DOE/CH/10539--T1

FINAL

***Report on
Accelerated Life Test of the
USDOE OC-OTEC Experimental System
Refurbished with Magnetic Bearings
for the 3rd Stage Vacuum Compressor***

Prepared for

***U.S. Dept. of Energy
NREL Area Office 9
DE-AC36-92CH10538***

RECEIVED

APR 27 1998

OSTI

Prepared by

Luis A. Vega, Ph.D.

***Pacific International Center for
High Technology Research***

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

MASTER

April 1997

DISCLAIMER

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

DISCLAIMER

**Portions of this document may be illegible
electronic image products. Images are
produced from the best available original
document.**

Executive Summary

This report documents the accelerated life test performed to evaluate the viability of the magnetic bearing system installed in the stage 3 vacuum pump of the OC-OTEC Experimental Apparatus. PICHTR was directed by DOE to operate the OC-OTEC plant for at least 500 hours over a three month period, with the system operating at 100% design and above.

It had been previously determined that the maximum achievable operating parameters for the OC-OTEC plant are dictated by the speed and (motor) current limitations of the centrifugal vacuum pumps. As presently configured, without the stage 4 vacuum pump, this results in practical upper limits for system operation given by having the stage 1 and 2 vacuum pumps operate at design speeds and the stage 3 pump at $\approx 94\%$ of design speed. These limits were observed during the three month test phase reported here.

The plant was operated for 535 hours during the period September-November 1996. The time history records required to evaluate the performance of the system and its subsystems were obtained and stored for on-line and future analysis, as is done whenever the plant is operated. Data records obtained since the initiation of operations in 1993 were previously used to confirm the predicted performance of the experimental OC-OTEC Apparatus. For the purpose of this report, one representative set of records is used to estimate and document the performance of the vacuum compressors subsystem as well as the OC-OTEC system performance during the accelerated life test. This is followed by the discussion of the life tests (time-to-failure) of the vacuum compressors subsystem.

It is concluded that units with magnetic bearings will provide the required thousands of hours of operation. The power consumption estimated for a vacuum compressors subsystem, including a new stage 4 with magnetic bearings, is estimated at 35 kW under design conditions, meeting the DOE goal of ≤ 40 kW. It is recommended that high speed centrifugal pumps with magnetic bearings be used in future OC-OTEC systems to achieve extended life cycles, relatively low power consumption and, therefore, optimum net power.

Table of Contents

Introduction	1
Section I:	
System Performance	1
Measured Time History Records.....	1
Evaporator Subsystem Performance Parameters	3
Evaporator Thermal Effectiveness	3
Steam Generation Rate.....	4
Turbine-Diffuser-Generator Subsystem Performance Parameters	4
Turbine Isentropic Power	4
Gross Power from Turbine Generator	5
Turbine-Generator System Efficiency	5
Direct Contact Condenser Subsystem Performance Parameters	6
Fraction of Steam Condensed	6
DCC Steam Side Pressure Drop.....	6
DCC Thermal Effectiveness.....	6
Vacuum Vent Ratio.....	7
Vacuum Compressor Subsystem Performance Parameters	8
Section II:	
Vacuum Pumps Life Cycle Data.....	9
Third Stage Vacuum Compressor Refurbishment	10
Conclusions	12
References	13

List of Tables

Table 1A. Stage 1A Centrifugal Compressor	14,15
Table 1B. Stage 1B Centrifugal Compressor	16,17
Table 1C. Stage 1C Centrifugal Compressor	18
Table 2. Stage 2 Centrifugal Compressor	19
Table 3. Stage 3 Centrifugal Compressor	20
Table 4. Stage 4 Centrifugal Compressor	21

List of Figures

Figure 1. Evaporator Instrumentation Schematic.....	22
Figure 2. Turbine-Generator Instrumentation Schematic	22
Figure 3. Direct-Contact Condenser Instrumentation Schematic.....	23
Figure 4. Vacuum Compressor Subsystem Instrumentation	23
Figure 5. OC-OTEC System Schematic.....	24

Appendices

Appendix A	Representative Time History Records
Appendix B	Uncertainty of Derived Parameters (Power Production Mode)

Introduction

This report documents the accelerated life test (time-to-failure) performed, at the request of DOE, to evaluate the viability of the magnetic bearing system installed in the stage 3 vacuum pump. To this effect the plant was successfully operated for over 500 hours during the period September-November 1996. The first part of this report discusses system performance by deriving subsystem and system performance parameters from a typical record. This is followed by the discussion of the life tests. The instrumentation used to estimate the performance parameters given here is depicted in Figures 1 through 5. The third stage pump was operated for 535 hours without incident. It is concluded that magnetic bearings are the preferable choice for the OC-OTEC centrifugal vacuum pumps.

Section I: System Performance

The OC-OTEC Experimental Apparatus can be considered as encompassing two parts that are tested simultaneously:

(1) **Subsystem Tests.** Subsystem Tests, in the power production mode, include the performance measurements on (a) spout evaporator (Figure 1), (b) turbine-generator (Figure 2), (c) direct-contact condenser (Figure 3), and (d) vacuum compression system (Figure 4).

(2) **System Tests.** The system tests focus on the power production of the OC-OTEC system. The system schematic is given in Figure 5.

Measured Time History Records

The following is a list of all parameters recorded during the accelerated life cycle test:

Warm Seawater Supply Flow Rate	FE-10
Cold Seawater Supply Flow Rate	FE-20
Exhaust Mass Flow Meter (exit of V.C.)	FE-60
Turbine Inlet Total Steam Pressure	PT-32
Diffuser Outlet Steam Pressure	PT-33
1st Stage V.C. Inlet Pressure	PT-105
2nd Stage V.C. Inlet Pressure	PT-104
3rd Stage V.C. Inlet Pressure	PT-103
4th Stage V.C. Inlet Pressure	PT-102
5th Stage V.C. Inlet Pressure	PT-101
Turbine Pressure Drop	DPT-31
Diffuser Pressure Recovery	DPT-32
Co-Current (1st Stage) DCC Pressure Drop	DPT-33

Counter-Current (2nd Stage) DCC Pressure Drop	DPT-34
Warm Seawater Supply Temperature	TE-10
Warm Seawater Discharge Temperature	TE-12
Cold Seawater Supply Temperature	TE-20
Cold Seawater Discharge Temperature	TE-22
Cold Water Temperature Out of DCC 1st Stage	TE-37,30 &32
Cold Water Temperature Out of DCC 2nd Stage	TE-31
Turbine Rotational Speed	SI-41
Generator Power Output	WT-40
Essential Loads Power Consumption	WT-42
Non-Essential Loads Power Consumption	WT-43
Net Power : (Generator—Essentials)	WT-44

A tabulation of representative time history records, typical of November conditions at the site, is given in Appendix A. The average of these records are used in this report to estimate performance parameters. The uncertainty of derived parameters is discussed in Appendix B. Time history records are routinely recorded for future use and on-line display and analysis. The nominal sampling frequency is set at 1 Hz (sample/second). The record length and average interval for display purposed is specified by the operator. However, one-hour averages of all measured and derived parameters are routinely displayed along with the record standard deviation, and maximum and minimum values. A minimum of one hour record length is required for every subsystem test.

The records given in Appendix A give the average value of the control parameters resulting in the indicated net power output. The gross power as well as the power consumption are also given. The average control parameters for the 60 minutes long record considered here are :

- Warm water 8419 gpm at 26.33 °C
- Cold water into DCC 5400 gpm at 5.67 °C
- Inlet pressure into Vacuum Pumps 1293 Pa

resulting in

- net power output of 54.8 kW

Evaporator Subsystem Performance Parameters

The performance of the evaporator subsystem is quantified by estimating the value of the thermal effectiveness (ϵ_{evap}) and steam generation rate (m_{stm}).

Evaporator Thermal Effectiveness

Thermal effectiveness of the evaporator, ϵ_{evap} , is defined as

$$\epsilon_{\text{evap}} = (T_{\text{wwi}} - T_{\text{wwo}}) / (T_{\text{wwi}} - T_{\text{satsw}})$$

where

T_{wwi} = warm-water inlet temperature ($^{\circ}\text{C}$),
measured by TE-10 (see Figure 1)

T_{wwo} = warm-water outlet (discharge) temperature ($^{\circ}\text{C}$),
measured by TE-12

T_{satsw} = seawater temperature in equilibrium with the steam generated at the evaporator ($^{\circ}\text{C}$). Saturation temperature relative to seawater is about 0.31°C higher than the saturation temperature relative to fresh water (T_{satfw}) at the nominal evaporator pressure

T_{satfw} is estimated from the steam tables at the vapor pressure (for 100% quality steam), measured with the pressure sensor PT-32 at the turbine inlet. T_{satsw} has been determined to be given by:

$$T_{\text{satsw}} = T_{\text{satfw}} + 0.31$$

for the range of evaporator pressures measured.

The effectiveness of the evaporator subsystem has been previously estimated to be in the range of 0.80 to 0.95 (e.g., see References). This was determined with a pressure transducer installed in the evaporator chamber that was not available at the time of the life cycle test requested by DOE (September-November 1996). The evaporator instrumentation schematic corresponding to the life-cycle test reported here is given in Figure 1. The value corresponding to the parameters given in Appendix A is 0.8. This is considered to be a lower limit because the pressure measurement was performed downstream of the mist eliminator. The pressure in the evaporator chamber should be approximately 10 Pa higher than PT-32.

Steam Generation Rate

Since the evaporator outlet seawater flow is approximately 99.5% of the inlet flow, standard flow measurement devices are not accurate enough to determine steam flow from the seawater flow difference. Neither is it feasible to measure the steam flow directly since gas flow-meters are not reliable at the low steam density encountered under open-cycle OTEC conditions.

It was determined that the most accurate method of determining steam generation rate out of the evaporator is balancing the heat absorbed from the seawater with the heat of evaporation of steam:

$$\dot{m}_{\text{stm}} = \dot{m}_{\text{wwi}} c_{\text{psw}} (T_{\text{wwi}} - T_{\text{wwo}}) / h_{\text{fg}}$$

where

c_{psw} = specific heat of seawater (kJ/kg $^{\circ}\text{C}$),

h_{fg} = heat of evaporation of steam (kJ/kg)

the other parameters are defined above.

For the evaporator pressure and temperature and the warm water flow rate and temperatures given in Appendix A, the specific heat of seawater is 4 kJ/kg $^{\circ}\text{C}$; the heat of evaporation of steam 2,451.6 kJ/kg resulting in an estimate of 3.41 kg/s for the steam generation rate.

Turbine-Diffuser-Generator Subsystem Performance Parameters

The performance of the turbine subsystem is quantified by measuring the power produced from the generator PW_{TG} , and the turbine-diffuser exit pressure (i.e., back pressure). These values are 191.6 kW (WT-40) and 1362 Pa (PT-33) respectively from the records under discussion. The turbine-generator instrumentation schematic corresponding to the life-cycle test reported here is given in Figure 2.

Turbine Isentropic Power: Isentropic power from the steam turbine is computed based on pressure measurements at the turbine inlet and outlet, and steam flow given above.

For the records given in Appendix A;

$$h_1 = h_{g1} \text{ (enthalpy of dry saturated steam at turbine inlet pressure kJ/kg)}$$

$$h_1 = 2540.3 \text{ kJ/kg}$$

$$s_1 = s_{g1} \text{ (entropy of dry saturated steam at turbine inlet pressure kJ/kg}^\circ\text{C)}$$

$$s_1 = s_{g1} = 8.6426 \text{ kJ/kg}^\circ\text{C}$$

$$x_2 = (s_2 - s_{f2}) / s_{fg2} \text{ (quality of steam at turbine outlet)}$$

where subscripts 1 and 2 indicate conditions at the inlet and outlet of the turbine-diffuser subsystem. Under isentropic expansion ($s_2 = s_1$) at the diffuser outlet the quality and enthalpy of steam are:

$$X_{2ise} = (s_1 - s_{f2}) / s_{fg2}$$

$$h_{2ise} = h_{f2} + x_{2ise} \times h_{fg2}$$

from the ASME steam tables:

$$s_{f2} = 0.1740 \text{ kJ/kg}^\circ\text{C}$$

$$s_{fg2} = 8.6902 \text{ kJ/kg}^\circ\text{C}$$

$$h_{f2} = 48.53 \text{ kJ/kg}$$

$$h_{fg2} = 2474.2 \text{ kJ/kg}$$

resulting in

$$X_{2ise} = 0.9745$$

$$h_{2ise} = 2459.7 \text{ kJ/kg}$$

yielding an isentropic enthalpy drop (or isentropic expansion specific work) of

$$\Delta h_{ise} = h_1 - h_{2ise}$$

$$\Delta h_{ise} = 2540.3 - 2459.7 = 80.6 \text{ kJ/kg}$$

such that the isentropic power ($= m_{stm} \times \Delta h_{ise}$) is:

$$\text{Isentropic power} = 3.41 \times 80.6 = 274.9 \text{ kW (kJ/s)}$$

Gross Power from Turbine-Generator System: The average gross electrical output from the generator is 191.6 kW (Appendix A).

