jet-condenser inlet pressure increases.

(d) The diameter of the jet-condenser mixing chamber at the vapor inlet decreases as the degree of the liquid subcooling increases.

(e) The length of the jet-condenser mixing chamber decreases as the degree of liquid subcooling increases.

(f) The pressure recovery of the jet-condenser increases as the degree of liquid subcooling increases.

(g) The diameter of the jet-condenser mixing chamber at the vapor inlet increases as the liquid droplet radius increases.

(h) The length of the jet-condenser mixing chamber decreases as the liquid droplet radius decreases.

2. The thirty-nine configurations generated in Task I were carefully compared (Task II) with respect to differences in working fluids, geometry, and performance. Three configurations corresponding to operation with Dowtherm A as the working fluid at a vapor flow rate consistent with generation of 12 kwe in a suitable power conversion system operating at a jet condenser inlet pressure of 0.1 psia were chosen for detailed mechanical design. The three configurations will be operated with 25, 50, and 75°F subcooled liquid for condensation purposes. It was agreed by Aerojet-General Corporation and the AEC Project Manager, Mr. G. Leighton, that only the two configurations corresponding to operation with 50°F and 75°F subcooled liquid would be fabricated. The jet condenser assembly corresponding to operation with 25°F subcooled liquid is very large physically and designed to operate with subsonic hot vapor. This configuration will be reduced to a layout drawing only.

3. The detailed mechanical design of the configurations corresponding to operation with 50°F and 75°F subcooled liquid was completed, and a layout drawing of the 25°F configuration was also generated. All detail parts for configurations (a) 50°F subcooled liquid and (b) 75°F subcooled liquid were machined of aluminum for minimum cost and weight at Aerojet-General Corporation. Several of the detail parts were initially cast of aluminum by an outside vendor and then final machined at Aerojet. Complete jet condenser assemblies corresponding to configurations (a) and (b) have been assembled and are now ready for hydraulic (Task IV) and hot vapor (Task V) testing.

REFERENCES

1. Donald B. Mackay, Design of Space Powerplants, Prentice-Hall, Inc., 1963, Chapter 9.
2. M. G. Coombs and R. A. Stone, Progress in Astronautics and Rocketry - Space Power Systems, Volume 4, "SNAP-2 Radiative Condenser Design," Academic Press, New York, 1961, p. 309.
3. M. G. Coombs, R. A. Stone, and T. Kapus, "The SNAP-2 Radiative Condenser Analysis," NAA-SR-5317.

TABLE 1 - Summary of Jet-Condenser Cases Analyzed

Working Fluid	Dowtherm A	Biphenyl	MIPB
Vapor Flow Rate, lb/sec	0.0273, 0.136, 0.327, 0.410	0.327	0.327
Electrical Power Output, kwe	1, 5, 12, 15	12	12
Inlet Jet-Condenser Pressure, psia	0.1, 0.3, 0.5	0.5	0.1, 0.5
Degree of Liquid Subcooling, °F	25, 50, 75	50	50
No. of Cases Analyzed	36	1	2

Table 1

TABLE 2 - Summary of Vapor and Subcooled Liquid Inlet

Conditions to Jet-Condenser

<u>Vapor Conditions</u>		<u>Subcooled Liquid</u>		
<u>Dowtherm A</u>				
<u>Inlet Pressure</u>	<u>Sat. Temperature</u>	<u>$\Delta T = 25^{\circ}\text{F}$</u>	<u>$\Delta T = 50^{\circ}\text{F}$</u>	<u>$\Delta T = 75^{\circ}\text{F}$</u>
0.1 psia	223°F	198°F	173°F	148°F
0.3 psia	267°F	242°F	217°F	192°F
0.5 psia	289°F	264°F	239°F	214°F
<u>Biphenyl</u>				
0.1 psia	218°F	193°F	168°F	143°F
0.3 psia	251°F	226°F	201°F	176°F
0.5 psia	284°F	259°F	234°F	209°F
<u>Monoisopropyl Biphenyl (MIPB)</u>				
0.1 psia	265°F	240°F	215°F	190°F
0.3 psia	315°F	290°F	265°F	240°F
0.5 psia	340°F	315°F	290°F	265°F

