PROJECT TITLE: Optimizing Heat Recovery Systems for Power Generation in Rural Alaska

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PROJECT OBJECTIVE: The objective of this project is to conduct laboratory performance test on an ORC system and performance and economic comparison of two different ORC systems in capturing waste heat from diesel generators for rural applications.
TABLE OF CONTENTS

Cover Page ........................................................................................................... 1
Table of Contents .............................................................................................. 2
Introduction ....................................................................................................... 3
Current Project Status and Expected Delivery Date ....................................... 3
  Project Time Line .......................................................................................... 3
  Procurement ................................................................................................. 3
  Test System Modeling and Simulation Methodology ..................................... 4
  Current Test System Modeling and Simulation Status ................................... 8
Installation and Instrumentation ....................................................................... 12
Reliability Testing ............................................................................................ 12
Performance Testing ......................................................................................... 12
Comparison in Performance and Economic Impact between the 50 kW and 250 kW ORC Systems .............................................................. 12
Report Writing and Dissemination ................................................................. 12
Major Progress of This Quarter ......................................................................... 13
Future Work ...................................................................................................... 13
References ....................................................................................................... 14
Appendix I: Experimental Building Floor Plan .............................................. 15
Appendix II: Expressions for Single Phase and Two Phase Heat Transfer Coefficient of Fluids in Plate Heat Exchangers ....................................................... 16
INTRODUCTION:
The objective of this project has three folds. The first is to improve the efficiency of the diesel power plant of a village by 10%, which also results in a reduction of CO₂ by 2.2 lb for each gallon of diesel fuel saved, through the use of a 50kW Organic Rankine cycle (ORC) system to recover waste heat contained in diesel engine jacket water and exhaust. The second is to evaluate feasibility, operation and maintenance requirements, and payback time of applying the ORC system on the village power plant mentioned above. The third is to develop guidelines for ORC system selection, operation, and maintenance and to evaluate the potential impact of applying waste heat ORC systems on rural Alaska economy, fuel consumption, and green house gas reduction.

CURRENT PROJECT STATUS AND EXPECTED DELIVERY DATE:
The approach of this project includes eight parts, which are procurement, modeling and simulation, installation and instrumentation, durability test for system reliability, controlled test for system performance verification, performance and economic comparison of two ORC systems (one is 50kW and the other is 250kW), and report writing and dissemination. Details of the expected delivery date, tasks of each part and its current status are listed in the following paragraphs.

In the first report (June 2010), the seventh task was to conduct performance and economic comparison of the ORC system and an ammonia-water system. Since then a technical problem of the ammonia-water system has been identified by the manufacturer, which might delay indefinitely the delivery time and in turn the test of the ammonia water system and the progress of this project. Therefore, task seven has been modified to the conduction of performance and economic comparison of the Electatherm (ET) 50kW ORC unit and a Pratt & Whitney (P&W) 250kW ORC unit. This modification will not alter the objectives of this project. The P&W 250kW unit, purchased by Cordova Electric Association with funding through the Alaska Energy Authority (AEA), is expected to be installed early next year and to be the first ORC system installed and operated using waste heat from a rural (3.6MW) Alaskan diesel generator. A 50kW capacity ORC system, which uses screw expander, belongs to the category of emerging technology and is still in testing stage. A 250kW capacity ORC system, which uses a radial turbine, belongs to the category of well developed technology for field application, although primarily using geothermal fluids rather than industrial waste heat as heat source. The modification could also provide added benefit of allowing us using the result obtained from data collection and analysis of the 250kW unit to future feasibility studies of the system to other larger rural Alaskan villages and comparing performance and economic data between a well developed, large ORC unit and an emerging small ORC unit, which is ultimately more appropriate for application in numerous medium sized villages around the state.