Turbine-Generator System Efficiency: The ratio of the gross generator output (WT 40) to the isentropic power yields the combined efficiency of the turbine-diffuser-generator system.

$$\eta_{TG} = PW_{TG} / \text{isentropic power}$$

Under conditions encountered during this phase of testing the efficiency is estimated at 70% ($= 191.6/274.9$). The efficiency of the turbine can be estimated, by accounting for

the mechanical losses and generator efficiency quoted at 15 kW and 90.7% respectively by the manufacturer, as 82% $(=(191.6/0.907 + 15)/274.9)$.

Direct Contact Condenser Subsystem Performance Parameters

The performance of the direct-contact-condenser (DCC) subsystem is quantified by estimating the value of the overall fraction of steam condensed (F), steam side pressure drop ($\Delta P_{s_{DCC}}$), the thermal effectiveness (ϵ_{DCC}), and the vacuum vent ratio (V). The DCC instrumentation schematic corresponding to the life-cycle test reported here is given in Figure 3.

Fraction of Steam Condensed

The overall fraction of steam condensed (F) is estimated by equating the heat transferred to the cold seawater in the DCC, $m_{cwi} c_{psw} (T_{cwo} - T_{cwi})$, to the energy absorbed from the steam entering the DCC, $X m_{stm} h_{fg}$; therefore, the fraction of steam condensed in the DCC is given by:

$$F = (m_{cwi} c_{psw} (T_{cwo} - T_{cwi}) / (X m_{stm} h_{fg}))$$

For the parameters given in Appendix A the overall fraction of steam condensed is estimated at 0.98.

DCC Steam Side Pressure Drop

The vapor pressure loss is important since it has an impact on the available driving potential for condensation and the inlet pressure seen by the vacuum compressor. The average vapor pressure loss for the co-current stage was 16 Pa as given by DPT-33 in Appendix A, and for the counter-current stage 60 Pa as given by DPT-34.

DCC Thermal Effectiveness

The thermal effectiveness is defined as the ratio of the actual heat-transfer rate to the maximum possible heat-transfer rate. This parameter varies from zero to one with a value of one indicating a perfect device, such as an infinitely long counter-current heat exchanger. For a condenser with non-condensable gases in the vapor, the maximum possible heat transfer is attained when the water outlet temperature is equal to the steam inlet temperature. Assuming a constant liquid specific heat and low condensate to coolant flow ratios, the effectiveness can be defined as :

$$\epsilon_{DCC} = (T_{cwo} - T_{cwi}) / (T_{satsw} - T_{cwi})$$

where

T_{satsw} = seawater temperature ($^{\circ}\text{C}$) that is in equilibrium with the steam entering each stage. For the expected range of condenser pressure, this temperature is 0.28°C higher than the freshwater saturation temperature.

The saturation temperature is obtained from the ASME Steam Tables for the pressure at the inlet of the DCC (11.56 °C for a pressure of 1362 Pa in Appendix A)

T_{cwi} = cold-water inlet (supply) temperature (°C), measured by the temperature element TE-20,

T_{cwo} = temperature (°C) of the seawater exiting the condenser stages. TE-22 for the overall effectiveness

The DCC average overall thermal effectiveness for the typical data given in Appendix A is 0.94.

Vacuum Vent Ratio

When non-condensable gases are introduced into the system by leaks and by desorption from the process water, a venting system must be used to remove them and maintain the operating pressure. Under these conditions, a quantity of steam must also be exhausted with the non-condensable gas. The vent ratio compares the actual condenser outlet volumetric flow rate to the ideal outlet volumetric flow rate. The ideal (smallest) volumetric flow rate requires both a condenser with zero pressure losses and one in which the steam reaches the minimum possible partial pressure (the saturation pressure at the inlet water temperature). The vent ration can be expressed as:

$$V = \frac{[P_{i,1} - \Sigma\Delta P - P_{sat}(T_{so,2})] (T_{cwi,2} + 273.15)}{[P_{i,1} - P_{sat}(T_{cwi,2})] (T_{so,2} + 273.15)}$$

where (values given within parenthesis are from Appendix A):

$P_{i,1}$ = pressure of vapor (steam and non-condensables) entering the co-current condenser stage, measured by PT-33 (= 1362 Pa)

$\Sigma\Delta P$ = total vapor pressure loss, measured by the sum of DPT-33 and DPT-34 (= 76 Pa)

$T_{so,2}$ = outlet steam temperature from the counter-current stage, estimated to be equal to the seawater inlet temperature

$P_{sat}(T_{so,2})$ = saturation pressure of the steam exiting the counter-current stage with the outlet temperature $T_{so,2}$. It can be calculated from the Steam Tables as 913 Pa for a temperature of 5.67°C [The pressure

at the exit of the DCC 1286 PA (=1362-76) is that of the gas, leaving the DCC, containing non-condensable in addition to the steam,

$T_{cwi,2}$ = inlet water temperature for the counter-current stage. It is equal to the temperature of the cold water supply (5.67°C), measured by TE-20

$P_{sat}(T_{cwi,2})$ = saturation pressure of the steam in equilibrium with the cold water supply to the counter-current stage $T_{cwi,2}$ (= 5.67 °C). It is estimated to be 913 Pa from the Steam Tables.

The vent ratio is less than 1.0 in a practical condenser because of either one or both of the following; (1) $T_{so,2}$ being slightly higher than $T_{cwi,2}$; or, (2) a drop in the total system pressure (vapor + non-condensable) between the inlet and outlet of the overall DCC, i.e., $\Sigma\Delta P \neq 0$. Under the typical conditions given in Appendix A the vent ratio is estimated at 83%.

Vacuum Compressor Subsystem Performance Parameters

The performance of the compressor subsystem was quantified by determining the overall power consumption of the compressors (PW_{nc}). In the early stages of the operational phase of the project, before the failure of stage 4 in 1994, the overall power consumption was determined (with the watt-hour meter used to estimate the total power consumed by the vacuum pumps) to be no more than 35 kW under design conditions. Unfortunately without stage 4 it is necessary to operate the plant with an additional (inefficient) positive displacement compressor resulting in a net increase of power consumption of approximately 20 kW for a total consumption of approximately 55 kW, or 38% higher than the DOE goal of ≤ 40 kW. However, given that the third stage vacuum pump refurbished with magnetic bearings does not consume more power than those using grease lubricated bearings, the projected power consumption for a system including a new stage 4 with magnetic bearings would be estimated at 35 kW under design conditions. What was previously missing was the indication of longer life before failure with the high speed centrifugal compressors. The accelerated life test, discussed in the following Section, indicates that centrifugal compressor using magnetic bearings ought to provide a cost effective solution for OC-OTEC systems.

Compressor parameters like pressure ratio, speed and current, for a given measured mass flow rate of noncondensables, are given in Appendix A. The vacuum compressor subsystem instrumentation schematic corresponding to the life-cycle test reported here is given in Figure 4.

Section II: Vacuum Pumps Life Cycle Data

A major problem with the design of the OC-OTEC experimental facility was due, in part, to the emphasis on power consumption (equipment performance) over reliability. The vacuum compression pumps deliver the required aerodynamic performance. Designers and manufacturers of high speed spindles know that the high speed centrifugal vacuum pumps should not have been designed with grease lubricated bearings. Although, during the design phase, excessive bearing temperatures were predicted under operational conditions, this was not entered into the bearing design nor were adequate instruments specified to monitor conditions in and around the bearings. A large effort in the field resulted as complete instrument systems were installed in order to analyze new bearing configurations and predict performance.

The vacuum compression train is not a standard commercial item. The first four stages, designed and manufactured by Barber-Nichols, are high speed centrifugal compressors operating at speeds ranging from about 25,800 rpm (1st stage) to 42,800 rpm (4th stage). To avoid the need for a shaft seal, the electric motor operates in the same vacuum environment as the impeller. The fifth and final stage is a helical screw-type positive displacement unit made by Quincy. A more powerful Quincy unit, called the roughing compressor, was originally intended to be used only for initial vacuum pump down, but can now be operated in that mode or connected in parallel with the 5th stage to provide more pumping capacity during normal operation. For various reasons, including the ability to fine tune system performance to minimize power consumption, the five stages are powered by variable frequency drives (VFD) which allow the compressors to be operated over a broad speed range.

The unreliability of the centrifugal compressors has been the greatest operational problem to date. The failure of these units resulted from the inability to properly cool the bearings. The factors which compound to cause this problem were the use of grease lubricated ball bearings, together with high operating temperatures resulting from the motor and bearings operating in the same vacuum environment as the impeller.

Analysis and field testing indicated that the compressor design was deficient by not accounting for the reduction in the internal-diametral-clearance of the bearing races induced by the relatively high temperature differential ($\approx 40^\circ\text{C}$) between the inner and outer bearing races. One solution for units operating at speeds below 27,000 rpm has been determined to be to exchange the ceramic balls with smaller balls to accommodate the differential thermal expansion of the bearing cases and the use of a grease which has improved high temperature performance. This redesigned bearing system has resulted in our ability to operate the vacuum pumps for approximately 2,000 hours instead of tens of hours, as in the beginning stages of the project. (See Tables 1A, 1B, 1C and 2). In the case of the higher speed units ($> 27,000$ rpm) we selected a magnetic bearing system for the third

stage and an air bearing system for the fourth stage. To date, the pump refurbished with magnetic bearings has been operated for 535 hours without problems. (See Table 3) It is expected that magnetic bearings will provide the required thousands of hours of operation. Unfortunately, the air bearing system installed in stage 4 failed to operate (Table 4).

Third Stage Vacuum Compressor Refurbishment

The third stage compressor is a 5 kW integral motor driven centrifugal compressor, vertically mounted, with a maximum operating speed of 38,500 rpm. Five stages of compression are used to attain the required vacuum. Four of these stages use single stage centrifugal compressors. The fifth is a positive displacement compressor. This letter report concentrates on the third stage compressor that was experiencing bearing reliability problems. The original bearing configuration consisted of spring loaded precision angular contact bearings mounted on each end of the motor rotor, with an overhung impeller. The intent was to convert the compressor to magnetic bearings.

As a first step in the design process in applying magnetic bearings, an estimation of the forces on the rotor is required, both in the radial and axial direction. This is not an easy process, since the forces on a rotor are not generally well known. Static, dynamic, and process fluid forces must all be considered. The dynamic forces can originate from a variety of sources including imbalance, external disturbances, or in this case electrical imbalance from the motor. If the forces are known the bearings can be sized such that they have the capacity to hold the shaft rigidly in place with all these forces combined. In the case of this compressor, these forces were not known well enough to accurately be predicted, and therefore, a conservative estimate was used.

Once this estimation of the force requirements was made by PICHTR, the bearing and rotor layout was undertaken by Revolve. A conical bearing design was chosen to eliminate the need for a thrust bearing, and a tie bolt was used to hold the bearing and motor rotors together as an assembly. The impeller was held onto the tie bolt independently. With this layout completed, the rotor-dynamic analysis was initiated.

As a baseline, information on the free-free modes and frequencies was provided by the compressor OEM. Since the majority of the mass on the rotor was due to the motor rotor and the impeller, it was important to get a good approximation of the effective stiffness diameter of each component. A model of the rotor was generated by Revolve, using an in-house code, and the relative contribution of stiffness and mass of the motor rotor and impeller were adjusted until good correlation existed between the model and the measured modes.

The next step was to model the rotor with magnetic bearings. The initial design used conical bearings of equal capacity. A shoulder on the tie bolt was located between the

impeller and impeller end bearing. This allowed the bearings and motor to be stacked up together as one assembly, and the impeller mounted independently. Prediction of the bending modes of the shaft was a difficult task, since the rotor essentially consisted of three complex laminated sleeves held in compression with the tie bolt, and an overhung impeller.

In order to get an accurate estimate of each component's contribution to stiffness, an initial set of bearing rotors were built, assembled on the shaft and tested to determine the undamped natural frequencies and mode shapes. The rotor-dynamic model was adjusted to correlate with the measured frequencies, specifically by adjusting the effective stiffness diameter of each laminated section. It must be noted that great care was taken to ensure good shoulder to shoulder fits between each piece and the tie bolt. Otherwise, the effective stiffness diameter of each piece would be reduced and the corresponding overall shaft stiffness reduced.

Once the model adjustments were completed, a full analysis was undertaken incorporating the active magnetic bearing control system effects. Given the speed of rotation and the relatively large size of the impeller to the shaft, gyroscopic effects were significant. Although our ring tests indicated the first bending mode was at the lower end of the operating range (around 530 Hz), our model predicted the gyroscopic stiffening effect would raise the bending mode frequency to the high end of the operating range (around 600 Hz for the forward whirl mode). It also indicated this mode was poorly damped, meaning operation close to the critical would likely not be possible.

The next step was to verify the model predictions by confirming the location of the mode at speed. These tests confirmed that the mode was in fact where we had predicted, and was poorly damped. This mode would have to be raised in order to attain the desired operating speed range.

Analysis of the mode shapes identified that the location on the rotor where changes in geometry would have the greatest impact was on the impeller and bearing. The design of the rotor was modified until the first bending mode was moved sufficiently out of the operating range.

Several changes were made to the bearing rotor to increase its stiffness contribution. The first was to maximize the lamination inner diameters under the bearing and sensor. This allowed reducing the cone angle of the bearing to reduce its axial force by the rotor weight to match the axial capacity of the upper bearing. Finally, the end ring on the rotor was redesigned to maximize the diameter at which contact was made with the tie bolt shoulder.