TABLE 3

Summary of Parameters for Computer Analysis

Dowtherm A; $P_3 = 0.5$ psia; $T_3 = 289^\circ\text{F}$; $H_3^g = 244.1$ Btu/lb

	T_{mix} $^\circ\text{F}$	P_3/P_{mix}	M	V ft/sec	P_{mix} psia	α
$\Delta T = 25^\circ\text{F}$	287.3	1.045	0.298	143.3	0.478	28
$T_L = 264^\circ\text{F}$						
$H_L = 86.2$ Btu/lb						
$\Delta T = 50^\circ\text{F}$	264.7	1.75	1.04	495	0.286	28
$T_L = 239^\circ\text{F}$						
$H_L = 75.2$ Btu/lb						
$\Delta T = 75^\circ\text{F}$	251.7	2.41	1.3	618	0.208	16
$T_L = 214^\circ\text{F}$						
$H_L = 64.2$ Btu/lb						
<u>Dowtherm A</u> ; $P_3 = 0.3$ psia; $T_3 = 267^\circ\text{F}$; $H_3^g = 235.9$ Btu/lb						
$\Delta T = 25^\circ\text{F}$	266.2	1.01	0.13	61.5	0.298	28
$T_L = 242^\circ\text{F}$						
$H_L = 76.5$ Btu/lb						
$\Delta T = 50^\circ\text{F}$	243.2	1.78	1.052	494	0.169	28
$T_L = 217^\circ\text{F}$						
$H_L = 65.5$ Btu/lb						

	$T_{\text{mix}}^{\circ\text{F}}$	P_3/P_{mix}	M	V ft/sec	P_{mix} psia	α
$\Delta T = 75^{\circ}\text{F}$	238.6	2.00	1.15	540	0.15	16
$T_L = 192^{\circ}\text{F}$						
$H_L = 54.8 \text{ Btu/lb}$						
<u>Dowtherm A</u> ; $P_3 = 0.1 \text{ psia}$; $T_3 = 223^{\circ}\text{F}$; $H_3^g = 220.1 \text{ Btu/lb}$						
$\Delta T = 25^{\circ}\text{F}$	222.2	1.03	0.24	109.5	0.097	28
$T_L = 198^{\circ}\text{F}$						
$H_L = 57 \text{ Btu/lb}$						
$\Delta T = 50^{\circ}\text{F}$	204.6	1.77	1.048	477	0.0566	24
$T_L = 173^{\circ}\text{F}$						
$H_L = 47 \text{ Btu/lb}$						
$\Delta T = 75^{\circ}\text{F}$	196.7	2.26	1.25	567	0.0443	16
$T_L = 148^{\circ}\text{F}$						
$H_L = 36.5 \text{ Btu/lb}$						
<u>Biphenyl</u> ; $P_3 = 0.5 \text{ psia}$; $T_3 = 285^{\circ}\text{F}$; $H_3^g = 209.8 \text{ Btu/lb}$						
$\Delta T = 50^{\circ}\text{F}$	263	1.66	1.0	490	0.301	28
$T_L = 235^{\circ}\text{F}$						
$H_L = 31.6 \text{ Btu/lb}$						
<u>Monoisopropyl Biphenyl</u> ; $P_3 = 0.5 \text{ psia}$; $T_3 = 338.2^{\circ}\text{F}$; $H_3^g = 94.47 \text{ Btu/lb}$						
$\Delta T = 50^{\circ}\text{F}$	315	1.667	1.0	490	0.3	24
$T_L = 288.2^{\circ}\text{F}$						
$H_L = 60.0 \text{ Btu/lb}$						
<u>Monoisopropyl Biphenyl</u> ; $P_3 = 0.1 \text{ psia}$; $T_3 = 268.1^{\circ}\text{F}$; $H_3^g = 68.8 \text{ Btu/lb}$						

