I, Vishakhdutt Ranade, hereby submit this original work as part of the requirements for the degree of Master of Science in Mechanical Engineering.

It is entitled:
Dynamic Modeling of Rankine Cycle using Arbitrary Lagrangian Eulerian Method

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Dynamic Modelling of Rankine Cycle using Arbitrary Lagrangian Eulerian Method

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Abstract

Thermoelectric power plants are often based on Rankine cycle where the steam is cooled using water from a lake or a river. This water is recirculated in a cooling tower where a portion of the cooling water is lost to evaporation. To reduce water consumption in thermoelectric power plants, there is an urgent need to develop efficient air-cooled condensers for Rankine cycles. To this end, understanding how the Rankine cycle performance changes with diurnal temperature variation is essential. In the present study, a computational model was developed to simulate the transient behavior of a Rankine cycle and its response to changing ambient air temperature. The computation model is based on the Arbitrary Lagrangian Eulerian method incorporating the merits of Lagrangian and Eulerian techniques aiming towards a higher computational efficiency while accurately tracking the H₂O mass moving through the boiler, turbine, condenser, and the pump. The model was developed using MATLAB and validated using available data. The model was first applied to simulate transient dynamics of a vapor compression cycle for which data were readily available in the literature. The validated model was then used to simulate a Rankine cycle. Parameters of interest for the computation included the delivery conditions for boiler, turbine, condenser and the resulting power output. Simulations were run with varying ambient temperature throughout the day. The maximum ambient temperature was varied to represent four locations in different regions of the United States. These results were compared to cycle operation under air pre-cooling which maintains a fixed maximum air temperature reaching the condenser. When the ambient air temperature increases, the condenser temperature and consequently its pressure increases. This results in a decrease in the turbine power output. This reduction in power output can be mitigated by maintaining the temperature of the air going to the condenser at the design condition. The increase in cycle efficiency obtained with air pre-cooling is plotted for four different locations. The developed computational model based on the
Arbitrary-Lagrangian-Eulerian method is able to accurately capture the transient variation in Rankine cycle performance with varying ambient conditions.
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Nomenclature

A  Interface Area (m²)                  Pmax  Maximum Array Pressure (kPa)

df Dryness Fraction Matrix             Pmin  Minimum Array Pressure (kPa)

e Normalized change in mass per       \bar{P}v Pressure Array 
   Iteration

H  Enthalpy Matrix (kJ/kg)             S    Entropy (kJ/kg)

h  Heat Transfer Coefficient (W/m²-K)  Sp. Vol Specific Volume (m³/kg)

HED Temperature Matrix (K)             t    Time-Step

i  Iteration Serial Number             Tmin Minimum Ambient Temperature (K)

j  Control Volume Index                V    Control Volume (m³)

L  Exchanger Pipe Length               Vdisc Discretized Control Volume (m³)

M  Mass Matrix                         W    Turbine Power(kW)

m Delivery Rate (kg/hr)                Z    Axial Length

P  Heat Exchanger Pressure (kPa)       \Psi  Tolerance Error Definition
Calculation Parameter

\( \partial m_{\text{exit}} \) \hspace{1cm} \text{Mass element at Pipe Exit}

\( \partial m_{\text{entry}} \) \hspace{1cm} \text{Mass Element at Pipe Entry}
1 Introduction

Thermoelectric power plants typically run on the Rankine cycle or Brayton/Rankine combined cycle. Figure 1 shows a simple Rankine cycle comprising of a boiler, turbine, condenser and a pump. Pump increases the pressure of the feed-water and supplies it to the boiler. Heat addition occurring in the boiler conditions the fluid to a delivery temperature and pressure subsequently driving the turbine to generate useful power. In the condenser following the turbine, the operating fluid is condensed. The condensed liquid then is supplied to the pump completing the cycle. Large amount of heat must be removed in the condenser. For a power plant with 200 MW of electrical output running with a 40% thermodynamic efficiency would require removal of 300 MW in the condenser. Most power plants in the United States utilize water from either a river or a lake for heat removal. To reduce water withdrawals from such sources, recirculating cooling tower based system are often utilized. In these systems, the water warmed by condensing steam is recirculated through a cooling tower. A significant portion of the water evaporates and regulates water temperature. This loss of water to evaporation has to be offset by make up water from the source. Such requirement of water in energy conversion and utilization is sometimes referred to as part of the “Energy-Water-Nexus.” (Hussey, Pittock., 2012)