The rotor was then reassembled and a ring test performed. Unexpectedly, the bending mode frequency actually decreased. The analysis identified a manufacturing problem, resulting from improper compression of the laminations as the cause.

The bearing rotor was then rebuilt, assembled on the compressor rotor and re-tested. The ring test now showed that the rotor behaved as predicted with these design changes. With the new design change, full speed was attained without exciting the forward whirl mode. Small amplitude swept sine tests were conducted as a final test to locate the critical at speed to confirm the model once more, and verify a reasonable stability margin existed. The tests by the manufacturer were successful, and installed at the OC-OTEC facility. To date, the third stage pump has been operational for 535 hours without incident. (See Table 3) Magnetic bearings are the preferable choice for OC-OTEC centrifugal vacuum pumps.

Conclusions

As previously reported (see References), the 210 kW OC-OTEC Experimental Apparatus has provided valuable data and pointed the way for future modifications and improvements in the OC-OTEC process. The data demonstrate that the OTEC process is technically feasible for the production of base load electricity and desalinated water. Data records obtained since the initiation of operations in 1993 have been previously used to confirm the predicted performance of the experimental OC-OTEC Apparatus.

The most annoying problem has been the frequent failures of the grease lubricated bearings of the centrifugal pumps used for the vacuum and exhaust system. In retrospect this was due to a major design oversight. Equipment operating at speeds higher than approximately 27,000 rpm should, in general, not use grease lubricated bearings. Magnetic bearings like the ones described here are the preferable choice.

Other significant lessons, previously reported, learned (or relearned) and observations from the perspective of an operator of the OTEC experimental facility are:

- Specifications should be written to emphasize the particulars of the job excluding "boiler plate" information.
- Make the facility "user friendly" from the standpoint of troubleshooting, maintenance, repair and modification
- Include technical field support from suppliers of major equipment, but be prepared to solve most problems on your own.
- Select equipment with excess capacity. It is appropriate to optimize design point performance but there will always be off-design operations requiring additional capacity.
- Mechanical equipment specifically designed for OTEC must be instrumented to measure temperatures and pressures in as many locations as possible. For example, measurements performed with sensors installed, in the field, to estimate temperatures around the bearings

of the high speed centrifugal pumps revealed that they were failing because of two main causes: (i) deterioration of the bearing's lubricant grease due to high temperatures; and/or (ii) differential expansion of the outer and inner rings resulting in squeezing of the ball bearings.

- If equipment has moving parts evaluate the bearing system and ask potential supplier to provide references of successful application of their design before purchase.
- Consider the corrosive saltwater, condensate, and the typically harsh environment of OTEC sites when making design decisions, especially material selection and placement of mechanical and electrical equipment.
- Concrete is an excellent material for the vacuum structures required for OC-OTEC.
- Avoid metal components, but if unavoidable use the hot-dip-galvanized process from a factory with proven quality. We found that the quality control of most suppliers is poor.
- Fresh Water, instead of seawater, should be used as the Coolant for the intercoolers used with the vacuum compressors.

References

D. E. Evans, Operational Plan for the 210 kW OC-OTEC Experimental Apparatus, July 1992. Prepared for the National Renewable Energy Laboratory under Contract No. XC-0-10077-1.

Pacific International Center for High Technology Research, Final Design of the Open-Cycle OTEC Net Power Producing Experiment, November 1991. Prepared for the National Renewable Energy Laboratory under Contract No. XC-0-10077-1.

Idem, As-Built Conditions of the 210 kW OC-OTEC Experimental Apparatus, December 1992. Prepared for the US Department of Energy under Contract No. DE-AC02-92CH-10539.

Idem, 210 kW OC-OTEC Experimental Apparatus: Significant Lessons Learned, July 1994. Prepared for the US Department of Energy under Contract No. DE-AC02-92CH-10539.

Idem, 210 kW OC-OTEC Experimental Apparatus: Operations from January 1994 to July 1995 (DOE, FY 94), October 1995. Prepared for the US Department of Energy under Contract No. DE-AC02-92CH-10539.

L. A. Vega and D. E. Evans, Operation of a Small Open-Cycle OTEC Experimental Facility, in *Proceedings of Oceanology International 94*, Vol. 5, Brighton, UK, March 1994.

L. A. Vega, The 210 kW Open Cycle OTEC Experimental Apparatus: Status Report, in *Proceedings of Ocean 95 Conference*, San Diego, California, October 1995.

CENTRIFUGAL COMPRESSORS SUMMARY							
STAGE	1 A						
		Total Run	Upper Bearing		Lower Bearing		MOTOR
DATE	Remarks	Time	Type	Grease	Type	Grease	
7/15/93	Original	166 hrs	Fafnir Ceramic	Andok	Same as upper	Same	Original
	B-N setup	(meter +61)	Chrome Race	260			
9/11/93	Replaced Both Bearings	197 hrs	Barden Ceramic	Andok	Barden	Andok	
	LB failed visual inspection		C-105H	260	C-105H	260	
			Phenolic Race		Phenolic Race		
10/12/93	Replaced Upper Bearing	229 hrs	Failed after				
	failed visual inspection.		32 hours				
			Barden	Kluber	Unchanged		
			C-105H				
5/29/94	UB temp. increased	670 hrs	Failed after		Unchanged		Stator Moved
	from 165 °F to 209 °F		441 hours				to 1C, failed
	in 3 hours. Grease discolored.						after 700 hrs
							with 1C
6/4/94	Installed Reballled Bearings		Reballled	Kluber	Unchanged		"Other"
	Lubricated by		BARDEN	NCA-15			Stator
	Precise, Mr. Zenker		C-105 HX118				
7/27/94	Removed unit for		Unchanged		Unchanged		
	to Inspection & Decided to						
8/3/94	Rebalance Impeller						
	Reinstalled with same Brgs						

TABLE 1A. STAGE 1A CENTRIFUGAL COMPRESSOR (Continued)

DATE	Remarks	Total Run Time	Upper Bearing		Lower Bearing		MOTOR
			Type	Grease	Type	Grease	
9/13/96	High Current Stop	2128 hrs	Reballed replaced		Standard Barden		
	Remove & Inspect		after 1458 hrs		failed after		
	LB failed		(2128-670)		1931 hrs		
	UB grease was black decided				(2128-197)		
	to replace						
	Both Bearings Replaced		Barden C-105HJH		Zenker	Kluber	
	for 24 hrs/day test		Steel Balls/ Bronze Cage		C-105HX118	NCA-15	
12/3/96	Lower Bearing failed after	2268 hrs					
	140 hrs due to rotor unbalance						
	(operator error)						
• FROM JANUARY 1993 TO MAY 1994 WE EXPERIMENTED WITH DIFFERENT BEARING BRANDS AND GREASES;							
• SINCE JUNE 1994 WE HAVE BEEN USING REBALLED BEARINGS (SMALLER BALLS, MORE CLERANCE);							
• UPPER LOCATION WITH STANDARD BEARINGS ONE FAILURE AFTER 441 HOURS; WITH REBALLED BEARINGS 1458 HRS							
LOWER LOCATION WITH STANDARD BEARINGS 1931 HRS TO FAILURE.							
- STANDARD/DESIGN BEARINGS: BARDEN C-105H (Ceramic Balls) WITH ANDOK 260 GREASE.							
- REBALLED (ZENKER) BEARINGS: BARDEN C-105HX118 (Ceramic Balls) WITH KLUBER NCA15 GREASE.							

TABLE 1A. STAGE 1A CENTRIFUGAL COMPRESSOR

CENTRIFUGAL COMPRESSORS SUMMARY							
STAGE	1B						
		Total Run	Upper Bearing		Lower Bearing		MOTOR
DATE	Remarks	Time	Type	Grease	Type	Grease	
As of 7/14/93 there have been numerous bearing failures due to original shape of shaft. This was a MANUFACTURING PROBLEM. Go to 3/10/94 for first entry corresponding to unit with straight shaft.							
	New shaft	Set @0 hrs	C105HX118	Kluber	C105HX119	Kluber	
	Nilos grease shields		Reballed Barden				
	Rewound Motor (Hilo)		Ceramic				
3/10/94		20 hrs					
5/27/94	Operational Data						
	Unit/ T °F /ΔT °F/Amps						
	1A 156°F/na/5 amps						
	1B 131°F/28°F/3.9 amps						
	1C 145°F/43°F/4.7 amps						
5/29/94	Operational Data	286 hrs	266 hours without failures with reballed bearings				
7/21/94	"	711 hrs	691 hours without bearing failures				
11/9/95	Inverter fault.	1500 hrs					
	Rewind stator						
	Unit set aside as spare						
	Operated with 1A and 1C only						
6/19/96	Reinstalled Unit		Same as above		Same as above		
9/11/96	Lower Bearing Failure	1539 hrs		Reballed failed after 1519 hrs			

TABLE 1B. STAGE 1B CENTRIFUGAL COMPRESSOR (Continued)

		Total Run	Upper Bearing		Lower Bearing		MOTOR
DATE	Remarks	Time	Type	Grease	Type	Grease	
11/16/96	Reinstalled		105HJH	Kluber	105HJH	Kluber	
			Steel Balls				
			Bronze Cage				
• LOWER LOCATION WITH REBALLED (BARDEN CERAMIC C-105HX118 FROM ZENKER) BEARINGS ONE FAILURE							
AFTER 1519 HOURS							

TABLE 1B. STAGE 1B CENTRIFUGAL COMPRESSOR

CENTRIFUGAL COMPRESSORS SUMMARY							
STAGE	1C						
		Total Run	Upper Bearing		Lower Bearing		MOTOR
DATE	Remarks	Time	Type	Grease	Type	Grease	
7/14/93	Original Ceramic Brgs.	125 hrs	Fafnir Steel	Andok	Fafnir Steel	Andok	Original
	Replaced with Steel Balls			260		260	
	to compare with 1A	no failures					
10/8/93	Replaced Upper Bearing	152 hrs	Barden Steel	Kluber	Unchanged		
	after 27 hrs			NBU 15			
4/5/94	After 207 hrs. UB grease	332 hrs	Reballed	Kluber			
	deteriorating		BARDEN	NCA-15			
	Decided to Replace both with		C-105 HX118				
	Reballed Barden/Zenker		one Nillos Shield				
5/28/94	Motor Failure after	544 hrs	Unchanged		Unchanged		Stator from 1A
	Inverter Fault.						moved here
	Replaced with Stator						failed after 700 hrs
	from 1A						with 1C
9/24/96	VFD Failure after 1853 hrs		Unchanged		Unchanged		
	Unable to operate Stage 1						
11/15/96	High Current	1990 hrs			LB grease gone		
	Manual Trip				Inner race metal-blue color		
					FAILURE after 1865 hrs (1990-125)		
• LOWER LOCATION WITH REBALLED BEARINGS 1865 HRS TO FAILURE.							
- REBALLED (ZENKER) BEARINGS: BARDEN C-105HX118 (Ceramic Balls) WITH KLUBER NCA15 GREASE.							

TABLE 1C. STAGE 1C CENTRIFUGAL COMPRESSOR

CENTRIFUGAL COMPRESSORS SUMMARY							
STAGE	2						
		Total Run	Upper Bearing		Lower Bearing		MOTOR
DATE	Remarks	Time	Type	Grease	Type	Grease	
Through 9/93 there were several motor failures (1 original from Colorado plus 3 rewound in Hilo) due to UB Failures approximately every 20 hrs. The shaft was ruined (bent) and send to Colorado (B-N) for repairs.							
Apparently the shaft was the problem.							
9/29/93	Unit with Barden Bearings	112 hrs	Barden	Andok	Barden	Andok	
			C 105H	260	C 105H	260	
5/29/94	Operational Data	581 hrs	469 hrs (581-112) without bearing failure				
	No failure						
6/13/94	Standard Bearings	717 hrs	Barden	Krytox	Barden	Krytox	
	Failed after 605 hrs		C-105 HX118		C-105 HX118		
	Replaced with Reballled Bearings and Krytox Grease		by us				
7/21/94	Bearing Failure after	793 hrs	Barden		Barden		
	76 hrs		C-105 HX118		C-105 HX118		
			by us				
6/16/95	Bearing Failure after	1373 hrs	Reballled	Kluber	Reballled	Kluber	
	580 hrs		Barden	NCA-15	Barden	NCA-15	
	This time replaced		C-105 HX118		C-105 HX118		
	with Reballled Bearings		Precise/Zenker		Precise/Zenker		
	Greased by Precise/Zenker						
	Nylos Shield						
11/24/96	Ongoing 1092 hrs.	2465 hrs					
	with Zenker Bearings						

TABLE 2. STAGE 2 CENTRIFUGAL COMPRESSOR

CENTRIFUGAL COMPRESSORS SUMMARY						
STAGE	3					
		Total Run	Upper Bearing		Lower Bearing	
DATE	Remarks	Time	Type	Grease	Type	Grease
As of July 1993 we experimented with several bearings (104s) and greases and were never able to operate for more than $\approx$ 180 hrs (e.g., Barden C-104 h with Andok 260 grease).						
The stator was rewound several times due to bearing failure and/or arcing.						
9/27/93	Installed sensors to measure temperatures and ΔT across balls. ΔT of 125 °F measured with standard brgs					
3/16/94	Installed Reballled Brgs.		Reballled Barden	Kluber	Reballled Barden	Kluber
	Greased by Precise/Zenker.		C104 118	NCA-15	C104 118	NCA-15
4/15/94	Motor Stator fault afetr 260 hrs.					
5/24/94	Upper Bearing overheat after only 159 hrs		Replaced with Reballled		Unchanged	
5/26/94	Stator Phase to Phase short. After 68 hrs since last rewind.					
7/24/94	High vibrations and Temp. Replaced both		Failed after 362 hrs		Failed after 521 hrs	
After the July 94 failure with the reballled bearings it was concluded that the bearing system had to be refurbished.						
Dec-94	Unit shipped to Revolve Technologies Inc. to be refurbished with Magnetic Bearings.					
May-96	Refurbished Unit Reinstalled.					
11/24/96	535 hours of operation with Magnetic Bearings.					