	<u>T_{mix} °F</u>	<u>P₃ / P_{mix}</u>	<u>M</u>	<u>V ft/sec</u>	<u>P_{mix} psia</u>	<u>α</u>
ΔT = 50°F	245	1.667	1.0	470	0.06	26
T _L = 218.1°F						
H _L = -90.0 Btu/lb						

TABLE 4

Specific Radiator Area - Dowtherm A Fluid

P_3	ΔT_{sub}	$T_L = T_{\text{out}_R}$	$T_{\text{mix}} = T_{\text{in}_R}$	ΔT_{rad}	A/Q
0.1 psia	25°F	198°F	210°F	12°F	11.2 ft ² /Kw
0.1	50	173	210	37	11.9
0.1	75	148	210	62	12.9
0.3	25	242	246	4	8.9
0.3	50	217	246	29	9.3
0.3	75	192	246	54	10.1
0.5	25	264	267	3	7.1
0.5	50	239	267	28	8.4
0.5	75	214	267	53	9.0

Table 4

TABLE 5

Maximum System Efficiency and Total Required Radiator Area - Dowtherm A Fluid

P_3	ΔT_{sub}	$T_5 = T_L$	Maximum Efficiency	Specific Radiator Area	Heat Rejected	Total Area
0.1 psia	38°F	185°F	24.75%	11.5 ft ² /Kw	36.8 Kw	424 ft ²
0.3	57	210	20.70	9.6	46.1	442
0.5	59	230	18.80	8.6	51.9	446

Table 5

TABLE 6

Summary of Jet-Condenser Design Parameters: Dowtherm A, $P_3 = 0.1$ PSIA

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference- FIGURE
0.027 lb/sec	25°F	50 μ	3.4 in.	7.9 in.	20
0.136	25	50	7.7	7.9	20
0.327	25	50	11.9	7.9	20
0.410	25	50	13.3	7.9	20
0.027	25	100 μ	3.7	29.3	20
0.136	25	100	8.2	29.3	20
0.327	25	100	12.7	29.3	20
0.410	25	100	14.2	29.3	20
0.027	50	50	2.6	4.3	21
0.136	50	50	5.8	4.3	21
0.327	50	50	8.9	4.3	21
0.410	50	50	10.0	4.3	21
0.027	50	100	2.7	14.8	21
0.136	50	100	6.1	14.8	21
0.327	50	100	9.5	14.8	21
0.410	50	100	10.7	14.8	21

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference- FIGURE
0.027	75	50	2.4	2.9	22
0.136	75	50	5.5	2.9	22
0.327	75	50	8.5	2.9	22
0.410	75	50	9.5	2.9	22
0.027	75	100	2.8	9.8	22
0.136	75	100	6.2	9.8	22
0.327	75	100	9.6	9.8	22
0.410	75	100	10.7	9.8	22

*Slightly downstream of the inlet, corresponding to first computer output

**95% of the vapor condensed

TABLE 7

Summary of Jet-Condenser Design Parameters: Dowtherm A, $P_3 = 0.3$ PSIA

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference- FIGURE
0.027 lb/sec	25°F	50 μ	1.9 in.	6.7 in.	23
0.136	25	50	4.2	6.7	23
0.327	25	50	6.4	6.7	23
0.410	25	50	7.2	6.7	23
0.027	25	100 μ	1.9	24.1	23
0.136	25	100	4.3	24.1	23
0.327	25	100	6.7	24.1	23
0.410	25	100	7.5	24.1	23
0.027	50	50	1.7	5.0	24
0.136	50	50	3.8	5.0	24
0.327	50	50	5.8	5.0	24
0.410	50	50	6.5	5.0	24
0.027	50	100	1.7	18.1	24
0.136	50	100	3.7	18.1	24
0.327	50	100	5.7	18.1	24
0.410	50	100	6.4	18.1	24
0.027	75	50	1.2	2.4	25

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference- FIGURE
0.136	75	50	2.6	2.4	25
0.327	75	50	4.1	2.4	25
0.410	75	50	4.6	2.4	25
0.027	75	100	1.3	8.4	25
0.136	75	100	2.9	8.4	25
0.327	75	100	4.5	8.4	25
0.410	75	100	5.1	8.4	25