These water requirements severely limit the feasible locations for building a new power plant. Preferred location of Rankine cycle power plants near large water bodies stems from this consideration. Also, the water requirements of existing power plants compete with those of agriculture, industrial, and residential use. To alleviate these problems, efficient air-cooled condensers are being developed. For widespread use of such condensers, limitations of this approach must be addressed. First, the ambient air temperature varies during the day. As the temperature increases, the condenser operating temperature and the corresponding saturation
pressure increases. This reduces the turbine output and power plant efficiency. To overcome this second-law limitation, air pre-coolers are being considered along with a thermal energy storage.

To this end, understanding how the Rankine cycle performance changes with diurnal temperature variation along with its performance with air pre-cooling is essential. In this thesis, a computational model was developed to simulate the transient behavior of a Rankine cycle and its response to changing ambient air temperature. The computation model is based on the Arbitrary Lagrangian Eulerian method incorporating the merits of Lagrangian and Eulerian techniques aiming towards a higher computational efficiency while accurately tracking the water mass moving through the boiler, turbine, condenser, and the pump. The primary focus is on modeling the flow and heat transfer in the condenser and the boiler. Four locations in the US; Providence, RI; Seattle, WA; Las Vegas, NV, Tampa, FL were considered to understand the cycle dynamics under different temperature excursions. Because the second-law limitation increases with the condenser and ambient temperature, ambient temperature variation in the month of July was considered.

The Arbitrary Lagrangian-Eulerian method is used to model the Rankine cycle in the present study. The flexibility and robust nature of the ALE method has been documented in a variety of applications ((Somerton et al., 1987), Plana-Fattori et al., 2013), (Zhang & Hisada, 2001). Also, dynamic modeling of Rankine power cycles has been carried out (Colonna & van Putten, 2007) & (van Putten & Colonna, 2007), (Liska, Shashkov et al., 2010). (Kapadia & Wolgemuth, 1984). The model in the present study was developed using MATLAB and the results were validated using data available in the literature. The model was first applied to simulate transient dynamics of a vapor compression cycle for which data were readily available in the literature. The validated model was then used to simulate a Rankine cycle. Parameters of
interest for the computation included the delivery conditions for boiler, turbine, condenser and the resulting power output. Simulations were run with varying ambient temperature throughout the day.

The environmental concerns targeting lower usage of fossil fuels and net higher efficiencies of the power generation is leading to a desire to utilize low-grade waste heat. To run a Rankine cycle at lower temperatures, working fluids other than water are being considered. In particular, R245fa, R123, CO2 (Wei, Lu, Lu, & Gu, 2008), (Gnutek & Bryszewska-Mazurek, 2001), (Chen, Lundqvist et al., 2006) are being considered in organic Rankine cycles. (Chen et al., 2006). While it is not part of the present study, the computational model developed in this study can be used to simulate such organic Rankine cycles as well.
Boiler
Pressure - 2 Mpa
Temperature - 630 K
Mass Flow Rate - 3536.265 kg/hr

Isentropic Turbine
Entry Pressure – 2 MPa
Exit Pressure – 30 kPa
Isentropic Work – 760 kW