Project Time Line:
The expected delivery date for the ORC unit and related equipment/supplies is December 2010. Purchase Orders from both UAF and TCC were issued in summer 2010. The 2nd Quarter Progress Report (June 2010) listed the delivery date as summer/fall 2010; the delay in product delivery is due to the vendor’s production schedule. Project staff do not believe the delay will have a negative effect on the proposed workplan.
Procurement:

Major equipment, which are the devices absolutely needed for testing and measurement, include the 50kW ORC system, a hot water boiler (2.4MMBtu), a heating flow pump set (250GPM), a cooling tower (3.0MMBtu), a cooling flow pump (375GPM) set, two Btu meters, and a load bank. Purchase orders have been sent out and devices will flow in gradually. Final check with the engineers of the ORC system manufacturer has been conducted to confirm the appropriateness of the devices to be purchased for achieving the project objective.

Due to the property of ammonia-water working fluid, an ammonia-water system is expected to be able to take the advantage of high temperature heat source to improve its performance. Since the ammonia-water system is no longer involved in this project, a high temperature boiler may become unnecessary for this project. The new boiler recommended has an output water temperature of 235°F instead of 300°F as mentioned in the previous report. Therefore, other components (e.g. pump) selected for the heating loop do not need to have the capability of withstanding high temperature and high pressure accordingly. Considering cooling side of the ORC system, some popular applicable cooling devices include dry cooler, radiator, and cooling tower. Based on cost and operation power requirement, an outdoor cooling tower with an in-door storage tank is selected. In order to reduce power consumed to operate the ORC system at partial loads, variable frequency drives are selected to reduce power needed to operate the pumps and the cooling fan. Btu meters and Watt meters are used for direct power measurements. The Btu meter may also provide information of flow rate data. Load bank is used to receive power generated from the 50kW ORC system.

Components of the piping systems, which connect the boiler to the ORC system and the cooling tower to the ORC system, will be designed and purchased after the arrival of all the major equipment (expected schedule is December 2010). On shelf measurement devices (e.g. measurement devices for temperatures, pressures, and devices related to electrical power measurement), will also be purchased around the same time. In addition, installation and pipe and control system designs will be consulted with local experts for the concerns of regulations and safety.

Test System Modeling and Simulation Methodology:

The purpose of modeling and simulation is to make the model a tool for selecting village diesel engines by matching performance characteristics, load pattern, and environment conditions of the diesel engine to the performance of the ORC system. The model will also provide information for optimal application of the ORC system for efficiency and net gain in power including generated power and parasitical power.

The model includes three components: heat source loop, heat sink loop, and the ORC system. Fluid used in heating and cooling loops is water.

HEAT SOURCE

The physical heat source loop is expected to include a boiler, a VFD pump, and pipes and fittings. Other components for measurement and control are also included. Through the pipes, the hot fluid interacts with the heat source heat exchanger of the ORC system. The loop has controllability in temperature and flow rate control of the boiler output. Information of boiler natural gas consumption and pump power consumption corresponding to the operation
condition is also included in data to be collected. The model constructed includes all the function features of the expected heating loop and storage of all the data related to the data to be collected from the physical heating loop. In addition, the model can be easily modified to cope with different types of heat source and components to be adopted.

HEAT SINK
The physical heat sink loop includes a cooling tower (with a VFD cooling fan), a VFD pump, heat sink pipes and fittings, and needed measurement and controls devices. Through the pipes, the cooling fluid interacts with the condenser of the ORC system. The loop has controllability in temperature and flow rate of the cooling fluid. Information of pump power consumption, water consumption, and cooling fan power consumption corresponding to the operation condition is also included in data to be collected. The model constructed includes all the function features of the expected cooling loop and storage of all the data related to the data to be collected from the physical cooling loop. In addition, the model can be easily modified to cope with different types of heat sink and components to be adopted.