*Slightly downstream of the inlet, corresponding to first computer output

**95% of the vapor condensed

TABLE 8

Summary of Jet Condenser Design Parameters: Dowtherm A, $P_3 = 0.5$ PSIA

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference- FIGURE
0.027 lb/sec	25°F	50 μ	1.7 in	8.9 in	26
0.136	25	50	3.9	8.9	26
0.327	25	50	6.0	8.9	26
0.410	25	50	6.7	8.9	26
0.027	25	100 μ	1.8	32.6	26
0.136	25	100	4.1	32.6	26
0.327	25	100	6.4	32.6	26
0.410	25	100	7.1	32.6	26
0.027	50	50	1.4	5.6	27
0.136	50	50	3.2	5.6	27
0.327	50	50	4.9	5.6	27
0.410	50	50	5.5	5.6	27
0.027	50	100	1.4	20.0	27
0.136	50	100	3.1	20.0	27
0.327	50	100	4.7	20.0	27
0.410	50	100	5.3	20.0	27

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference- FIGURE
0.027	75	50	1.2	3.2	28
0.136	75	50	2.6	3.2	28
0.327	75	50	4.0	3.2	28
0.410	75	50	4.5	3.2	28
0.027	75	100	1.2	11.3	28
0.136	75	100	2.7	11.3	28
0.327	75	100	4.2	11.3	28
0.410	75	100	4.7	11.3	28

*Slightly downstream of the inlet, corresponding to first computer output

**95% of the vapor condensed

TABLE 9

Summary of Jet-Condenser Design Parameters: Biphenyl, $P_3 = 0.5$ PSIA

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference-FIGURE
0.327 lb/sec	50°F	50 μ	5.0 in.	9.5 in.	29
0.327	50	100 μ	4.7	35.0	29

*Slightly downstream of the inlet, corresponding to first computer output

**99% of the vapor condensed

TABLE 10

Summary of Jet-Condenser Design Parameters:
Monoisopropyl Biphenyl

Vapor Flow Rate	ΔT	Liquid Droplet Radius	Diameter at Inlet*	Length**	Reference-FIGURE
$P_3 = 0.1$ PSIA					
0.327 lb/sec	50°F	50 μ	10.9 in.	5.2 in.	30
0.327	50	100 μ	10.9	18.6	30
$P_3 = 0.5$ PSIA					
0.327	50	50 μ	5.3	4.6	31
0.327	50	100 μ	5.2	16.1	31

*Slightly downstream of the inlet, corresponding to first computer output

**99% of the vapor condensed

TABLE 11
Jet-Condenser Isentropic Pressure Recovery

Fluid	P ₃	ΔT	P ($\psi = 1$) IS
Dowtherm A	0.1 PSIA	25°F	121 PSIA
Dowtherm A	0.1	50	150
Dowtherm A	0.1	75	175
Dowtherm A	0.3	25	116
Dowtherm A	0.3	50	144
Dowtherm A	0.3	75	168
Dowtherm A	0.5	25	119
Dowtherm A	0.5	50	143
Dowtherm A	0.5	75	165
Biphenyl	0.5	50	139
MIPB	0.1	50	135
MIPB	0.5	50	134

TABLE 12

Comparison of Working Fluids: Geometry of Jet-Condenser

Fluid	α	Diameter at Inlet*	Length**
$P_3 = 0.5 \text{ PSIA}$			
Dowtherm A	28	4.7 in.	20.0 in.
Biphenyl	28	4.7	30.7
MIPB	24	5.2	14.2
$P_3 = 0.1 \text{ PSIA}$			
Dowtherm A	24	9.5 in.	14.8 in.
MIPB	26	10.9	17.2

*Slightly downstream of the inlet, corresponding to first computer output

**95% of the vapor condensed

Constant conditions for all of the above cases:

$$m_v = 0.327 \text{ lb/sec}$$

$$\Delta T = 50^\circ\text{F}$$

$$R_0 = 100\mu$$

TABLE 13

Comparison of Working Fluids: Performance of Jet-Condenser

Fluid	α	$P_{\text{Isentropic}} (\Psi = 1.0)$	$P (\Psi = 0.64)$
$P_3 = 0.5 \text{ PSIA}$			
Dowtherm A	28	143 psia	92 psia
Biphenyl	28	139	89
MIPB	24	134	86
$P_3 = 0.1 \text{ PSIA}$			
Dowtherm A	24	150 psia	96 psia
MIPB	26	135	87