TABLE 3. STAGE 3 CENTRIFUGAL COMPRESSOR

CENTRIFUGAL COMPRESSORS SUMMARY							
STAGE	4						
		Total Run	Upper Bearing		Lower Bearing		MOTOR
DATE	Remarks	Time	Type	Grease	Type	Grease	
Through 7/93 there were three bearing failures and 1 motor failure.							
7/14/93	Status		Fafnir Steel	Kluber	Fafnir Ceramic	Kluber	
9/23/93	After Failure		Barden 104	Andok	Barden 104	Andok	
	Replaced with Barden		Steel Balls	260	Steel Balls	260	
	Balanced at Field Office						
10/1/93	Upper Bearing Failure		Barden 104	Texaco	Unchanged		
	After 23 hrs		Steel Balls	Starflex			
10/6/93	UB failure after 4 hrs		Fafnir	Kluber	Unchanged		
	Grease is not the solution		Steel				
1/18/94	Lower Bearing Failure	Shaft bent sent to B-N for repairs					
	After 110 hrs						
3/11/94	Installed Reballled Bearings		Barden 104		Barden 104		
	to But only achieved ~30 hrs		Reballled		Reballled		
3/31/94	due to poor installation by us		by us		by us		
4/6/94	This time Reballled Brgs.		Reballled	Kluber	Reballled	Kluber	
	Greased by Precise/Zenker		Barden	NCA-15	Barden	NCA-15	
			C-104		C-104		
6/7/94	LB Failure after 260 hrs with Reballled Bearings						
	Concluded that grease bearings can not be used for the third and fourth stages						
Apr-95	The unit refurbished with air bearings failed. Air bearings were not the solution.						

TABLE 4. STAGE 4 CENTRIFUGAL COMPRESSOR

Figure 1.- EVAPORATOR INSTRUMENTATION SCHEMATIC

Figure 2.- TURBINE-GENERATOR INSTRUMENTATION SCHEMATIC

Figure 3.- DIRECT-CONTACT CONDENSER INSTRUMENTATION SCHEMATIC

Figure 4.- VACUUM COMPRESSOR SUBSYSTEM INSTRUMENTATION
Used during the life-cycle tests

(IC: Intercooler)

- WT-40: Generator Power Output (gross power)
- WT-42: Essential Loads (Seawater pumping, vacuum compression and essential auxiliaries)
- WT-44: $\text{Net Power} = (\text{WT-40}) - (\text{WT-42})$

Figure 5.- OC-OTEC SYSTEM SCHEMATIC
Used during the life-cycle test

Appendix A

Representative Time History Records

The SI system of units is used here unless otherwise noted. For example, pressures and temperatures are given in Pa and °C respectively, but water volumetric flowrates in gallons-per-minute (gpm). All power measurements are given in kW and currents in amperes. Rotational speeds are given in revolutions-per-minute (rpm) and the mass flowrate of noncondensables in g/s.

Table of Contents

Control/Output Parameters3

Evaporator/Turbine Measured Parameters4

DCC Measured Parameters5

Power Production (kW)6

Vacuum Compressor Subsystem Parameters7

Vacuum Compressor Subsystem Parameters8

Vacuum Pump Life Test for the 210 kW OC-OTEC Experimental Plant

Appendix A

CONTROL/OUTPUT PARAMETERS

Time History Record (60 minutes long) Typical Nov. Conditions						
Time	Q _{ww}	Q _{cw} dcc	T _{ww} in	T _{cw} in	P _{inv} c	P _W net
	gpm	gpm	deg C	deg C	Pa	kW
10:20	8345	5443	26.28	5.67	1299	55.4
10:21	8400	5460	26.28	5.65	1299	55
10:22	8416	5449	26.27	5.64	1299	55.4
10:23	8354	5425	26.29	5.66	1301	56.3
10:24	8499	5431	26.29	5.64	1300	55.4
10:25	8502	5439	26.31	5.64	1298	55.5
10:26	8339	5442	26.29	5.65	1300	54.2
10:27	8517	5471	26.29	5.65	1300	55.4
10:28	8374	5427	26.3	5.66	1298	55.9
10:29	8406	5474	26.31	5.65	1301	55.5
10:30	8414	5466	26.31	5.65	1300	55.3
10:31	8285	5442	26.31	5.66	1300	51.9
10:32	8337	5407	26.29	5.66	1302	51
10:33	8388	5432	26.3	5.66	1297	56.5
10:34	8565	5389	26.32	5.65	1300	55.6
10:35	8456	5474	26.31	5.63	1298	55.5
10:36	8420	5314	26.3	5.62	1301	55.4
10:37	8559	5390	26.32	5.6	1297	57
10:38	8443	5391	26.32	5.61	1299	57
10:39	8396	5302	26.35	5.62	1296	56.2
10:40	8448	5288	26.37	5.62	1293	56
10:41	8457	5363	26.37	5.62	1292	56.5
10:42	8450	5429	26.34	5.63	1293	55.7
10:43	8322	5393	26.36	5.65	1291	55.4
10:44	8405	5354	26.36	5.66	1293	55.7
10:45	8559	5500	26.35	5.65	1291	55.1
10:46	8408	5492	26.35	5.66	1293	56.4
10:47	8492	5376	26.34	5.67	1293	55.4
10:48	8328	5394	26.36	5.69	1289	55.2
10:49	8399	5434	26.36	5.68	1292	54.3
10:50	8343	5361	26.34	5.69	1290	54.8
10:51	8461	5457	26.33	5.66	1293	50.5
10:52	8415	5395	26.33	5.67	1293	50.2
10:53	8501	5384	26.33	5.69	1293	54
10:54	8464	5365	26.32	5.68	1288	55.4
10:55	8533	5346	26.35	5.68	1291	54
10:56	8489	5419	26.33	5.69	1292	54.7
10:57	8549	5425	26.34	5.69	1287	55
10:58	8370	5341	26.34	5.68	1291	55.6
10:59	8387	5276	26.34	5.68	1284	56.1
11:00	8463	5415	26.33	5.68	1286	56.1
11:01	8378	5374	26.35	5.68	1288	55.5
11:02	8321	5454	26.35	5.69	1289	54.8
11:03	8397	5379	26.38	5.67	1286	55.5
11:04	8312	5313	26.35	5.67	1289	54
11:05	8453	5358	26.37	5.66	1288	55.1
11:06	8436	5364	26.36	5.68	1288	54.8
11:07	8545	5363	26.37	5.68	1285	55.1
11:08	8375	5382	26.36	5.7	1287	55
11:09	8387	5379	26.37	5.71	1287	54.6
11:10	8359	5434	26.35	5.69	1285	55.2
11:11	8317	5395	26.36	5.69	1289	50.6
11:12	8347	5354	26.37	5.69	1287	51.2
11:13	8423	5514	26.37	5.68	1290	54.4
11:14	8438	5442	26.35	5.69	1285	55.1
11:15	8389	5264	26.35	5.7	1287	54.6
11:16	8377	5336	26.34	5.7	1290	54.5
11:17	8530	5387	26.31	5.71	1284	54.6
11:18	8316	5390	26.31	5.7	1291	53.6
11:19	8398	5467	26.33	5.71	1289	55
Average	8419	5400	26.33	5.67	1293	54.8
Stddev	72	56	0.03	0.03	5	1.5

Vacuum Pump Life Test for the 210 kW OC-OTEC Experimental Plant

Appendix A

EVAPORATOR/TURBINE MEASURED PARAMETERS

Time History Record (60 minutes long) Typical Nov. Conditions							
Time	Tevpin	Tevpout	Pintrb	Pouttrb	DPtrb	Prec.diff	TRBrpm
	TI-10	TI-12	PI-32	PI-33	PDI-31	PDI-32	SI-41
10:20	26.28	22.51	2517	1364	1199	32	1834.4
10:21	26.28	22.48	2516	1362	1199	32	1834.4
10:22	26.27	22.48	2515	1362	1197	32	1834.4
10:23	26.29	22.5	2519	1364	1201	32	1834.4
10:24	26.29	22.5	2517	1363	1199	32	1834.4
10:25	26.31	22.47	2515	1363	1197	32	1834.4
10:26	26.29	22.51	2516	1362	1200	32	1834.4
10:27	26.29	22.53	2517	1364	1199	32	1834.4
10:28	26.3	22.5	2516	1361	1201	33	1834.4
10:29	26.31	22.5	2516	1362	1200	32	1834.4
10:30	26.31	22.49	2515	1361	1200	32	1834.4
10:31	26.31	22.48	2517	1363	1200	33	1834.4
10:32	26.29	22.49	2515	1362	1199	33	1834.4
10:33	26.3	22.54	2520	1363	1204	33	1834.4
10:34	26.32	22.51	2518	1363	1200	33	1834.4
10:35	26.31	22.49	2514	1361	1200	33	1834.4
10:36	26.3	22.51	2515	1361	1200	33	1834.4
10:37	26.32	22.48	2517	1359	1205	33	1834.4
10:38	26.32	22.48	2518	1361	1204	33	1834.4
10:39	26.35	22.46	2518	1362	1202	33	1834.4
10:40	26.37	22.46	2514	1361	1200	33	1834.4
10:41	26.37	22.48	2515	1363	1199	33	1834.4
10:42	26.34	22.48	2514	1363	1199	32	1834.4
10:43	26.36	22.51	2516	1361	1202	32	1834.4
10:44	26.36	22.52	2511	1361	1198	32	1834.4
10:45	26.35	22.5	2510	1359	1197	32	1834.4
10:46	26.35	22.5	2512	1360	1199	31	1834.4
10:47	26.34	22.5	2510	1361	1196	31	1834.4
10:48	26.36	22.48	2510	1361	1196	31	1834.4
10:49	26.36	22.49	2507	1362	1192	31	1834.4
10:50	26.34	22.48	2505	1359	1194	31	1834.4
10:51	26.33	22.44	2508	1360	1195	31	1834.4
10:52	26.33	22.42	2505	1359	1194	31	1834.4
10:53	26.33	22.44	2507	1360	1195	31	1834.4
10:54	26.32	22.47	2509	1361	1195	31	1834.4
10:55	26.35	22.47	2507	1361	1194	31	1834.4
10:56	26.33	22.47	2508	1360	1196	31	1834.4
10:57	26.34	22.48	2504	1357	1195	31	1834.4
10:58	26.34	22.49	2505	1359	1194	31	1834.4
10:59	26.34	22.46	2505	1359	1194	31	1834.4
11:00	26.33	22.46	2507	1360	1195	31	1834.4
11:01	26.35	22.46	2508	1361	1196	31	1834.4
11:02	26.35	22.45	2509	1362	1196	31	1834.4
11:03	26.38	22.49	2509	1362	1195	30	1834.4
11:04	26.35	22.51	2508	1362	1194	30	1834.4
11:05	26.37	22.5	2511	1363	1197	30	1834.4
11:06	26.36	22.5	2510	1364	1194	30	1834.4
11:07	26.37	22.49	2508	1363	1193	30	1834.4
11:08	26.36	22.48	2508	1364	1192	30	1834.4
11:09	26.37	22.5	2508	1363	1191	30	1834.4
11:10	26.35	22.47	2508	1362	1194	29	1834.4
11:11	26.36	22.49	2511	1364	1194	29	1834.4
11:12	26.37	22.47	2512	1364	1195	29	1834.4
11:13	26.37	22.49	2508	1364	1191	29	1834.4
11:14	26.35	22.51	2510	1364	1194	29	1834.4
11:15	26.35	22.49	2511	1365	1194	29	1834.4
11:16	26.34	22.48	2509	1366	1191	29	1834.4
11:17	26.31	22.46	2513	1368	1192	29	1834.4
11:18	26.31	22.46	2507	1366	1188	29	1834.4
11:19	26.33	22.49	2512	1365	1194	30	1834.4
Average	26.33	22.48	2512	1362	1197	31	1834.4
Stdev	0.03	0.02	4	2	4	1	0