ORC SYSTEM
The physical ORC system to be used for this project is an integrated unit, for which to conduct tests on individual components, without modifying the system (modifying the system may results in losing warrantee), may not be possible. Also detailed engineering information of individual components may not be available for the concern of proprietary. In other words, the ORC system may need to be considered a black box of which operation characteristics cannot be altered by the user. A general ORC unit will include at least a pump, an evaporator, a heat to power converter, and a condenser. Other components needed in modeling will depend on the versatility of the physical ORC system to be modeled. The known property of the ORC system to be used includes components of a Rankine Cycle system with working fluid flow rate control for quality adjustment of the working fluid entering the screw expander. Working fluid used for this system is R245fa.

Considering ORC system reliability test and tests of ORC system performance related to system power generation under different heat source and sink conditions, detailed engineering data of individual components may not be needed. For investigation of optimal net power generation (including power generated by the ORC system and parasitic power consumed by cooling loop and/or heating loop), detailed engineering data may also not be needed. Therefore to achieve the objectives of this project listed in the section of Introduction, detailed engineering data of the ORC system components may not be critical. However, if the purposes of the tests include verification of the ORC system performance (e.g. optimal performance constantly) as claimed by the manufacturer and provision of comments to the design of the system, detailed engineering data of components will then become required.

For this project, the plan of the modeling may be divided into two to three stages depending on how feasible to reverse-engineering the performance parameters of the ORC system. The first stage is to model the ORC system using simple parameters for system components with quality and flow rate control of the working fluid. The purpose of this stage is to qualitatively understand the effects of operation conditions of heat source and heat sink on ORC system performance and the results will be used for test planning. The second stage is to fit the system parameters using data obtained from a limited number of experimental cases (i.e. heating and cooling conditions). If system simulation results obtained using fitted values of
system parameters can qualitatively match experimental results, but without appropriate accuracy, for extra operation conditions, a more complex model will be constructed for detailed modeling as the third stage. In other words, if reverse engineering of ORC system parameters shows feasibility, more accurate modeling effort will be conducted.

Stage One:
The simple model will include an evaporator, a screw expander, a condenser, and a VFD pump. The model allows the quality of the working fluid entering the expander adjustable via varying the flow rate of the working fluid for optimal ORC system performance. The results will be used for test planning. The expander is modeled by a single efficiency at this moment and will be modified as more information is available in publications and through experimental data, the pump is modeled with varying efficiency based on the operation condition of the pump, and the evaporator and condenser are modeled by their respective flow and heat transfer properties and heat transfer areas. Currently the evaporator is modeled with two sections; one of liquid and the other of liquid and vapor mixture. The condenser is modeled as a single section unit. If heat losses to the atmosphere are found significant, heat losses will also be included in the model.

Ranges of values of parameters used for simulation are based on specifications of the components (e.g. boiler output limits, pressure limit of ORC system), properties of the fluids (i.e. heating, cooling, and working fluid), asymptotes performance of sub-components (e.g. heat exchanger performance versus flow rate), etc. Known limits include maximum pressure of the ORC system (150psi), heat source temperature (235F), heat source capacity (2.4MMBtu), flow rates of pumps (250gpm for heating, 375gpm for cooling), and cooling sink capacity (3.0MMBtu recommended by an ORC engineer). Values of ORC components parameters adopted from publications (very limited data available) and conventional application practice [1] for system simulation include expander efficiency (e.g. 0.75), pump efficiency (e.g. 0.75), heat transfer coefficient of evaporator (e.g. 1500 W/m^2-K or 265Btu/ft^2-F), evaporator area (e.g. 100ft^2), heat transfer coefficient of condenser (e.g. 1400 W/m^2-K or 247Btu/ft^2-F), and condenser area (e.g. 200ft^2). Some of the values will be adjusted based on the match between the simulation results and the published and experimental data. Heat exchanger simulation model are based on standard practice [2, 3].

Since working fluid (R245fa) property will affect the ORC system performance and temperature and pressure limits to be used for testing, some of the physical properties are obtained and listed in Table 1 for reference. Detailed property of R245fa can be found in NIST document. Results of stage one simulation include system performance (e.g. net optimal efficiencies, expander power) of the ORC system, net optimal efficiencies of the test system, economic impacts, etc. as functions of operation parameters of heat source and heat sink. Results also include effects of sizes of heat exchangers and efficiencies of expander and pumps on system performance. The first stage results will be used to help design test plan of the ORC system.