Constant conditions for all of the above cases:

$$\dot{m}_v = 0.327 \text{ lb/sec}$$

$$\Delta T = 50^\circ\text{F}$$

$$R_0 = 100\mu$$

TABLE 14

Energy Removal During Isentropic Turbine Expansion

Fluid	Turbine Backpressure	Energy Removal	Melting Point
Dowtherm A	0.1 psia	90 Btu/lbm	53.6°F
Biphenyl	0.1	92	158
MIPB	0.1	69	-60
Dowtherm A	0.3	75	
Biphenyl	0.3	78	
MIPB	0.3	57	
Dowtherm A	0.5	69	
Biphenyl	0.5	71	
MIPB	0.5	52	

Initial state of fluid at entry to turbine: 700°F Saturated Vapor

APPENDIX

Work Statement

The contractor will provide the necessary personnel, facilities and services to conduct exploratory development to determine the design and performance of a jet-condenser as applied to 1-15 kwe organic Rankine-cycle power systems for turbine exhaust pressures ranging from 0.5 psia toward an ultimate goal of 0.1 psia.

The units which actually are tested will be sized for a single power level of 12 kwe. (Corresponding to the 0.327 lb/sec flow rate shown below.) One fluid shall be used for hot vapor testing, the eutectic mixture of biphenyl and phenyl ether (Dowtherm A). Water shall be used for the Hydraulic Evaluation Tests.

A. TASK I MATHEMATICAL DESIGN ANALYSIS

The contractor shall employ his previously developed jet-condenser evaluation computer program to establish the following:

1. The physical dimensions and theoretical performance that can be anticipated for jet-condensers operating under the following approximate conditions:

a. The eutectic mixture of biphenyl and phenyl ether shall be used as the working fluid and at flow rates of 0.0273 lb/sec, 0.136 lb/sec, 0.327 lb/sec and 0.410 lb/sec. For each value of flow rate, the following entry conditions to the jet condenser shall be used:

- (1) 0.5 psia
- (2) 0.3 psia
- (3) 0.1 psia

The liquid coolant will be taken to be 25°F, 50°F, and 75°F lower than the vapor saturation temperature corresponding to each of the above cited inlet pressures.

b. Biphenyl shall be used as the working fluid at a flow rate of 0.327 lb/sec and at a jet-condenser inlet pressure of 0.5 psia. The liquid coolant temperature will be taken to be 50°F lower than the saturation temperature associated with a jet-condenser inlet pressure of 0.5 psia.

c. Monoisopropyl biphenyl shall be used as the working fluid at a flow rate of 0.327 lb/sec, and a jet-condenser inlet pressure of 0.1 and 0.5 psia. The liquid coolant temperature will be taken to be 50°F lower than the saturation temperature associated with each of the two jet-condenser inlet pressures.

B. TASK II - ANALYTICAL RESULT EVALUATION AND CONCEPT SELECTION

The contractor shall evaluate the results from Task I to determine the mechanical concepts and operating conditions to be used in the design experimental units. From this evaluation, the contractor will recommend to the Project Manager, one or more promising configurations. Within 10 working days following receipt of the contractor's recommendations, the Project Manager will transmit his approval of the configurations to be designed and fabricated under Task III to the contractor. No response shall constitute approval.

C. TASK III - FABRICATE EXPERIMENTAL UNITS

The contractor shall reduce the configurations approved by the Project Manager to detailed mechanical drawings and manufacture one or more units of each selected configuration. This fabrication shall employ readily available, conventional materials such as aluminum, brass, or carbon steel.

D. TASK IV - HYDRAULIC EVALUATION TEST

A test plan document shall be submitted to the Project Manager for approval, at least thirty (30) days prior to the initiation of test. Within ten (10) working days following receipt of the test plan document by the Project Manager, notice shall be given to the contractor of approval to proceed with work under this task. No response shall constitute approval.