Condenser
Pressure – 30 KPa

Centrifugal Pump
Exit Pressure 2 Mpa

Figure 1 - Rankine Cycle Schematic
2 Mathematical Model

2.1 Arbitrary Lagrangian Eulerian Method

The simulation of fluid flows is traditionally carried out using one of two methods – the Lagrangian method or the Eulerian method. In the Eulerian method, governing conservation equations for fluid flow are written in a stationary reference frame while in the Lagrangian formulation the reference frame moves with individual material particles down the flow path. The techniques carry their own merits and demerits. In a 1974 paper, foundations for a technique combining the merits of both methods were developed by Hirt et al. (Hirt, Amsden et al., 1974). This was termed as the Arbitrary-Lagrangian-Eulerian or ALE method. This method has since been used in a wide spectrum of physical simulation and analyses. For example, application of ALE method to lubricated contact bearing was studied by (Boman & Ponthot, 2004) with a purpose of calculation of lubricant forces flow velocities. The study mentions dividing the time step dividing into two phases, Lagrangian followed by mesh regeneration Eulerian phase. Algorithm used by (Boman & Ponthot, 2004), (Benson, 1989)) has served as a backdrop for the error calculation flowchart used in the present study elaborated in the further sections.

Ganesan and Tobiska (Ganesan & Tobiska, 2012b) utilized the continuum conservation equations that were slightly modified to account for the soluble surfactants. They were able to determine the change in surfactant concentration, kinetic energy and the mass of total surfactants with the experimental results and found good agreement. A step further in comparison of ALE method with the classical purely Eulerian and Lagrangian methods can be found in (Plana-Fattori et al., 2013) using a case study of starch-granule swelling. Purely Eulerian and the ALE approach were considered and compared using sensitivity tests on the mesh size. Applicability of the method to non-Newtonian fluid flows and turbulent flows is also well-established (Plana-Fattori
et al., 2013). The ALE method has been used and proven to be effective in numerous physics simulations apart from fluid flows over the classical methods (Rakotomalala, Joyot et al., 1993), (Ganesan & Tobiska, 2012a), (Plana-Fattori et al., 2013). ALE computation strategy is robust, optimizes accuracy and computational efficiency (Liska, Shashkov et al., 2010).

In the current study, the Arbitrary Lagrangian–Eulerian method has been utilized to model flow and heat transfer in heat exchangers with a computational mesh moving in space with time. Each time increment is divided in two steps. In the Lagrangian step, mass in each control volume is allowed to move. Temperature and pressure change is calculated based on the amount of heat transfer. In the second (Eulerian) step, the mesh is redistributed so that the interfaces on cooling and heating side match once again. The amount of mass changes in the control volumes. As the domain is re-meshed, some fluid leaves the domain and some enters the domain.

The current study is built on a former masters’ thesis at the University of Cincinnati (Miller, 2012). (Miller, 2012) used a purely Lagrangian model of simulation with mass elements representing the working fluid and the coolant. His thesis builds a model to simulate a vapor compression refrigeration system through its components. Considering minute control volumes and tracking them across the length of the condenser and evaporator components, a visualization of intermediate heat transfers and refrigeration levels achieved was presented. The intermediate simulation steps involved tracking of the control volumes and matching of corresponding heat transfer interfaces. The heat transfer interfaces do not necessarily match exactly at each iteration. This phenomenon requires a separate logic for calculating and distributing the contact areas. Keeping track of appropriate heat transfer interfaces at every time-step is a cumbersome task. In contrast, in a purely Eulerian method, the control volumes remain fixed while fluid enters and leaves each control volume in each time step. This approach is the easiest to implement but
change in pressure and mass are more difficult to predict accurately with such formulation. The Lagrangian approach can track the mass and pressure changes easily but the book keeping required at each step is more tedious. The present study utilizes the ALE method and where the onerous tasks of matching heat transfer interfaces in each iteration is eliminated. This makes the implementation of the method easier by combining the advantages of Eulerian and Lagrangian approaches.

Figure 2 - Lagrangian Non-Porous Mesh - Iteration ‘j’ (*Miller, 2012*)

Figure 3 - Lagrangian Non-Porous Mesh - Iteration ‘j+n’ (*Miller, 2012*)

Figure 4 - ALE Porous Mesh – Iteration ‘j’
2.2 Rankine Vapor Power Cycle

The current study considers the dynamics of Rankine Power cycle. The following assumptions were made to simplify the computation.