<table>
<thead>
<tr>
<th>Safety</th>
<th>Vaporization Heat (1atm.)</th>
<th>Boiling T.(1atm.)</th>
<th>Critical Point</th>
<th>Saturation Slope</th>
<th>ODP*</th>
<th>GWP** 100 year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Non-Flammable</td>
<td>197.5 Kj/Kg (355.5 Btu/lb)</td>
<td>14.6C (58.3F)</td>
<td>154C (309.2F)</td>
<td></td>
<td>-0</td>
<td>1020</td>
</tr>
</tbody>
</table>

*Ozone depletion potential  
** Green house warming potential
Stage Two:
The second stage is to use measured data from experiment to fit the values of system parameters of the first stage model. Data to be used for fitting include inputs and outputs of the testing system. Inputs include, at least, inlet temperatures and flow rates of the heat source and heat sink heat exchangers. Outputs include, at least, outlet temperatures of heat exchangers and electrical power generated from the ORC system, of which the effect of mechanical to electrical power conversion efficiency is included. If other measured data become available (i.e. obtained from meters come with the ORC system or meters which are allowed to installed to the ORC system), they will also be used to fit the values of the parameters.

The fitted parameters values will then be used for model simulation of the test system under extra operation conditions of heat source and heat sink. The outputs will be compared with experimental data under the same operation conditions to investigate the discrepancies and to qualitatively determine the feasibility of modeling process. If the model is qualitatively feasible, but results predicted need to be improved, a more detailed model will then be adopted as the third stage of the simulation process.

Stage Three:
A more detailed model of an ORC system may need more than just four major subcomponents (i.e. expander, condenser, evaporator, and pump) as used for stage one modeling. Pipes and control elements (e.g. valves, fittings, and expansion tank) may affect heat transfer and pressure distribution and distribution of liquid and vapor in evaporator, condenser, and expansion tank. Therefore, pipes and control elements may affect performance of working fluid in the system and need to be included in system modeling. In addition, each of the sub components may be further divided into detailed elements to get better simulation results.

An expander may be further divided into its function elements, which include pressure drop element as nozzle effect, vapor cooling down element for working fluid before entering into the screw channel, isentropic expansion element accounting for the working fluid expansion in the channel between the screws, further expansion element due to ambient pressure, leakage element for leakage of working fluid from high pressure region to low pressure region, and existing cooling down element. In addition to the six elements, mechanical loss due to friction also needs to be considered for obtaining net expander work.

Inside the condenser, working fluid may experience three different conditions: vapor phase, mixed phase, and liquid phase. To better represent the performance of the condenser using simple mathematical model, a multi-section model may be needed with each section having its own constant heat transfer properties to approximate the heat transfer behavior of the particular zone. The spaces occupied by liquid, mixture, and vapor may be estimated using the amount of heat transferred. Pressure drop element is also needed for the condenser simulation. Similar to condenser, evaporator can be modeled using the same modeling method.

Pumps used in the testing system can be modeled using their performance characteristics given in the respective manuals and experimental data. Pipes and pipe control units can be
modeled using standard method recommended in handbooks for pressure drops. Heat transfer effect of the pipe may be less critical, if all the pipes are short and insulated properly as for the expected experimental setup.

If the third stage is conducted, the model parameters will be fitted and extra simulations will be performed and results will be used to verify the model.

Current Test System Modeling and Simulation Status:
This quarter, the first stage of test system modeling has been completed and the effects of flow properties of heat source and heat sink on net efficiencies of the ORC system and the test system (i.e. including parasitical power consumptions through heating loop and cooling loop) have been obtained through simulation.