Vacuum Pump Life Test for the 210 kW OC-OTEC Experimental Plant

Appendix A

DCC MEASURED PARAMETERS

Time History Record Typical Nov. Conditions						T1ststg	T1ststg	T1ststg	T2ndstg
	Tdccc	Tdccc	Pin dcc	DP1stddcc	DP2ndddcc	outer	middle	inner	middle
Time	TI-20	TI-22	PI-33	PDI-33	PDI-34	TI-37	TI-30	TI-32	TI-31
10:20	5.67	11.53	1364	16	59	11.77	11.71	11.79	11.29
10:21	5.65	11.53	1362	16	58	11.76	11.66	11.8	11.3
10:22	5.64	11.55	1362	16	59	11.73	11.66	11.79	11.27
10:23	5.66	11.53	1364	16	58	11.76	11.71	11.82	11.29
10:24	5.64	11.53	1363	16	61	11.75	11.72	11.8	11.29
10:25	5.64	11.52	1363	16	61	11.73	11.71	11.8	11.29
10:26	5.65	11.51	1362	16	60	11.76	11.71	11.81	11.25
10:27	5.65	11.49	1364	16	62	11.74	11.72	11.82	11.28
10:28	5.66	11.49	1361	16	60	11.72	11.68	11.8	11.25
10:29	5.65	11.49	1362	16	58	11.73	11.68	11.78	11.3
10:30	5.65	11.49	1361	17	57	11.72	11.68	11.8	11.3
10:31	5.66	11.49	1363	16	60	11.74	11.7	11.8	11.31
10:32	5.66	11.48	1362	16	59	11.75	11.71	11.8	11.29
10:33	5.66	11.48	1363	17	63	11.74	11.71	11.82	11.28
10:34	5.65	11.48	1363	16	63	11.76	11.73	11.83	11.28
10:35	5.63	11.51	1361	16	63	11.74	11.72	11.8	11.28
10:36	5.62	11.49	1361	17	59	11.73	11.72	11.76	11.26
10:37	5.6	11.47	1359	17	61	11.73	11.7	11.78	11.28
10:38	5.61	11.48	1361	17	58	11.73	11.7	11.76	11.3
10:39	5.62	11.47	1362	17	61	11.72	11.7	11.78	11.28
10:40	5.62	11.47	1361	16	64	11.7	11.7	11.77	11.27
10:41	5.62	11.47	1363	16	64	11.7	11.69	11.78	11.28
10:42	5.63	11.48	1363	16	64	11.66	11.68	11.77	11.24
10:43	5.65	11.48	1361	16	60	11.68	11.67	11.79	11.27
10:44	5.66	11.47	1361	16	58	11.68	11.66	11.78	11.26
10:45	5.65	11.44	1359	17	59	11.65	11.59	11.76	11.23
10:46	5.66	11.47	1360	17	57	11.69	11.61	11.75	11.24
10:47	5.67	11.47	1361	16	59	11.67	11.59	11.74	11.21
10:48	5.69	11.46	1361	16	63	11.61	11.62	11.73	11.18
10:49	5.68	11.46	1362	16	61	11.56	11.61	11.69	11.17
10:50	5.69	11.46	1359	16	59	11.57	11.56	11.7	11.17
10:51	5.66	11.48	1360	17	58	11.58	11.56	11.67	11.2
10:52	5.67	11.47	1359	17	57	11.53	11.56	11.65	11.18
10:53	5.69	11.46	1360	17	57	11.56	11.52	11.66	11.17
10:54	5.68	11.46	1361	17	61	11.58	11.5	11.67	11.13
10:55	5.68	11.45	1361	17	60	11.55	11.5	11.69	11.11
10:56	5.69	11.46	1360	17	57	11.53	11.49	11.69	11.13
10:57	5.69	11.45	1357	17	57	11.52	11.5	11.64	11.15
10:58	5.68	11.47	1359	17	56	11.53	11.44	11.6	11.12
10:59	5.68	11.47	1359	16	61	11.51	11.42	11.61	11.13
11:00	5.68	11.45	1360	16	60	11.55	11.43	11.65	11.11
11:01	5.68	11.43	1361	16	59	11.54	11.45	11.66	11.11
11:02	5.69	11.39	1362	16	59	11.54	11.43	11.63	11.11
11:03	5.67	11.46	1362	16	62	11.53	11.41	11.64	11.11
11:04	5.67	11.48	1362	16	62	11.53	11.42	11.66	11.1
11:05	5.66	11.47	1363	16	60	11.53	11.42	11.64	11.1
11:06	5.68	11.47	1364	16	63	11.52	11.39	11.62	11.1
11:07	5.68	11.49	1363	16	61	11.55	11.4	11.64	11.09
11:08	5.7	11.49	1364	16	61	11.53	11.37	11.65	11.06
11:09	5.71	11.49	1363	15	60	11.47	11.35	11.61	11.06
11:10	5.69	11.48	1362	16	60	11.46	11.35	11.58	11.07
11:11	5.69	11.5	1364	16	58	11.46	11.35	11.57	11.07
11:12	5.69	11.49	1364	16	60	11.44	11.36	11.59	11.05
11:13	5.68	11.49	1364	16	58	11.47	11.37	11.6	11.03
11:14	5.69	11.48	1364	16	61	11.51	11.31	11.58	10.99
11:15	5.7	11.48	1365	16	61	11.47	11.29	11.58	11.02
11:16	5.7	11.51	1366	16	59	11.47	11.28	11.58	11.03
11:17	5.71	11.5	1368	16	66	11.47	11.31	11.63	11.05
11:18	5.7	11.52	1366	16	60	11.5	11.3	11.63	11.03
11:19	5.71	11.52	1365	16	60	11.5	11.29	11.55	11.04
Average	5.67	11.48	1362	16	60	11.61	11.55	11.71	11.18
Stdev	0.03	0.03	2	0	2	0.11	0.15	0.08	0.1

Vacuum Pump Life Test for the 210 kW OC-OTEC Experimental Plant

Appendix A

POWER PRODUCTION (KW)

Time History Record Typical Nov. Conditions				
	PWgross	PWesstia	WPnones.	PWnet
Time				
10:20	192.5	137.4	9.8	55.4
10:21	191.7	137.2	9.8	55
10:22	191.9	136.9	9.8	55.4
10:23	192.7	136.8	9.8	56.3
10:24	191.7	136.7	9.8	55.4
10:25	191.7	136.7	9.9	55.5
10:26	190.6	136.7	9.8	54.2
10:27	191.7	136.9	9.8	55.4
10:28	192.4	136.8	9.9	55.9
10:29	192.2	137.1	9.8	55.5
10:30	191.8	136.9	9.9	55.3
10:31	192.8	141.2	9.9	51.9
10:32	192	141.4	9.9	51
10:33	193	137	9.8	56.5
10:34	191.8	136.7	9.9	55.6
10:35	192	137	9.8	55.5
10:36	192.1	137	9.9	55.4
10:37	193.6	137	9.9	57
10:38	193.2	136.6	9.9	57
10:39	192.5	136.7	9.9	56.2
10:40	192.2	136.5	10	56
10:41	192.6	136.5	9.9	56.5
10:42	192.2	137	9.9	55.7
10:43	191.8	136.7	10	55.4
10:44	192.1	136.9	9.9	55.7
10:45	191.6	137	9.9	55.1
10:46	192.6	136.8	9.9	56.4
10:47	191.4	136.4	10	55.4
10:48	191.2	136.5	9.9	55.2
10:49	191	137	9.9	54.3
10:50	191	136.7	9.9	54.8
10:51	191.3	141.3	9.9	50.5
10:52	191.2	141.5	9.9	50.2
10:53	191	137.6	9.9	54
10:54	191.2	136.3	9.9	55.4
10:55	190.8	137.1	9.9	54
10:56	191.3	137.1	9.9	54.7
10:57	191.4	136.9	9.9	55
10:58	191.6	136.5	9.9	55.6
10:59	191.9	136.3	9.9	56.1
11:00	191.8	136.2	9.9	56.1
11:01	191.4	136.4	9.9	55.5
11:02	190.7	136.4	9.8	54.8
11:03	191.3	136.3	9.9	55.5
11:04	189.9	136.5	9.9	54
11:05	191.2	136.5	9.9	55.1
11:06	190.6	136.4	9.9	54.8
11:07	190.8	136.2	9.8	55.1
11:08	190.9	136.3	9.9	55
11:09	190.6	136.5	9.9	54.6
11:10	191.1	136.3	9.9	55.2
11:11	191	141	9.9	50.6
11:12	191.8	141.2	10.6	51.2
11:13	191.3	137.3	10.2	54.4
11:14	191.2	136.3	10.4	55.1
11:15	191.4	137.4	9.9	54.6
11:16	190.3	136.1	9.9	54.5
11:17	190.4	136.1	9.9	54.6
11:18	189.9	136.7	9.9	53.6
11:19	191.2	136.8	9.9	55
Average	191.6	137.2	9.9	54.8
Stdev	0.8	1.4	0.1	1.5

Vacuum Pump Life Test for the 210 kW OC-OTEC Experimental Plant

Appendix A

VACUUM COMPRESSOR SUBSYSTEM PARAMETERS

Time History Record (60 minutes long) Typical Nov. Conditions												
	Pin1st	Pin2nd	Pin3rd	Pin4th	Pin5th	Mnc.g/s	AMP	RPM	AMP	RPM	AMP	RPM
Time	PI-105	PI-104	PI-103	PI-102	PI-101	FI-60	AMP-1A	RPM-1A	AMP-1B	RPM-1B	AMP-1C	RPM-1C
10:20	1299	2154	4487	8717	8374	18.5	4.5	25798	4.4	25798	4.4	25798
10:21	1299	2163	4506	8762	8420	18.7	4.4	25797	4.4	25797	4.4	25797
10:22	1299	2164	4509	8766	8418	18.6	4.4	25797	4.4	25797	4.4	25797
10:23	1301	2161	4501	8745	8401	18.6	4.4	25799	4.4	25799	4.4	25799
10:24	1300	2166	4511	8754	8413	18.6	4.4	25797	4.4	25797	4.3	25797
10:25	1298	2154	4487	8720	8376	18.5	4.4	25796	4.4	25796	4.3	25796
10:26	1300	2167	4516	8778	8432	18.6	4.4	25795	4.4	25795	4.3	25795
10:27	1300	2164	4513	8770	8426	18.7	4.4	25795	4.4	25795	4.3	25795
10:28	1298	2148	4481	8708	8369	18.5	4.4	25801	4.4	25801	4.4	25801
10:29	1301	2156	4505	8746	8406	18.6	4.4	25799	4.4	25799	4.4	25799
10:30	1300	2158	4509	8760	8418	18.6	4.4	25803	4.4	25803	4.3	25803
10:31	1300	2150	4500	8714	8369	18.4	4.4	25798	4.4	25798	4.4	25798
10:32	1302	2169	4539	8788	8445	18.6	4.4	25800	4.4	25800	4.3	25800
10:33	1297	2147	4498	8703	8362	18.4	4.4	25799	4.4	25799	4.3	25799
10:34	1300	2166	4531	8800	8453	18.6	4.4	25800	4.4	25800	4.3	25800
10:35	1298	2149	4501	8714	8371	18.4	4.4	25797	4.4	25797	4.3	25797
10:36	1301	2168	4532	8805	8458	18.6	4.4	25796	4.4	25796	4.3	25796
10:37	1297	2146	4485	8707	8366	18.4	4.4	25799	4.4	25799	4.3	25799
10:38	1299	2159	4513	8767	8423	18.4	4.4	25793	4.4	25793	4.4	25793
10:39	1296	2158	4512	8760	8420	18.4	4.4	25801	4.4	25801	4.3	25801
10:40	1293	2152	4494	8731	8391	18.4	4.4	25800	4.4	25800	4.3	25800
10:41	1292	2152	4494	8727	8384	18.3	4.4	25800	4.4	25800	4.4	25800
10:42	1293	2154	4497	8735	8392	18.3	4.4	25798	4.4	25798	4.4	25798
10:43	1291	2146	4474	8687	8352	18.2	4.4	25799	4.4	25799	4.4	25799
10:44	1293	2163	4509	8755	8412	18.3	4.4	25800	4.4	25800	4.4	25800
10:45	1291	2148	4469	8679	8342	18.2	4.4	25804	4.4	25804	4.4	25804
10:46	1293	2159	4494	8730	8390	18.3	4.4	25800	4.4	25800	4.4	25800
10:47	1293	2163	4500	8743	8401	18.4	4.4	25800	4.4	25800	4.4	25800
10:48	1289	2154	4471	8690	8355	18.3	4.4	25802	4.4	25802	4.4	25802
10:49	1292	2168	4507	8756	8413	18.2	4.4	25802	4.4	25802	4.4	25802
10:50	1290	2161	4488	8706	8368	18.3	4.4	25802	4.4	25802	4.3	25802
10:51	1293	2155	4478	8688	8349	18.2	4.4	25798	4.4	25798	4.4	25798
10:52	1293	2166	4504	8725	8384	18.3	4.4	25802	4.4	25802	4.4	25802
10:53	1293	2180	4526	8762	8421	18.4	4.4	25802	4.4	25802	4.3	25802
10:54	1288	2156	4483	8682	8343	18.2	4.4	25803	4.4	25803	4.4	25803
10:55	1291	2167	4507	8754	8411	18.3	4.4	25802	4.4	25802	4.3	25802
10:56	1292	2167	4508	8738	8397	18.2	4.4	25803	4.4	25803	4.4	25803
10:57	1287	2157	4501	8736	8398	18.3	4.4	25802	4.4	25802	4.4	25802
10:58	1291	2168	4517	8756	8410	18.2	4.4	25801	4.4	25801	4.4	25801
10:59	1284	2141	4498	8729	8389	18.3	4.4	25801	4.4	25801	4.4	25801
11:00	1286	2158	4490	8722	8381	18.2	4.4	25807	4.4	25807	4.4	25807
11:01	1288	2167	4508	8764	8421	18.3	4.4	25800	4.4	25800	4.3	25800
11:02	1289	2169	4514	8753	8412	18.3	4.4	25801	4.4	25801	4.4	25801
11:03	1286	2152	4488	8684	8345	18.1	4.4	25801	4.4	25801	4.4	25801
11:04	1289	2164	4513	8753	8410	18.3	4.4	25801	4.4	25801	4.3	25801
11:05	1288	2152	4486	8716	8375	18.1	4.4	25797	4.4	25797	4.4	25797
11:06	1288	2161	4502	8753	8412	18.2	4.4	25801	4.4	25801	4.4	25801
11:07	1285	2140	4463	8669	8333	18	4.4	25797	4.4	25797	4.4	25797
11:08	1287	2164	4516	8751	8409	18.2	4.4	25802	4.4	25802	4.4	25802
11:09	1287	2161	4516	8739	8397	18.1	4.4	25799	4.4	25799	4.4	25799
11:10	1285	2156	4495	8701	8368	18.2	4.4	25802	4.4	25802	4.4	25802
11:11	1289	2168	4518	8747	8410	18.2	4.4	25799	4.4	25799	4.4	25799
11:12	1287	2153	4482	8682	8348	18.1	4.5	25804	4.4	25804	4.4	25804
11:13	1290	2174	4528	8791	8452	18.3	4.4	25802	4.4	25802	4.4	25802
11:14	1285	2146	4463	8674	8343	18.1	4.5	25802	4.4	25802	4.4	25802
11:15	1287	2158	4491	8728	8392	18.2	4.5	25804	4.4	25804	4.4	25804
11:16	1290	2176	4526	8799	8462	18.3	4.4	25800	4.4	25800	4.4	25800
11:17	1284	2149	4469	8683	8348	18.2	4.4	25798	4.4	25798	4.4	25798
11:18	1291	2175	4527	8800	8460	18.4	4.4	25800	4.4	25800	4.4	25800
11:19	1289	2143	4460	8666	8330	18.1	4.5	25799	4.4	25799	4.4	25799
Average	1293	2159	4500	8734	8394	18.3	4.4	25800	4.4	25800	4.4	25800
Stdev	5	9	18	36	34	0.2	0.0	3	0	3	0	3