1. Fluid flow and heat exchange modelled in a single dimension
2. The heat exchangers assumed to be at a uniform pressure across the length
3. The working fluid and the heating/cooling medium assumed to be in thermal contact with each other by assuming the thermal resistance of the intermediate metal conducting surface to be negligible.
4. Turbine expansion process is considered isentropic.
5. Axial heat transfers among the control volumes was neglected

2.3 Control Volume Discretization

The computation method follows a path initiating with discretization of the component geometries. As explained earlier, the method follows an ALE approach. The approach calls for splitting the component into finite control volumes. Current study diverts from the one conducted by (Miller, 2012) at this point. As opposed to the cited study, the control volumes have steady dimensions throughout the study. The finite control volumes allow the fluids passage through their porous boundaries. The mass contained in the control volumes is ultimately a function of the prevalent pressure, enthalpy and the subsequent dryness fraction.

In the perspective of computation, each control volume is a ‘matrix element’ that acts as a register for instantaneous temperature, mass, dryness fraction and enthalpy values (Table 1 - Finite Control Volume Configuration - 'Matrix Element'). Instantaneous temperature is the governing property that is recalculated at each iteration. The mass, dryness fraction and enthalpy
form auxiliary values that undergo change governed not just by the temperature but also the persisting pressure in the heat exchanger.

The grid resolution of the model controls the tracking of fine progressive changes and phase transition planes as the control volume moves along the component length. Recent studies (Bendapudi, Braun et al., 2008) have shown a necessity of having at least 15 grid elements longitudinally to predict the phase transition plane with acceptable accuracy. The control volume schematic diagram (Figure 5 - Cycle Components- Control Volume Discretization Schematic) shows the system in a discretized state with fluid flow directions.
\[ V_{\text{disc}(i,j)} = V_{\text{heat exchanger}/J} \quad (2) \]

Equation 1 - Discrete Control Volume

Table 1 - Finite Control Volume Configuration - 'Matrix Element'

<table>
<thead>
<tr>
<th>Finite Control Volumes - Boiler Tube</th>
<th>HED(i)</th>
<th>HED(i+1)</th>
<th>HED(i+2)</th>
<th>HED(i+3)</th>
<th>HED(i+n)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Temperature</strong></td>
<td>H(i)</td>
<td>H(i+1)</td>
<td>H(i+2)</td>
<td>H(i+3)</td>
<td>H(i+n)</td>
</tr>
<tr>
<td><strong>Enthalpy</strong></td>
<td>df(i)</td>
<td>df(i+1)</td>
<td>df(i+2)</td>
<td>df(i+3)</td>
<td>df(i+n)</td>
</tr>
<tr>
<td><strong>Dryness Fraction</strong></td>
<td>M(i)</td>
<td>M(i+1)</td>
<td>M(i+2)</td>
<td>M(i+3)</td>
<td>M(i+n)</td>
</tr>
<tr>
<td><strong>Mass</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pressure 'P'</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
2.4 Energy Transfer

The energy transfer occurring between the heat source and the sink are a function of the fluid properties, heat transfer coefficients and the contact time between corresponding control volumes. As stated in the assumptions above, axial heat transfer between the control volumes is neglected. Only the influence of heat source/sink on the control volumes is considered thus limiting the heat transfer to single dimension.

\[ Q_{i,j} = h_{i,j} \times A_{i,j} \times |T_{source/sink} - T_{i,j}| \times t_{i,j} \]  (3)

In the computation, the heat exchanger fluid elements are progressively treated through the increase and decrease in individual enthalpies. Heat transfer and temperature change occurring in the time step is directly reflected into a change in enthalpy of the control volume. Since internal energy of the fluid is assumed to remain constant in the model, the fluid enthalpy forms one of the principal basis for assessing fluid quality.