The Organic Rankine Cycle (ORC) used in this simulation model is shown in Figure–1. The working fluid used in this simulation model is refrigerant R–245fa. The saturated liquid refrigerant from the condenser is pumped at high pressure to pre–heater. In pre–heater the refrigerant is heated to saturated liquid state and this saturated liquid then goes to evaporator. In evaporator the saturated liquid is heated to the required saturated liquid–vapor quality. This high pressure liquid–vapor mixture is converted to low pressure liquid–vapor mixture (to the condenser pressure) using a screw expander which is connected to the generator to produce power. The low pressure refrigerant from the screw expander is cooled to the saturated liquid condition in condenser which is again pumped back to pre–heater and the cycle continues.

The above mentioned system model has three major components: the heat source loop, the heat sink loop and the ORC system. In the heat source loop the heating fluid may be from engine jacket water or 50/50 glycol/water mixture for exhaust heat exchanger or both combined. In the heat sink loop the cooling fluid may be from cooling tower, radiator, or large water body (may be from nearby river or lake).

The modeling and simulation is a continuous process. As more system component information is available from experiments and open literature, the model will be updated to give more realistic and accurate results. The system component information such as for screw expander, boiling and condensing heat transfer coefficient of refrigerant in evaporator and condenser etc. is not available in open literature and need to be obtained from experimental analysis. As this data will be obtained from the experimental analysis will be used to tune the model so that in future it can be applied to any waste heat source for economic and feasibility analysis of the ORC system.

In this simulation basically five system parameters are being controlled they are inlet temperature and flow rates for heat source and heat sink input loops and the quality of the refrigerant inlet to the expander. The quality of refrigerant inlet to the expander is controlled to optimize the power output and efficiency of the ORC system for a given heat source and heat sink condition.
Figure–1: Schematic of Organic Rankine Cycle System

The system simulation has been performed for different screw expander inlet refrigerant quality (from 0.75 to 0.95) for given heat source and heat sink inlet conditions. The heat source and heat sink inlet conditions are flow rate and inlet temperatures of respective fluids (here water is considered for both heat source and sink). In the simulation following assumptions were made:

1. All the heat exchangers i.e. evaporator, pre–heater and condenser, are 100% efficient.
2. The quality of refrigerant out of the evaporator in the ORC system is controlled.
3. The quality of liquid out of pre–heater and condenser are saturated liquid.
4. The isentropic efficiency of screw expander and pump (within the ORC system) are taken to be constant at 78% and 70% respectively.

The assumptions can be modified easily, once the true performance characteristics of components are known from published or experimental data.

In simulating the ORC system performance explicit formulae for heat transfer coefficients of refrigerant on one side of the heat exchanger and water on other side of the heat exchanger should be known. Generally the heat transfer coefficient of a fluid is expressed in terms of its thermodynamic and transport properties. The heat transfer coefficient also depends on the geometry of heat exchanger, material of construction etc.

All the heat exchangers considered in the present case are plate heat exchangers (PHE). A widely accepted expression for heat transfer coefficient of single phase fluids in a plate heat exchanger is given by Muley and Manglik [4]. In ORC system this expression is used for calculating heat transfer coefficient of hot water and cold water in evaporator and condenser respectively and heat transfer coefficient of hot water and refrigerant in pre–heater. In all the above cases of calculating heat transfer coefficient the fluid thermo–physical properties were taken at average fluid temperature.
The expression for heat transfer coefficient of evaporating refrigerant liquid-vapor mixture in the evaporator is given by Ayub [5]. The expression for heat transfer coefficient of condensing refrigerant liquid-vapor mixture in the condenser is given by Selvam et al [6]. All the above expressions are presented in the appendix.

SIMULATION CASE STUDY
The constructed ORC system model has been used to simulate an example 50kW ORC system with the system parameters listed in Table–IIA of Appendix-II. Figure–2 and Figure–3 show the plots for efficiency vs. expander inlet quality. These figures also show the effect of parasitic power and heat sink supply temperature on system efficiency. The parasitic power is the power needed to pump the heat source and heat sink fluids to/from the ORC system. As the heat sink supply temperature decreases (in this case from 21°C to 5°C), to remove the same amount of heat from the condensing refrigerant in the condenser less amount of cooling fluid is required thereby decreasing the parasitic power and increasing the efficiency of the system. This may be one of the advantages of using the ORC system during the winter months.