Vacuum Pump Life Test for the 210 kW OC-OTEC Experimental Plant

Appendix A

VACUUM COMPRESSOR SUBSYSTEM PARAMETERS

Time History Record (60 minutes long) Typical Nov. Conditions						
	AMP	RPM	AMP	RPM	RPM	AMP
Time	AMP-2	RPM-2	AMP-3	RPM-3	RPM-5B	AMP-5B
10:20	7.3	27784	6.3	35995	4149	28.6
10:21	7.3	27781	6.3	35987	4148	28.6
10:22	7.3	27786	6.3	35989	4147	28.6
10:23	7.3	27781	6.3	35990	4148	28.7
10:24	7.3	27784	6.3	35996	4148	28.7
10:25	7.3	27789	6.3	35990	4147	28.6
10:26	7.3	27783	6.3	36000	4147	28.7
10:27	7.3	27791	6.3	36000	4147	28.7
10:28	7.3	27791	6.2	35998	4147	28.6
10:29	7.3	27785	6.3	35992	4148	28.7
10:30	7.3	27785	6.3	35995	4147	28.7
10:31	7.3	27785	6.3	35992	4147	28.6
10:32	7.3	27786	6.3	35996	4147	28.7
10:33	7.3	27785	6.2	35987	4148	28.6
10:34	7.4	27790	6.3	36003	4148	28.6
10:35	7.3	27788	6.3	35988	4147	28.6
10:36	7.4	27791	6.3	35991	4147	28.7
10:37	7.3	27787	6.3	35989	4147	28.7
10:38	7.3	27785	6.3	36000	4147	28.7
10:39	7.3	27781	6.3	35995	4147	28.7
10:40	7.3	27791	6.3	35997	4147	28.6
10:41	7.3	27789	6.3	35997	4147	28.7
10:42	7.3	27783	6.3	35990	4147	28.6
10:43	7.3	27789	6.3	35988	4147	28.6
10:44	7.3	27788	6.3	35994	4147	28.7
10:45	7.3	27783	6.2	35987	4147	28.6
10:46	7.3	27785	6.3	36001	4147	28.6
10:47	7.3	27785	6.3	35989	4147	28.6
10:48	7.3	27791	6.2	35998	4147	28.6
10:49	7.3	27790	6.3	35997	4147	28.6
10:50	7.3	27791	6.3	36004	4147	28.5
10:51	7.3	27786	6.3	35996	4147	28.6
10:52	7.3	27792	6.3	35992	4146	28.6
10:53	7.3	27792	6.3	35996	4146	28.6
10:54	7.3	27788	6.3	36003	4145	28.6
10:55	7.3	27790	6.3	36001	4145	28.6
10:56	7.3	27794	6.3	36005	4146	28.6
10:57	7.2	27792	6.3	35994	4145	28.6
10:58	7.3	27789	6.3	35998	4145	28.6
10:59	7.2	27787	6.3	35998	4145	28.6
11:00	7.3	27786	6.3	35990	4145	28.5
11:01	7.3	27787	6.3	35991	4145	28.6
11:02	7.3	27788	6.3	35997	4145	28.6
11:03	7.3	27788	6.3	35993	4145	28.6
11:04	7.3	27788	6.3	35986	4145	28.6
11:05	7.3	27794	6.3	36008	4145	28.6
11:06	7.3	27784	6.3	35992	4146	28.6
11:07	7.3	27788	6.3	36000	4146	28.6
11:08	7.3	27788	6.3	35997	4147	28.6
11:09	7.3	27786	6.3	35995	4145	28.6
11:10	7.3	27787	6.3	35978	4145	28.6
11:11	7.4	27789	6.3	35998	4145	28.6
11:12	7.3	27787	6.3	35987	4145	28.6
11:13	7.4	27784	6.3	35988	4145	28.6
11:14	7.3	27785	6.3	35989	4145	28.6
11:15	7.3	27789	6.3	35997	4145	28.6
11:16	7.3	27788	6.3	35982	4145	28.6
11:17	7.3	27787	6.3	35986	4146	28.6
11:18	7.4	27784	6.4	35996	4146	28.6
11:19	7.3	27786	6.3	35990	4146	28.6
Average	7.3	27787	6.3	35994	4147	28.6
Stdev	0	3	0.0	6	1	0

Appendix B

UNCERTAINTY OF DERIVED PARAMETERS

Power Production Mode

This Appendix discusses the uncertainty of the parameters routinely derived at the 210 OC-OTEC Experimental Facility. The following end-to-end precisions were specified for the primary measurements at the design point:

Temperatures	± 0.05 °C
Steam Pressures	± 5 Pa
Water Flowrates	± 100 gpm
Mass Flowrate NC	± 1 g/s
Power	± 1 kW

Table of Contents

General Remarks	3
Evaporator Effectiveness	4
Direct Contact Condenser (DCC) Effectiveness	6
Steam Mass Flow Rate	9
Isentropic Power across the Turbine	10
Overall Fraction of Steam Condensed	12
References	13

UNCERTAINTY OF DERIVED PARAMETERS

General Remarks

The conceptual discussion below summarizes the ideas presented in Benedict (1984).

The measurement of any physical parameter is tainted by sources of error traditionally assigned to two types: a systematic bias, and a random error. Use of these concepts implies that the measurement is actually performed N times for a given input.

The average value μ , taken over the set of N readings X_k , is the estimator of the physical parameter, and the separation between μ and the true value is called bias, B. Strictly speaking, good accuracy is associated with small bias. Moreover, a 'perfect' instrument calibration, prior to the performance of the desired measurements, would actually eliminate any bias.

Similarly, the scatter of the N readings, away from their average, results from random errors, and strictly speaking, good precision is associated with small such errors. To quantify precision, or define confidence intervals, one may use the unbiased estimate of the standard deviation of the set, defined by

$$S = \sqrt{\frac{1}{N-1} \sum_{k=1}^N (X_k - \mu)^2}.$$

S is multiplied by $t_{N-1,p}$, the Student t distribution for N-1 degrees of freedom and probability p. If N becomes very large, the t statistics asymptotically tends to the better-known normal Gaussian distribution. In summary, μ will belong to the interval $X_k \pm t_{N-1,p}S$, defined by any measured value X_k , with the probability p.

It must be noted that unlike random errors, bias errors do not lend themselves to any statistical treatment. This presents a dilemma when one wishes to simply quantify the uncertainty U of a measurement, whereby a single confidence interval would encompass both the effect of systematic and random errors. Experience proved that in general, defining U as the sum $B + t_{N-1,p}S$ is too conservative. A favored formula is rather:

$$U = \sqrt{B^2 + (t_{N-1,0.95} S)^2}$$

The simple concepts outlined above, however, are less easily applicable as they seem. In general, the manufacturers of measuring devices do not go into sufficient details for a clear distinction between accuracy and precision. Upon Benedict's recommendation, it may be preferable to consider the uncertainty information provided without statistical qualification, or simply estimated by the performer of the measurements, as systematic error, or bias. On the other hand, any attempt to determine S experimentally is only possible if one can reasonably satisfy the given (fixed) input requirement.

Therefore, estimated uncertainties δP_i , for individual devices measuring the physical parameters P_i , will be treated as bias. Once again with little statistical justification, the uncertainty δQ on a derived parameter $Q(P_i)$, function of M parameters P_i , is estimated as:

$$\delta Q = \sqrt{\sum_{i=1}^M \left(\frac{\partial Q}{\partial P_i} \delta P_i \right)^2}$$

Although the expression above cannot be quantified from a probabilistic standpoint like random errors, its use seems to have proved satisfactory on the basis of experience. We will therefore state that the true value of the derived physical parameter Q is 'likely' to belong to the interval $Q \pm \delta Q$. This interval could be restricted further when there is a theoretical reason to do so: e.g., a heat exchanger effectiveness cannot exceed 1.0 (100%).

Examples will now be given to illustrate the above formula in the context of the 210 kW OC-OTEC Experimental Apparatus.

Evaporator Effectiveness

The definition of evaporator effectiveness is provided below:

$$\epsilon = \frac{T_{in} - T_{out}}{T_{in} - \{T_{sat}(P_{ev}) + 0.31\}}$$

The baseline values used henceforth are given as $T_{in} = 26^\circ\text{C}$, $T_{out} = 22.5^\circ\text{C}$, $P_{ev} = 2620$ Pa and the following curve fit for T_{sat} , in $^\circ\text{C}$:

$$T_{sat}(P_{ev}) = \frac{4026.9759}{-\text{Log}\left\{\frac{P_{ev} - 3.74}{161.7574}\right\} + 18.4779} - 234.7384$$

This input yields $T_{sat}(P_{ev}) = 21.85^\circ\text{C}$ and $\epsilon = 0.91$.