The fluid elements that enter and exit the heat exchanger contain an inherent enthalpy reflective of their temperature, pressure and quality (4). The inherent enthalpy forms the reference value and new amounts of enthalpy are further added here. The enthalpy values form an integral part of the matrix element. For ease of calculation and better resolution of values, calculations deal with the specific enthalpy values of each control volume at the requisite conditions. The enthalpy change due to heat transfer is divided by the mass in the control volume to obtain the specific value (5).

\[ H_{entry} = f(P_{pump}, T_{pump}, q_{pump}) \]  (4)
\[ H_{i,j+1} = H_{i,j} + \frac{Q_{i,j+1}}{m_{i,j+1}} \]  

(5)

\[ H_{i,j} \geq f \left( P_{i,j}, T_{i,j}, 0, \bar{H} \right) \]  

(6)

The enthalpy addition algorithm is provided guards in order to prevent outputs contradictory to the natural values. The purpose of the boiler being addition of enthalpy and improving fluid quality, an unbounded enthalpy increase is permissible. However, in the condenser, enthalpy decrease occurs and care is taken to guard the enthalpy value from falling below a value corresponding to that of liquid state (6).

2.5 Mass Balance

The mass of the fluid in the heat exchanger and prevalent pressure are interlinked. The mass in the heat exchanger is the reference value and the calculation of pressure revolves around the same. The instantaneous value of the mass present in the heat exchanger is a simple accumulation of masses present in the control volumes across the length of the heat exchanger (7). Every iteration causes a change in heat exchanger mass due to the fluid entering at the pump delivery properties and the fluid leaving at the heat exchanger delivery values (8). The subscript\((j)\) noting the iteration changes on the either side of the equals sign conveys the mass transfer occurring (8).

\[ M_j = \int_1^L \partial m_{i,j} \]  

(7)

\[ M_{j+1} = M_j - \partial m_{exit,j} + \partial m_{entry,j} \]  

(8)
The amount of mass present in the control volumes is calculated by two methods. The convergence between the two values is the correction factor in pressure recalculation process explained shortly in the text. During the computation (6), the overall mass is calculated taking into account the impending pressure, individual enthalpy and resultant dryness fractions. Since control volumes do not change in size, an internal mass transfer between control volumes is imminent. As stated above, the control volumes are bounded by porous boundaries allowing for mass to flow through. This methodical step lends the Eulerian aspect to the calculation. Mass value arrived leveraging (8) acts as a constraining value. Achieving a balance between the values is a main aim of the method (9). A flow chart for the mass calculation algorithm is shown in Fig. 6.
Mass Calculation Algorithm

\[ M_{\text{known},j} = M_{j-1} - \partial m_{\text{exit},j-1} + \partial m_{\text{entry},j-1} \]

\[ \text{Sp. Vol}_{i,j} = f \left( P_j, H_{i,j}, q_{i,j} \right) \]

\[ \partial m_{i,j} = \frac{V_{\text{control volume } i,j}}{\text{Sp. Vol}_{i,j}} \]

\[ M_{P,j} = \int_{1}^{L} \partial m_{i,j} \]

Balance \((M_{\text{known},j} = M_{P,j})\)

Figure 6 - Mass Calculation Algorithm
2.6 Pressure Calculation

The pressure across the length of the heat exchanger is assumed to be uniform. Addition of heat and changes in mass and fluid composition results in pressure changes at each iteration. The increased enthalpy and the corresponding change in the dryness fraction alter the composition of control volume elements. A progressive change in the control volume composition over time compels the pressure to rise. The pressure and temperature are allowed to rise for the boiler until the preset delivery values.

To describe the state of the heat exchanger at the end of each iteration and take the computation forward, pressure recalculation through a predefined algorithm is necessary. As stated previously, the mass and pressure calculation are interlinked. The pressure calculation is carried out across the constraint of the fluid mass present in the heat exchanger. The net mass of the fluid in the heat exchanger undergoes a change at every iteration. The fluid elements added at the pump delivery properties and the departing fluid at delivery pressure induce a change in the net mass and overall composition.