The effect of heat source temperature on the ORC system efficiency can also be observed in Figure–2 (93°C) and Figure–3 (121.11°C). As the heat source temperature increases the efficiency of the ORC system increases. Here in simulating the system for different heat source temperatures the screw expander inlet pressure conditions (or evaporator exit conditions) were different though all other system parameters remained same (i.e. condenser pressure, expander and pump efficiencies, heat source flow rate etc). For heat source temperature of 93°C case the expander inlet pressure was 6.95 bar and for 121.11°C case the pressure was 15.7 bar. As more work is produced by the expander when it goes from high pressure at expander inlet to same condenser pressure, this may be the reason for increase in system efficiency for different heat source temperatures.

To obtain good match between simulation and experimental results, improvement in performance characteristic expressions of critical components may be needed. These expressions may be obtained from fitting parameter values which describe the performance characteristics of components using published and experimental data. According to the simulation, recommended data to be measured from the test system for individual component parameters are inlet and outlet temperatures, pressures, qualities, and flow rates of the major components (i.e. evaporator, condenser, pre-heater, expander, and pumps).
Figure–2: Efficiency of ORC system with varying screw expander inlet quality for heat source temperature of 200°F (93°C).

Figure–3: Efficiency of ORC system with varying screw expander inlet quality for heat source temperature of 250°F (121.11°C).
Installation and Instrumentation:
The task of installation and instrumentation include installation of the complete testing system (i.e. ORC system, heat source system, cooling system, electrical power consumption device, and measurement and data acquisition devices). Detection, rectification, and modifications (if it is needed) are also part of this task. Since regulations, safety, and reliability are important to this project, experts in related areas will be consulted for piping and control system design and installation.

Currently, the space for the experiment has been located. Action in installation will start after some of the equipment being shipped in and their details becoming available. The experiment space floor plan is given in Appendix I.

Reliability Testing:
Reliability test, which is a 600-hour test, considers the durability of the system for long term operation. The test results will include details of malfunctions, repairs, maintenance performed, and other major system performance (e.g. fuel consumption, performance stability, power generated) during the test. The results will also include technologies needed for applying these systems.

Performance Testing:
Performance test considers the performance of the ORC system and the performance of the test system (i.e. ORC system, heating loop, and cooling loop) as functions of operation conditions of the heat source and heat sink. Performance of ORC system includes net system efficiency, system consistency, and more if more measurement devices are available. Test performance result includes detailed relationships between electrical output, parasitic power, and operation parameters of heat source and heat sink. This information may be used for the selection of appropriate diesel generators for this particular ORC system. The information may also be used to estimate electrical energy, which can be generated from a particular diesel generator with given load pattern and environmental conditions. If more measurement data (e.g. information of working fluid states along the ORC loop, power consumption of the working fluid pump) becomes available, the data can also be used for performance verification of the ORC system and design parameters recommendation.

Comparison in Performance and Economic Impact between the 50kW and 250kW ORC Systems:
Performance comparison will include discussions of efficiencies and powers generated of the two systems; net efficiencies and net powers generated of the systems with respect to different possible cooling systems; feasibilities of the systems for different sizes and types of heat sources (e.g. exhaust, jacket water); economic analyses (e.g. total benefit, cost return time) for different heat sources. Other discussions include issues related to installation, maintenance, operation, operation stability, technology requirement in applying these systems.