The contractor shall conduct a performance screening evaluation of the experimental units. These tests shall be conducted in the Von Karman Center Hydraulic Laboratory and shall employ ambient-temperature water as the working fluid. This screening will include one or more redesigns or modifications of the experimental units in which test results shall be employed in conjunction with the computerized analytical model with the objective of improving operational performance.

At the conclusion of the screening tests, one or more promising design configurations shall be recommended to the Project Manager for further evaluation in a hot-vapor test. Within 10 working days following receipt of the contractor's recommendations, the Project Manager will transmit his approval of the configurations to be evaluated by hot vapor testing. No response shall constitute approval.

E. TASK V - EVALUATION TEST WITH HOT DOWTHERM A VAPOR

A test plan document shall be submitted to the Project Manager for approval, at least thirty (30) days prior to the initiation of test. Within ten (10) working days following receipt of the test plan document by the Project Manager, notice shall be given to the contractor of approval to proceed with work under this task. No response shall constitute approval.

The contractor shall design, fabricate and assemble the necessary components to provide inlet vapor and liquid conditions at the approved hot-test operating point.

It is anticipated that these components will include, but not necessarily be limited to, the following:

1. Turbine simulator throttle and cooler.
2. A recuperator or heat exchanger simulating recuperator thermal and flow performance.
3. Support test stand.
4. Temperature-flow and pressure sensors as required for performance evaluation of the jet-condenser.

The contractor will conduct a program of hot-vapor testing. This testing will be conducted in the Von Karman Center "ORACLE" Power System Experimental Facility, on a 40-hour work week, 8-hour work day basis, i.e., no endurance testing is anticipated in the program.

The hot-test period shall include the time required for one or more cycles of test unit removal, and replacement with alternate or modified experimental test assemblies.

F. TASK VI - REPORTING (can be covered by separate administrative letter)

1. Monthly Progress Reports

Informal monthly progress reports will be submitted 10 days after the close of the reporting period. These reports shall be in the form given in Exhibit 1 of this Appendix.

2. Midterm Report

Four weeks after the completion of Task IV the contractor shall submit a draft of a formal midterm report, including the results of Tasks I, II, III, and IV. This report shall contain a technical discussion of the work accomplished, technical problems encountered

and their solution, pertinent test data and test results. Drawings, sketches and photographs shall be included to support the written discussion.

3. Final Report

Six weeks after the conclusion of the testing program the contractor shall submit a draft of a formal final report including a summary of the midterm report and the results of Task V. The content of this report shall correspond to the requirements for the midterm report.

4. Oral Report

The contractor shall conduct a briefing at U.S.A.E.C. Headquarters, summarizing the work up to the time of the briefing. This briefing shall be conducted at a time requested by the Project Manager.

5. Test Plans

Test Plans shall be submitted covering Tasks IV and V at least thirty (30) days in advance of the test. The plan shall include, but not be limited to the following:

- a. Test objectives
- b. Description of test.
- c. Discussion of potential problems.
- d. List of all data to be taken, accuracy required, and procedure to be used for acquiring data.
- e. Discussion of the usefulness and adequacy of the data and analyses to be performed.
- f. List of specific types of instrumentation to be used and their limitations.
- g. Number and approximate location of items of instrumentation if pertinent to the objectives of the test.
- h. Schedule of testing to be performed.
- i. Number and type of data points to be taken for each test run.
- j. Number of test runs necessary to complete the task.

G. GOVERNMENT FURNISHED DATA

On or before the date of work initiation under this contract the Government shall supply the contractor with all necessary thermodynamic and physical properties of the three fluids mentioned in Task I.

1166-764

Breadboard Model of 12-kw Organic
Rankine-Cycle Power System Schematic
Figure 1

ORACLE POWER SYSTEM TEST FACILITY

Figure 2

VON KARMAN CENTER

NOTE:

- 1 Circled numbers denote stations, See Figure 1.