The algorithm designed to calculate pressure takes into account the mass constraint, control volume properties, and the fluid properties to arrive at a pressure value. The algorithm initiates as a shooting method; the earlier prevalent pressure value acting as reference. A resultant array of pressures is produced and the algorithm works towards a predefined convergence criterion. The convergence criterion and a well distributed pressure array are crucial for efficient computation and quick convergence.

The pressure calculation algorithm takes into account individual specific volume values at different control volumes. The specific volume values are a result of the trial pressure value, enthalpy and the resultant dryness fraction. Through the calculation of densities and control
volumes, a net mass value is achieved and compared with the known value. The flowchart for pressure calculation denotes quantities with vector notation signifying the calculation being carried out for the entire array simultaneously.

The tolerance on the normalized change in mass per iteration is predefined in the computation, a decision pertaining to the model resolution and accuracy.

Figure 7 - Pressure Error Convergence
Figure 8 - Pressure Calculation Flowchart
2.7 Element Relocation

The transfer of elements across the length of the heat exchanger is an integral part of the computation. The transfer simulates the flow of the fluid subject to the conditions of pressure differential and the resultant fluid flow rate. Fluid flow is represented by a unidirectional irreversible transfer between adjacent control volumes. The migration of properties across control volumes occurs towards the end of each iteration. The control volume configuration (Table 1 - Finite Control Volume Configuration - 'Matrix Element') entails that all the properties must travel across the length of the exchanger. The transfer of the elements follows an algorithm (Table 2 - Element Relocation Algorithm). End points of the heat exchanger and the control volumes residing therein are subject to special treatment. The control volume at the entry receives fluid at the pump delivery properties that are generally quite distinctive from the overall composition of the heat exchanger. The control volume at the other end that exits the heat exchanger is indicative of the instantaneous mass flow rate of the system. Properties of the control volume thus leaving the heat exchanger are recorded in a different matrix and subject to treatment through the following turbine or pump.

<table>
<thead>
<tr>
<th>Heat Exchanger Intermediate</th>
<th>( X_{i+1,j} = X_{i,j} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Exchanger Entry</td>
<td>( X_{1,j} = f (P_{pump}, T_{pump}, H_{pump}, \dot{m}_{pump}) )</td>
</tr>
<tr>
<td>Heat Exchanger Exit</td>
<td>( X_{n,j} = \dot{m}_{system} )</td>
</tr>
</tbody>
</table>
Table 3 - Control Volumes Across Heat Exchanger Length

<table>
<thead>
<tr>
<th>Working Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
</tr>
<tr>
<td>----------------</td>
</tr>
<tr>
<td>Heat Source</td>
</tr>
</tbody>
</table>

Table 4 - Control Volume at Heat Exchanger Exit

<table>
<thead>
<tr>
<th>Working Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
</tr>
<tr>
<td>----------------</td>
</tr>
<tr>
<td>Heat Source</td>
</tr>
</tbody>
</table>
3 Implementation

3.1 Sectional Algorithm

The dynamic modeling of Rankine cycle has been carried out in the MATLAB environment. The method has been written separately for the conditions initiating a startup condition and a steady state condition. The program was written in a modular fashion. (Figure 9 - Computation Modules). The calculation of parameters as explained in the previous sections is performed during each time step. The code simulates components serially. The process freezes in time for the component properties to be calculated and its progress though the time interval to be tracked.
Figure 9 - Computation Modules
Figure 10 - Summary of Algorithm presenting the computation process and internal steps
3.2 Fluid Properties

Figure 11 - Pressure Enthalpy Diagram for Water (Webbook, 2008)

The working fluid for simulating the Rankine cycle has been assumed to be water. The properties of the fluid have been sourced from the NIST database. The database is an exhaustive source of saturation and superheated steam properties. The relevant properties for the computation were sourced using interpolation functions built in MATLAB.