Report Writing and Dissemination:
Detailed information of all the above mentioned tasks will be covered in regular quarterly reports and final report. In addition, technical findings will be submitted to appropriate journals and conferences.
MAJOR PROGRESS OF THIS QUARTER:
A. Modification of one of the project tasks.
   The project task of “performance and economic comparison of the ORC system and the
ammonia-water system” has been modified to “performance and economic comparison of
the two (50kW and 250kW) ORC systems.” Reason for this modification is given in the
beginning of the section of “CURRENT PROJECT STATUS AND EXPECTED
DELIVERY DATE.” Both Cordova Electrical Association and AEA have been contacted
and agreement of allowing us conducting data collection and analysis of the 250kW ORC
unit was received.

B. Purchases of major equipment (i.e. a boiler, an ORC system, a cooling tower, a VFD
pump, two Btu meters, a dynamometer, etc.) have been confirmed. The equipment is
expected to arrive at Fairbanks in December 2010 or earlier. Contacts to local experts
(e.g. Fairbanks fire marshal, mechanical contractors) for consultation in areas (e.g. boiler
installation and startup, piping system design and installation) related to this project have
been started.

C. A preliminary computer program of the first stage modeling has been completed and
simulation result for a 50kW ORC system has been conducted. The computation process
has shown the data, which are related to the parameters of critical system components,
may also be the data needed from experimental measurement for data fitting for better
parameters values of system components. The program is also designed for flexibility for
future modification (e.g. modification of performance characteristics of system
components) and more complicated data fitting (e.g. applying published and experimental
data to fit appropriate values for components parameters).

FUTURE WORK:
A. Continuation in procurement.
   Purchase of pipes and control elements.
B. Installation and instrumentation.
   Consultation of reputable mechanical companies for installation and piping
design of critical components.
C. Scheduling for reliability testing and performance testing.
D. Continuation in system modeling and simulation.
   Development of methodology for second stage and third stage.
E. Analyses of system performance data and economic impact.
F. Continuation of report writing and dissemination.
REFERENCES


APPENDIX I
Experimental Building Space Floor Plan

Figure IA: Floor Plan
Detailed system lay out will be provided after arrival of major test system components
Appendix-II
Expressions for Single Phase and Two-Phase Heat Transfer Coefficient of Fluids in Plate Heat Exchangers

In calculating the heat transfer coefficients of fluids we need to know the physical parameters of the plate heat exchanger. Table–IIA gives the physical parameters of a typical plate heat exchanger taken from open literature [4] for a 50 kW ORC system.

Table–IIA: Plate Heat Exchanger Physical Parameters considered in the present simulation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate Width (w in m)</td>
<td>0.5</td>
</tr>
<tr>
<td>Plate Length (L_p in m)</td>
<td>1.1</td>
</tr>
<tr>
<td>Channel Spacing (m)</td>
<td>3.50E-03</td>
</tr>
<tr>
<td>Thickness of Plate (t in m)</td>
<td>6.00E-04</td>
</tr>
<tr>
<td>Cheveron angle (β in degrees)</td>
<td>45</td>
</tr>
<tr>
<td>Enlargement Factor (Φ)</td>
<td>1.29</td>
</tr>
<tr>
<td>Corrugation Pitch</td>
<td>7.00E-03</td>
</tr>
<tr>
<td>Equivalent diameter (D_e in m)</td>
<td>7.00E-03</td>
</tr>
<tr>
<td>Projected area per plate (A_p in m²)</td>
<td>0.55</td>
</tr>
<tr>
<td>Flow area on one fluid side (A_f in m²)</td>
<td>4.20E-02</td>
</tr>
<tr>
<td>Surface area on one fluid side (m²)</td>
<td>26.4</td>
</tr>
<tr>
<td>Hydraulic Diameter (D_h in m)</td>
<td>0.005426</td>
</tr>
<tr>
<td>Pressure drop in heat exchanger (MPa)</td>
<td>0.05</td>
</tr>
<tr>
<td>Plate Thermal Conductivity (K_p in W/m-K)</td>
<td>13.8</td>
</tr>
<tr>
<td>Plate Thermal Resistance (m²-K/W)</td>
<td>4.35E-05</td>
</tr>
<tr>
<td>Area of Evaporator (m²)</td>
<td>33.26</td>
</tr>
<tr>
<td>Area of Condenser (m²)</td>
<td>28.3</td>
</tr>
<tr>
<td>Area of Pre–Heater (m²)</td>
<td>15</td>
</tr>
<tr>
<td>Critical Pressure of R245fa (MPa)</td>
<td>3.651</td>
</tr>
<tr>
<td>Critical Temperature of R245fa (°C)</td>
<td>154.01</td>
</tr>
</tbody>
</table>