Two alternate ways are considered to experimentally determine ϵ : both utilize a pressure reading of P_{ev} and a differential temperature measurement of the numerator, $\Delta T_{ev} = T_{in} - T_{out}$; the inlet water temperature in the numerator, however, can be obtained either a) directly or b) by adding ΔT_{ev} to T_{out} .

$$\text{a) } \epsilon = \frac{\Delta T_{ev}}{T_{in} - \{T_{sat}(P_{ev}) + 0.31\}}$$

The following sensitivity coefficients must be considered:

$$\frac{\partial \epsilon}{\partial T_{in}} = - \frac{\Delta T_{ev}}{(T_{in} - \{T_{sat}(P_{ev}) + 0.31\})^2} = - \frac{\epsilon^2}{\Delta T_{ev}} \approx -0.237 \text{ K}^{-1}$$

OC-OTEC Vacuum Pumps Life Cycle Tests

Appendix B

$$\frac{\partial \epsilon}{\partial \Delta T_{ev}} = \frac{1}{T_{in} - \{T_{sat}(p_{ev}) + 0.31\}} = \frac{\epsilon}{\Delta T_{ev}} \approx 0.260 \text{ K}^{-1}$$

$$\frac{\partial \epsilon}{\partial p_{ev}} = \frac{\partial \epsilon}{\partial T_{sat}} \frac{\partial T_{sat}}{\partial p_{ev}} \approx 0.001481 \text{ Pa}^{-1}$$

$$\text{since } \frac{\partial \epsilon}{\partial T_{sat}} = -\frac{\partial \epsilon}{\partial T_{in}} \approx 0.237 \text{ K}^{-1},$$

$$\text{and } \frac{\partial T_{sat}}{\partial p_{ev}} = \frac{4026.9759 \left\{ \frac{1}{p_{ev} - 3.74} \right\}}{\left\{ -\text{Log} \frac{p_{ev} - 3.74}{1617574} + 18.4779 \right\}^2} = 0.00625 \text{ K/Pa}$$

$$\text{Moreover, } \delta \epsilon = \sqrt{\left(\frac{\partial \epsilon}{\partial T_{in}} \delta T_{in} \right)^2 + \left(\frac{\partial \epsilon}{\partial \Delta T_{ev}} \delta \Delta T_{ev} \right)^2 + \left(\frac{\partial \epsilon}{\partial p_{ev}} \delta p_{ev} \right)^2}$$

With $\delta T_{in} = 0.05^\circ\text{C}$ (TE-10), $\delta \Delta T_{ev} = 0.02^\circ\text{C}$ (TdT-11) and $\delta p_{ev} = 5.4 \text{ Pa}$ (PT-30), one obtains $\delta \epsilon \approx 0.015$

$$\text{i.e. } \frac{\delta \epsilon}{\epsilon} = 1.7 \%$$

$$\text{b) } \epsilon = \frac{\Delta T_{ev}}{T_{out} + \Delta T_{ev} - \{T_{sat}(p_{ev}) + 0.31\}}$$

The following sensitivity coefficients must be considered:

$$\frac{\partial \epsilon}{\partial T_{out}} = -\frac{\epsilon^2}{\Delta T_{ev}} \approx -0.237 \text{ K}^{-1}$$

$$\frac{\partial \epsilon}{\partial \Delta T_{ev}} = \frac{T_{out} - \{T_{sat}(p_{ev}) + 0.31\}}{(T_{in} - \{T_{sat}(p_{ev}) + 0.31\})^2} = \frac{\epsilon - \epsilon^2}{\Delta T_{ev}} \approx 0.023 \text{ K}^{-1}$$

$$\frac{\partial \epsilon}{\partial p_{ev}} \text{ is the same as above.}$$

$$\text{Moreover, } \delta \epsilon = \sqrt{\left(\frac{\partial \epsilon}{\partial T_{out}} \delta T_{out} \right)^2 + \left(\frac{\partial \epsilon}{\partial \Delta T_{ev}} \delta \Delta T_{ev} \right)^2 + \left(\frac{\partial \epsilon}{\partial p_{ev}} \delta p_{ev} \right)^2}$$

With $\delta T_{out} = 0.05^\circ\text{C}$ (TE-12), $\delta \Delta T_{ev} = 0.02^\circ\text{C}$ (TdT-11) and $\delta p_{ev} = 5.4 \text{ Pa}$ (PT-30), one obtains $\delta \epsilon \approx 0.014$

$$\text{i.e. } \frac{\delta \epsilon}{\epsilon} = 1.6 \%$$

It is interesting to note that the latter method for determining evaporator effectiveness is slightly more accurate: the presence of ΔT_{ev} in both numerator and denominator, in the definition of ϵ , reduces the corresponding sensitivity coefficient by one order of magnitude.

Direct Contact Condenser (DCC) Effectiveness

The definition of the DCC effectiveness mirrors that of the evaporator discussed in the previous section:

$$\epsilon_{DCC} = \frac{T_{out} - T_{in}}{\{T_{sat}(p_{steam}) + 0.28\} - T_{in}}$$

where p_{steam} is the partial pressure of steam at the inlet of the condenser. Such a definition may be applied separately to the first (co-current) and second (counter-current) DCC stages, as well as to the overall heat exchanger.

All seawater temperatures are measured directly for the two individual DCC stages (TE-30 and TE-31, respectively), but the temperature rise across the overall heat exchanger is provided by a differential temperature measurement, ΔT_{DCC} (TdT-21). Consequently, the overall effectiveness should be evaluated from:

$$\epsilon_{DCC} = \frac{\Delta T_{DCC}}{\{T_{sat}(p_{inlet}) + 0.28\} + \Delta T_{DCC} - T_{out}}$$

For the first-stage and overall DCC, the amount of non-condensibles at the inlet is negligible, and p_{steam} is taken as the overall pressure p_{inlet} , measured in those cases by PT-33. In the absence of a direct temperature measurement for the vapor phase between the two DCC stages, the second-stage inlet partial pressure requires a careful evaluation: calling the total interstage pressure $(p - \Delta p)$, known by the difference between PT-33 and PdT-33, the ratio $p_{steam}/(p - \Delta p)$ is equal to the ratio of steam molar flow rate over total molar flow rate; thus:

$$p_{steam} = \frac{(p - \Delta p)}{1 + \frac{18}{29} \frac{m_{nc1}}{m_{steam}}}$$

m_{nc1} represents the mass flow of non-condensibles between DCC stages, and can be estimated by subtracting the estimated release in the second DCC stage alone from the total amount m_{nc} , measured by FE-60:

$$m_{nc1} = m_{nc} - f_{cw} \rho_{cw} Q_{cw} R_{2t}$$

in the above formula, f_{cw} is a constant, equal to 18.22×10^{-6} kg/kg, expressing cold seawater non-condensable content, ρ_{cw} is the cold seawater density, Q_{cw} the cold seawater flow rate, measured by FE-20, whereas the second-stage seawater distribution ratio R_{2t} is $\frac{(T_{cw1} - T_{cwo})}{(T_{cw1} - T_{cw2})}$, the latter quantity involves the measurement of seawater DCC outlet temperatures T_{cw1} (first stage), $T_{cw2} = T_{out}$ (second stage) and

OC-OTEC Vacuum Pumps Life Cycle Tests

Appendix B

T_{CWO} (overall) by, respectively, TE-30, TE-31 and TE-22. It can be verified that m_{nc1} has a baseline value of 19.86 g/s.

m_{steam} represents what has not been condensed through the first DCC stage, that is $\chi m_s(1 - F_1)$; the total DCC inlet steam mass flow rate χm_s may be treated as a pseudo-measurement; moreover, we have:

$$\chi m_s F_1 = \frac{c_{p,cw} Q_{cw} (T_{cw1} - T_{in})(T_{cwo} - T_{out})}{h_{fg}(T_{cw1} - T_{out})},$$

where the only additional temperature measurement is that of the cold seawater inlet, T_{in} , by TE-20.

For the second stage, the baseline values of p_{steam} and $(p - \Delta p)$ are, respectively, 1328 Pa and 1353 Pa.

Baseline values of the main terms in the thermal effectiveness formula are given as:

	T_{sat}	T_{in}	T_{out}	ΔT_{DCC}	ϵ_{DCC}
First-stage	11.8°C	6.1°C	11.4°C		0.89
Second-stage	11.1°C	6.1°C	10.5°C		0.83
Overall	11.8°C		11.2°C	5.1°C	0.85

For the first stage and overall DCC, the relevant sensitivity coefficients are obtained readily:

$$\frac{\partial \epsilon}{\partial T_{in}} = \frac{T_{out} - \{T_{sat}(p_{steam}) + 0.28\}}{(\{T_{sat}(p_{steam}) + 0.28\} - T_{in})^2} = -\frac{\epsilon - \epsilon^2}{T_{out} - T_{in}},$$

$$\frac{\partial \epsilon}{\partial T_{out}} = \frac{\epsilon}{T_{out} - T_{in}} \text{ for the first stage,}$$

$$\frac{\partial \epsilon}{\partial T_{out}} = \frac{\epsilon^2}{T_{out} - T_{in}} \text{ for the overall DCC,}$$

$$\frac{\partial \epsilon}{\partial p_{inlet}} = \frac{\partial \epsilon}{\partial T_{sat}} \frac{\partial T_{sat}}{\partial p_{inlet}} = -\frac{\epsilon^2}{T_{out} - T_{in}} \frac{\partial T_{sat}}{\partial p_{inlet}},$$

$$\text{with } \frac{\partial T_{sat}}{\partial p_{inlet}}(1383 \text{ Pa}) = 0.01094 \text{ K/Pa,}$$

$$\frac{\partial \epsilon}{\partial \Delta T_{DCC}} = \frac{\{T_{sat}(p_{inlet}) + 0.28\} - T_{out}}{(\{T_{sat}(p_{inlet}) + 0.28\} - T_{in})^2} = \frac{\epsilon - \epsilon^2}{T_{out} - T_{in}}.$$

OC-OTEC Vacuum Pumps Life Cycle Tests

Appendix B

Numerical values of these coefficients are presented below as well as the resulting uncertainty on the DCC effectiveness based on a temperature measurement accuracy of 0.05°C (TE-20 and TE-30), $\delta p_{\text{inlet}} = 3 \text{ Pa}$ (PT-33) and $\delta \Delta T_{\text{DCC}} = 0.02^\circ\text{C}$ (TdT-21):

	$\frac{\partial \epsilon}{\partial T_{\text{in}}}$	$\frac{\partial \epsilon}{\partial \Delta T_{\text{DCC}}}$	$\frac{\partial \epsilon}{\partial T_{\text{out}}}$	$\frac{\partial \epsilon}{\partial p_{\text{inlet}}}$	$\delta \epsilon$	$\frac{\delta \epsilon}{\epsilon}$
First-stage	-0.019		0.168	1.635×10^{-3}	9.8×10^{-3}	1.1%
Overall		0.025	0.142	1.553×10^{-3}	8.5×10^{-3}	1%

As has been discussed, the analysis for the second stage is complicated by the cumbersome determination of the inlet partial pressure p_{steam} . The following sensitivity coefficients have to be considered:

$$\frac{\partial \epsilon}{\partial T_{\text{in}}} = -\frac{\epsilon - \epsilon^2}{T_{\text{out}} - T_{\text{in}}} - \frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial T_{\text{in}}} = 1.503 \times 10^{-3} \text{ K}^{-1},$$

$$\frac{\partial \epsilon}{\partial T_{\text{out}}} = \frac{\epsilon}{T_{\text{out}} - T_{\text{in}}} - \frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial T_{\text{out}}} = 0.24935 \text{ K}^{-1},$$

$$\frac{\partial \epsilon}{\partial T_{\text{cw1}}} = -\frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial T_{\text{cw1}}} = 0.17787 \text{ K}^{-1},$$

$$\frac{\partial \epsilon}{\partial T_{\text{cwo}}} = -\frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial T_{\text{cwo}}} = -0.27154 \text{ K}^{-1},$$

$$\frac{\partial \epsilon}{\partial p} = -\frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial p} = -1.743 \times 10^{-3} \text{ Pa}^{-1},$$

$$\frac{\partial \epsilon}{\partial \Delta p} = -\frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial \Delta p} = 1.743 \times 10^{-3} \text{ Pa}^{-1},$$

$$\frac{\partial \epsilon}{\partial m_{\text{nc}}} = -\frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial m_{\text{nc}}} = -2.1572 \times 10^{-3} (\text{g/s})^{-1},$$

$$\frac{\partial \epsilon}{\partial Q_{\text{cw}}} = -\frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial Q_{\text{cw}}} = -0.4376 (\text{m}^3/\text{s})^{-1},$$

$$\frac{\partial \epsilon}{\partial \chi_{\text{ms}}} = -\frac{\epsilon^2}{T_{\text{out}} - T_{\text{in}}} \frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} \frac{\partial p_{\text{steam}}}{\partial \chi_{\text{ms}}} = 0.06434 (\text{kg/s})^{-1}.$$

The numerical values above were obtained with $\frac{\partial T_{\text{sat}}}{\partial p_{\text{steam}}} (1328.4) = 0.01134 \text{ K/Pa}$.

Details of the straightforward, though tedious, algebraic steps required for the calculation of the various partial derivatives of p_{steam} are left out. With an

uncertainty of 0.05°C on all temperature readings, $\delta m_{nc} = 1.6 \text{ g/s}$, $\delta p = 3 \text{ Pa}$, $\delta \Delta p = 2.4 \text{ Pa}$, $\delta Q_{cw} = 0.0061 \text{ m}^3/\text{s}$ and $\delta \chi m_s \approx \chi \delta m_s = 5.55 \times 10^{-2} \text{ kg/s}$ (δm_s is evaluated elsewhere), the uncertainty for the effectiveness of the second DCC stage is $\delta \epsilon = 0.0223$, i.e. $\frac{\partial \epsilon}{\epsilon} = 2.7\%$.

Steam Mass Flow Rate

The steam mass flow rate, in kg/s, is estimated from the formula:

$$m_s = \frac{\rho_w c_p Q_{ww} \Delta T_{ev}}{h_{fg}}$$

The warm seawater volume flow rate Q_{ww} is measured directly, as well as the temperature difference ΔT_{ev} across the evaporator. The other parameters, however, are defined from appropriate curve fits functions of $T_{avg} = \frac{T_{in} - T_{out}}{2}$ and $T_{sat}(p_{ev})$, both expressed in $^{\circ}\text{C}$. More explicitly:

$$\rho_w = 1027.7154 - 0.04992 T_{avg} - 0.006773 T_{avg}^2 + 4.766 \times 10^{-5} T_{avg}^3$$

$$c_p = 3.9867 + 4.773 \times 10^{-4} T_{avg}$$

$$h_{fg} = 2501.6 + (1.86 - c_p) \{T_{sat}(p_{ev}) + 0.31\}$$

With $T_{in} = 26^{\circ}\text{C}$ and $T_{sat}(2620) = 21.85^{\circ}\text{C}$, the baseline values of the three parameters above are, respectively, 1023.20 kg/m^3 , 3.998 kJ/kg-K and 2454.2 kJ/kg . With $Q_{ww} = 0.6057 \text{ m}^3/\text{s}$ and $\Delta T_{ev} = 3.5^{\circ}\text{C}$, the nominal steam flow rate is 3.536 kg/s .