α Minimum Required for Complete Condensation vs ΔT , Degree of Liquid Subcooling--Dowtherm A Fluid

Figure 4

α Minimum Required for Complete Condensation vs P_3 , Jet Condenser Inlet Pressure--Dowtherm A Fluid

Figure 5

α Required for $M = 1$ vs ΔT , Degree of Liquid Subcooling--Dowtherm A Fluid
Figure 6

1267-NF-1150

α Required for $M = 1$ vs P_3 , Jet Condenser Inlet Pressure--Dowtherm A Fluid

Figure 7

Specific Radiator Area vs ΔT , Degree of Liquid Subcooling--Dowtherm A Fluid

Figure 8

System Efficiency of the ORACLE Power Conversion System
vs Liquid Inlet Temperature to the Jet Condenser--Dowtherm A Fluid

Figure 9

1267-NF-1153

Radiator Area vs Jet Condenser Inlet Pressure
12 kwe Organic Rankine Cycle System - Dowtherm A Fluid

Figure 10

LEGEND:

DOWTHERM A, $P_3 = 0.1$ PSIA

$R_0 = 50$ Microns

Curve Line

m_v
lb/sec

1
2
3
4

.027
.136
.327
.410

$R_0 = 100$ Microns

5
6
7
8

.027
.136
.327
.410

Jet Condenser Mixing Chamber Profile
Figure 12

Jet Condenser Mixing Chamber Profile

Figure 13

1

Jet Condenser Mixing Chamber Profile

Figure 14

1267-NF-1158

LEGEND:

DOWTHERM A, P = 0.3 PSIA

$R_0 = 50$ Microns

m_v
lb/sec

$R_0 = 100$ Microns

Jet Condenser Mixing Chamber Profile

Figure 15

Jet Condenser Mixing Chamber Profile

Figure 16

Jet Condenser Mixing Chamber Profile

Figure 17

Jet Condenser Mixing Chamber Profile

Figure 18

LEGEND:

DOWTHERM A, P = 0.5 PSIA
3

R₀ = 50 Microns

Curve Line

1
2
3
4

m_v
lb/sec

.027
.136
.327
.410

R₀ = 100 Microns

5
6
7
8

.027
.136
.327
.410

Jet Condenser Mixing Chamber Profile

Figure 19

Jet Condenser Mixing Chamber Profile

Figure 21

Jet Condenser Mixing Chamber Profile

Figure 22

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Jet Condenser Mixing Chamber Inlet Diameter vs Vapor Flow Rate

Figure 23

1267-NF-1167

Jet Condenser Vixing Chamber Length vs Vapor Flow Rate

Figure 24

Jet Condenser Mixing Chamber Inlet Diameter vs Jet Condenser Inlet Pressure

Jet Condenser Mixing Chamber Length vs Jet Condenser Inlet Pressure

Jet Condenser Pressure Recovery vs Jet Condenser Inlet Pressure
for Dowtherm A

Jet Condenser Mixing Chamber Inlet Diameter vs Degree of Liquid Subcooling

Jet Condenser Mixing Chamber Length vs Degree of Liquid Subcooling

Figure 29

Jet Condenser Pressure Recovery vs Degree of Liquid Subcooling-Dowtherm A

Figure 30

Jet Condenser Mixing Chamber Inlet Diameter vs Liquid Droplet Radius

Jet Condenser Mixing Chamber Length vs. Liquid Droplet Radius

1267-NF-1176

Energy Removal During Isentropic Turbine
Expansion vs Turbine Backpressure

Figure 33

NOTES:
 Δ PART NUMBER 1265602-1 AND-2 THROAT EXTENSION MAY BE
 SUBSTITUTED FOR PART NUMBER 1265602-3 THROAT EXTENSION.

Jet Condenser Assembly Drawing, Configuration (a)

NOTES: 1. REMOVE ALL BURRS AND SHARP EDGES

Jet Condenser Assembly Drawing, Configuration (b)

Figure 36

JET CONDENSER, CONFIGURATION (a) FULLY ASSEMBLED

JET CONDENSER, CONFIGURATION (b) FULLY ASSEMBLED