Heat Transfer Coefficient for Single Phase Fluids

The expression for convective heat transfer coefficient for single phase fluids in a plate heat exchanger is given by Muley and Manglik [4] and is read as

\[ h_f = \frac{(Nu \times K_f)}{D_e} \]  \hspace{1cm} (1)

\[ Nu = [0.2668 - 0.006967\beta + 7.244 \times 10^{-5} \beta] \times \frac{Re^{0.728+0.0543\sin((\eta/45)+3.7)}}{Pr^{1/3} (\mu / \mu_n)^{0.14}} \] \hspace{1cm} (2)

In the above equation \( K_f \) is thermal conductivity of fluid and the Reynolds number (Re) is based on equivalent diameter of the plate heat exchanger and is calculated as,

\[ Re = \frac{(G \times D_e)}{\mu} \] \hspace{1cm} (3)
Where $G$ is the mass velocity of the fluid, $D_e$ is the equivalent diameter and $\mu$ is the dynamic viscosity of the fluid. The mass velocity ($G$) is the ratio of mass flow rate of the fluid to flow area of the heat exchanger.

**Heat Transfer Coefficient for Evaporating Fluids**

The expression for convective heat transfer coefficient for evaporating fluids in a plate heat exchanger is given by Ayub [5] and is read as

$$h_{Eva} = 0.0675(K_f / D_e)[Re^2 h_{fg} / L_p]^{0.4124} (p / p_{cr})^{0.12} (65 / \beta)^{0.35} \text{[BTU/hr-ft}^2\text{-oF}]$$

(4)

In the above equation $K_f$ is the conductivity of the fluid, $h_{fg}$ is the enthalpy difference between outlet and inlet of heat exchanger, $L_p$ is length of the plate, $p$ is pressure and $p_{cr}$ is critical pressure. Reynolds number (Re) is based on equivalent diameter and its calculation is similar to the above equation for single phase fluids (Eq. (3)).

**Heat Transfer Coefficient for Condensing Fluids**

The expression for convective heat transfer coefficient for condensing fluids in a plate heat exchanger is given by Selvam et al [6] and is read as

$$h_{Cond} = \frac{(Nu_{Cond} \times K_f)}{D_h}$$

(5)

$$Nu_{Cond} = Ge_1 \frac{Re_{Eq} Ge_2 Pr^{1/3}}{G_{Eq}}$$

(6)

$$Ge_1 = 11.22 \left( \frac{p_{co}}{D_h} \right)^{-2.83} \left( \frac{\pi}{2} - \beta \right)^{-4.5}$$

(7)

$$Ge_2 = 0.35 \left( \frac{p_{co}}{D_h} \right)^{0.23} \left( \frac{\pi}{2} - \beta \right)^{1.48}$$

(8)

$$Re_{Eq} = \frac{G_{Eq} D_h}{\mu_f}$$

(9)

$$G_{Eq} = G_e [1 - x + x(\rho_f / \rho_g)^{0.5}]$$

(10)

$$G_e = \frac{m}{A_f}$$

(11)

Where $K_f$ is the thermal conductivity of the fluid, $p_{co}$ is plate corrugation pitch, $D_h$ is hydraulic diameter, $x$ is quality of fluid entering the condenser, $\rho_f$ and $\rho_g$ are the density of...
saturated liquid and vapor at condenser inlet condition. Here we need to observe that the Reynolds number (Re) is based on hydraulic diameter.