The following sensitivity coefficients need to be considered:

$$\frac{\partial m_s}{\partial Q_{ww}} = \frac{\rho_w c_p \Delta T_{ev}}{h_{fg}} = \frac{m_s}{Q_{ww}} \approx 5.8377 \text{ kg/m}^3$$

$$\frac{\partial m_s}{\partial \Delta T_{ev}} = \frac{\rho_w c_p Q_{ww}}{h_{fg}} = \frac{m_s}{\Delta T_{ev}} \approx 1.0114 \text{ kg/s-K}$$

$$\frac{\partial m_s}{\partial p_{ev}} = \frac{\partial m_s}{\partial h_{fg}} \frac{\partial h_{fg}}{\partial T_{sat}} \frac{\partial T_{sat}}{\partial p_{ev}} = -\frac{(1.86 - c_p) m_s}{h_{fg}} \frac{\partial T_{sat}}{\partial p_{ev}} \approx 1.93 \times 10^{-5} \text{ kg/s-Pa}$$

$$\frac{\partial m_s}{\partial T_{in}} = \frac{\partial m_s}{\partial T_{out}} = 0.5 \frac{\partial m_s}{\partial T_{avg}} \approx -0.0005 \text{ kg/s-K}$$

$$\text{with } \frac{\partial m_s}{\partial T_{avg}} = \frac{\partial m_s}{\partial \rho_w} \frac{\partial \rho_w}{\partial T_{avg}} + \frac{\partial m_s}{\partial c_p} \frac{\partial c_p}{\partial T_{avg}}$$

$$= \frac{m_s}{\rho_w} \frac{\partial \rho_w}{\partial T_{avg}} + \frac{h_{fg} + c_p \{T_{sat}(p_{ev}) + 0.31\}}{h_{fg}^2} \frac{\partial c_p}{\partial T_{avg}}$$

With $\delta Q_{ww} = 0.0091 \text{ m}^3/\text{s}$ (FE-10), $\delta \Delta T_{ev} = 0.02^\circ\text{C}$ (TdT-11), $\delta p_{ev} = 5.4 \text{ Pa}$ (PT-30), $\delta T_{in} = \delta T_{out} = 0.05^\circ\text{C}$ (TE-10 and TE-11), and the above sensitivity coefficients, it is obvious that the contributions of T_{in} , T_{out} and p_{ev} to δm_s are negligible. Consequently:

$$\delta m_s \approx \sqrt{\left(\frac{\partial m_s}{\partial Q_{ww}} \delta Q_{ww}\right)^2 + \left(\frac{\partial m_s}{\partial \Delta T_{ev}} \delta \Delta T_{ev}\right)^2} \approx 0.057 \text{ kg/s}$$

i.e. $\frac{\delta m_s}{m_s} \approx 1.6 \%$

It is important here to perform a direct differential temperature measurement of ΔT_{ev} instead of taking the difference between T_{in} and T_{out} . Not only the differential device is more sensitive, but the dominant sensitivity factor in the above formula, which is related to ΔT_{ev} , would appear twice if the absolute temperatures were used:

the overall effect would approximately double $\frac{\delta m_s}{m_s}$.

Isentropic Power across the Turbine

The isentropic power across the turbine, all the way to the diffuser exit, is defined as $P_{ise} = m_s \Delta h_{ise}$. P_{ise} is an abstraction which is used in evaluating the performance of the turbine, via its efficiency.

While information on the steam mass flow rate was provided above, the nominal value of the isentropic enthalpy drop Δh_{ise} is 83.84 kJ/kg, corresponding to turbine inlet and outlet pressures of 2611 and 1383 Pa, respectively, and thermodynamic properties linearly interpolated from the ASME Steam Tables (1977). Thus, the baseline value of P_{ise} is 296.45 kW. More specifically, we have:

$$\Delta h_{ise} = h_1 - \{h_{f2} + \frac{s_1 - s_{f2}}{s_{fg2}} h_{fg2}\}$$

In the above formula, subscripts 1 and 2 define the turbine inlet (saturated steam) and the diffuser outlet (two-phase mixture), respectively. The isentropic nature of Δh_{ise} is reflected in the presence of s_1 (instead of the real, unknown s_2) in the steam

quality $x_{ise} = \frac{s_1 - s_{f2}}{s_{fg2}}$.

State 1 variables depend upon the measurement of the turbine inlet pressure p_1 , while state 2 variables are functions of the turbine outlet pressure p_2 . As mentioned above, all components of Δh_{ise} are read, for example, from the ASME Steam Tables (1977), with the appropriate pressure as an entry. Thus, we have:

h_1 (kJ/kg)	s_1 (kJ/kg-K)	s_{fg2} (kJ/kg-K)	s_{f2} (kJ/kg-K)	h_{fg2} (kJ/kg)	h_{f2} (kJ/kg)
2541.43	8.628	8.681	0.1775	2473.66	49.51

OC-OTEC Vacuum Pumps Life Cycle Tests

Appendix B

Neglecting the influences of p_{ev} , T_{in} and T_{out} upon m_s , as they proved to be negligible, the two measured quantities affecting P_{ise} through m_s are Q_{ww} and ΔT_{ev} :

$$\frac{\partial P_{ise}}{\partial \Delta T_{ev}} = \frac{\partial P_{ise}}{\partial m_s} \frac{\partial m_s}{\partial \Delta T_{ev}} = \Delta h_{ise} \frac{\partial m_s}{\partial \Delta T_{ev}} \approx 84.796 \text{ kW/K}$$

$$\frac{\partial P_{ise}}{\partial Q_{ww}} = \frac{\partial P_{ise}}{\partial m_s} \frac{\partial m_s}{\partial Q_{ww}} = \Delta h_{ise} \frac{\partial m_s}{\partial Q_{ww}} \approx 489.43 \text{ kJ/m}^3$$

Moreover, the other two measured parameters used to determine Δh_{ise} are p_1 and p_2 . From the ASME Steam Tables (1988), we obtain (in SI units):

$\frac{\partial h_1}{\partial p_1}$	$\frac{\partial s_1}{\partial p_1}$	$\frac{\partial s_{f2}}{\partial p_2}$	$\frac{\partial s_{fg2}}{\partial p_2}$	$\frac{\partial h_{f2}}{\partial p_2}$	$\frac{\partial h_{fg2}}{\partial p_2}$
1.15×10^{-2}	-1.33×10^{-4}	1.65×10^{-4}	-4.34×10^{-4}	4.69×10^{-2}	-2.70×10^{-2}

(the above values are linear slope coefficients between 2600 and 2800 Pa for subscript 1 parameters, and between 1300 and 1400 Pa for subscript 2 parameters)

On the other hand, we have:

$$\frac{\partial P_{ise}}{\partial p_1} = \frac{\partial P_{ise}}{\partial \Delta h_{ise}} \frac{\partial \Delta h_{ise}}{\partial p_1} = m_s \left\{ \frac{\partial h_1}{\partial p_1} - \frac{h_{fg2}}{s_{fg2}} \frac{\partial s_1}{\partial p_1} \right\} \approx 0.1747 \text{ kW/Pa}$$

$$\frac{\partial P_{ise}}{\partial p_2} = \frac{\partial P_{ise}}{\partial \Delta h_{ise}} \frac{\partial \Delta h_{ise}}{\partial p_2}$$

$$= m_s \left\{ -\frac{\partial h_{f2}}{\partial p_2} + \frac{s_{fg2} \left\{ (s_{f2} - s_1) \frac{\partial h_{fg2}}{\partial p_2} + h_{fg2} \frac{\partial s_{f2}}{\partial p_2} \right\} + \{ h_{fg2} (s_{f2} - s_1) \} \frac{\partial s_{fg2}}{\partial p_2}}{s_{fg2}^2} \right\}$$

$$\approx -0.3323 \text{ kW/Pa}$$

With $\delta Q_{ww} = 0.0091 \text{ m}^3/\text{s}$ (FE-10), $\delta \Delta T_{ev} = 0.02^\circ\text{C}$ (TdT-11), $\delta p_1 = 5.4 \text{ Pa}$ (PT-32), $\delta p_2 = 3 \text{ Pa}$ (PT-33), and the above sensitivity coefficients, we can write:

$$\delta P_{ise} = \sqrt{\left(\frac{\partial P_{ise}}{\partial Q_{ww}} \delta Q_{ww} \right)^2 + \left(\frac{\partial P_{ise}}{\partial \Delta T_{ev}} \delta \Delta T_{ev} \right)^2 + \left(\frac{\partial P_{ise}}{\partial p_1} \delta p_1 \right)^2 + \left(\frac{\partial P_{ise}}{\partial p_2} \delta p_2 \right)^2}$$

$$\text{i.e. } \delta P_{ise} \approx 4.96 \text{ kW and } \frac{\delta P_{ise}}{P_{ise}} \approx 1.7 \%$$

Overall Fraction of Steam Condensed

The formula for the fraction of steam condensed by the direct-contact condenser (DCC) is:

$$F = \frac{\rho_w c_p Q_{cw} \Delta T_{DCC}}{h_{fg}} \frac{1}{\chi m_s}$$

F has a nominal value of 0.99. Q_{cw} is the volumetric flow rate of cold seawater into the DCC, and has a baseline value of $0.4031 \text{ m}^3/\text{s}$, whereas the nominal cold seawater should be evaluated at an average temperature across the DCC. This ratio, however, is formally similar to the expression for m_s , illustrated earlier (we simply deal here with the DCC instead of the evaporator). It was shown that only the measurement of Q_{ww} and ΔT_{ev} influenced the uncertainty of the derived parameter m_s . Similarly, it is clear that only Q_{cw} and ΔT_{DCC} will practically affect the first ratio in the definition of F. Thus, we can immediately write:

$$\frac{\partial F}{\partial Q_{ww}} = -\frac{F}{Q_{ww}} \approx -1.62 \text{ s/m}^3$$

$$\frac{\partial F}{\partial \Delta T_{ev}} = -\frac{F}{\Delta T_{ev}} \approx -0.281 \text{ K}^{-1}$$

$$\frac{\partial F}{\partial Q_{cw}} = \frac{F}{Q_{cw}} \approx 2.46 \text{ s/m}^3$$

$$\frac{\partial F}{\partial \Delta T_{DCC}} = \frac{F}{\Delta T_{DCC}} \approx 0.186 \text{ K}^{-1}$$

On the other hand, χ is a function of turbine inlet and outlet pressures:

$$\chi = \frac{s_2 - s_{f2}}{s_{fg2}}$$

Since the entropy s_2 is unknown, χ is estimated by means of the baseline turbine efficiency η_t , defined as:

$$\eta_t = \frac{h_1 - \{h_{f2} + \chi h_{fg2}\}}{h_1 - \{h_{f2} + \frac{s_1 - s_{f2}}{s_{fg2}} h_{fg2}\}}$$

$$\text{Thus, } \chi = \frac{h_1 - h_{f2}}{h_{fg2}} (1 - \eta_t) + \eta_t \frac{s_1 - s_{f2}}{s_{fg2}}$$

Baseline values for χ and η_t are 0.98 and 0.84, respectively. Evaluating the sensitivity of F to p_1 and p_2 , via χ , is not easy because η_t also depends upon these pressures in a not straightforward way. 'Pretending', however, that χ can be replaced by χ_{ise} , in estimating the partial derivatives of steam quality upon turbine inlet and outlet

pressures, shows that the order of magnitude of $\frac{\partial F}{\partial p_1}$ and $\frac{\partial F}{\partial p_2}$ is probably too small, of the order of 10^{-5} Pa^{-1} , to affect the uncertainty on F. Therefore, with previously used instrument uncertainties and $\delta Q_{cw} = 0.0061 \text{ m}^3/\text{s}$ (FE-20):

$$\frac{\delta F}{F} = \sqrt{\left(\frac{\partial F}{\partial Q_{ww}} \delta Q_{ww}\right)^2 + \left(\frac{\partial F}{\partial \Delta T_{cv}} \delta T_{cv}\right)^2 + \left(\frac{\partial F}{\partial Q_{cw}} \delta Q_{cw}\right)^2 + \left(\frac{\partial F}{\partial T_{DCC}} \delta T_{DCC}\right)^2}$$

i.e. $\frac{\delta F}{F} \approx 2.2\%$

References

ASME Steam Tables in SI (Metric) Units for Instructional Use, ASME, New York, 1977.

Benedict, R. P., Fundamentals of Temperature, Pressure and Flow Measurements, John Wiley & Sons, New York, 1984.

Miller, R. W., Flow Measurement Engineering Handbook, McGraw-Hill, New York, 1983.

Oney, S. K., and H. J. Krock, "Noncondensable gas composition and outgassing rates in the open-cycle OTEC system with direct-contact condensation", Proc. Int. Conf. on Ocean Energy Recovery, ASCE Pub., 1989.