The heat transfer coefficients of the working fluid vary in a large range considering the change of phase that the fluid undergoes. Two phase fluid mixture accounts for a high heat transfer coefficient and the largest amount of heat transfer occurs in this phase. Importance of the condenser design can be emphasized even more when the heat transfer coefficient variation is considered. The condenser is required to accommodate the fluid until it undergoes a complete
phase change from vapor to liquid. The coolant for the condensing fluid is assumed to be air and hence the overall heat transfer coefficient has to be calculated (10)

$$\frac{1}{h_{\text{overall}}} = \frac{1}{h_{\text{working fluid}}} + \frac{1}{h_{\text{coolant}}} \quad (10)$$

It is evident from (10) that the controlling resistance for heat transfer is on the air side. For a typical heat exchanger rejecting heat to the ambient, resistances to the heat transfer are offered by the working fluid, duct walls and the ambient air. With respect to the current study, walls thin to the limit offering negligible heat resistance have been presumed eliminating the resistance offered by the duct walls. Heat transfer coefficient of the working fluids flowing through the heat exchanger undergoing a phase change are in the order of \(10^3\) W/m\(^2\)-K. (Incropera) lists the range of heat transfer coefficients for liquids under such conditions between 2,500 to 100,000. Thus the first term on the right hand side of the equation (10) tends towards zero due to a large value in the denominator. The overall heat transfer coefficient value thus determined is based mainly on the coolant heat transfer coefficient. The overall heat transfer coefficient for the simulations conducted in the study was calculated as 90 W/m\(^2\)K.

A discretized geometry of a typical condenser is a flat tube with long flat fins. A geometry of this kind exposed to air (coolant) resembles an array of flat plates arranged in a neighboring fashion. Coolant heat transfer coefficient used through the value of 90 W/m\(^2\)K was obtained from correlation for cooling turbulent external flow over heated flat plate (Incropera).

$$Nu = 0.0296 \times Re^{4/5} \times Pr^{1/3} \quad (11)$$

It should be mentioned though that the heat transfer coefficient would be different if a different types of fin and tube assembly is used in the condenser.
3.3 Component Geometry

Table 5 - Condenser Geometry

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Circular Perimeter</td>
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<td>15”</td>
</tr>
<tr>
<td>Length</td>
<td>$L$</td>
<td>100”</td>
</tr>
<tr>
<td>Material</td>
<td></td>
<td>Aluminum 2024</td>
</tr>
</tbody>
</table>

Table 6 - Boiler Geometry

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Circular Perimeter</td>
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<td>15”</td>
</tr>
<tr>
<td>Length</td>
<td>$L$</td>
<td>100”</td>
</tr>
<tr>
<td>Material</td>
<td></td>
<td>Aluminum 2024</td>
</tr>
</tbody>
</table>
4 Model Validation

The idea of simulating a computational model based on arbitrary Lagrangian Eulerian method was derived from an earlier study conducted at University of Cincinnati (Miller, 2012). The model concentrated on simulating the vapor compression refrigeration system through a purely Lagrangian approach. Validation for (Miller, 2012) was carried out using experimental data from Air Force Research Labs Two Phase Thermal Management System. To validate the present ALE model, identical vapor compression refrigeration cycle using the same component dimensions and properties as Miller (2012) were carried out.

Validation of the computational model developed for the current study was accomplished by comparing simulation results with those provided in (Miller, 2012). The referencing of the required properties was changed to those of R134a. Similarly, the vapor compression system was simulated with the boiler component acting as the corresponding evaporator and condenser function changed to suit the cycle requirements. Positioning of the turbine and the pump were changed to an expansion valve and a compressor, respectively. Oil with referenced properties acted as the target cooling object in the cycle while water acted as the coolant for the refrigerant at the exit of the compressor. Validation was achieved by comparing results of the simulation listed below:

1. Startup Refrigerant High Side Pressure (Figure 13 – Startup - Refrigerant High Side Pressure)

2. Step Increase in Oil Inlet Temperature (Figure 14 - Step Increase in Oil Inlet Temperature)

3. Step Increase in Water Inlet Temperature (Figure 15 - Step Increase in Cooling Water Inlet Temperature)
Figure 12 - Generic Vapor Compression System Component Layout
Figure 13 – Startup - Refrigerant High Side Pressure
The observed temperature of the refrigerant in the vapor compression model (Miller, 2012) at the startup is 75 °F as opposed to 55 °F during the cycle steady state. The initial pressure was set to 80 psi. During initiation phase the startup cycle perceives a relatively warmer refrigerant attempting the oil cooling process. Initial cooling of the oil is achieved by the refrigerator with a base temperature of 75 °F compared to the temperature of 55 °F observed at steady state. The increase in temperature of the refrigerant is always reflected in the changing evaporator pressure. Thus the intermediate pressure peak observed can be attributed to the phenomenon described. As observed (Figure 13 – Startup - Refrigerant High Side Pressure) the refrigerant pressure inside the evaporator increases to achieve a peak around 280 psi, but thereby lowering down in process to stabilize at pressure of 240 psi. The behavior of the refrigerant can be accounted to the startup conditions of the system components.

The vapor compression model was subjected to two tests similar to Miller (2012). The steps were achieved through a step increase in the temperature of the oil flowing to the evaporator (Fig. 14). An increase in the oil temperature is actually an increase in the load on the evaporator side and the whole vapor compression system consequently. The vapor compression system simulation with steady state condition inputs was initiated and oil with a sudden temperature increase of 10 deg F was supplied. The temperature of the incoming oil increased from 110 deg F to 120 deg F. Dashed line indicates the temperature of incoming oil with the step evidently visible, while the solid line indicates the temperature of leaving oil. As expected an increase in the temperature of the leaving oil is seen. The temperature of the leaving oil is not sudden through a step and occurs through a gradual ramp. The initiation of the ramp is also seen to occur after the incoming temperature step instant. This is attributed to the time required for the oil to reach the duct outlet and its temperature to be recorded. The leaving temperature of the oil is then seen to stabilize. Temperature of outgoing oil increases through 10 deg F during the
course of the test. Thus the degree of cooling is observed to remain the same in view of other factors remaining constant.

Subsequently, the vapor compression cycle was subjected to a step change in the temperature of the cooling water supplied to the condenser (Fig. 15). The condenser rejects the heat in the compressed refrigerant and conditions it to expand in the expansion valve. A sudden change in the temperature of the cooling water inlet temperature would hamper the operation of the condenser and supply a ‘hotter’ refrigerant than usual. The cooling water inlet temperature is represented by dashed line. The step change in temperature is evident and portrays increase in inlet water temperature by 10 deg F. Solid line indicates the outgoing temperature of the cooling water. The cooling water temperature is seen to increase by 15 deg F. The step change reflects in the temperature of cooling water outlet a few seconds later. As explained previously, this phenomenon is attributed to the time taken by the cooling water to reach the outlet.

Comparison between results obtained from (Miller, 2012) and the model prepared in the current study is shown in figure. Results from (Miller, 2012) have been pointed out through cross marks on the solid lines. These comparisons show very good agreements and act as validation of the developed model.
Figure 14 - Step Increase in Oil Inlet Temperature

Step Increase in Oil Inlet Temperature

Oil Outlet Temperature

X - (Miller, 2012)
Figure 15 - Step Increase in Cooling Water Inlet Temperature

- Cooling Water Inlet Temperature
- Cooling Water Outlet Temperature

X - (Miller, 2012)
5 Results and Discussions

5.1 Temperature - Entropy Diagram

Figure 16 shows the T-S diagram for a Rankine cycle illustrating the four major components of the cycle, viz., the pump, boiler, turbine and the condenser. In a T-S diagram, the area under the T-S curve representing the boiler stands for the heat added to the boiler, while the area under the T-S curve for the condenser represents the heat rejected inside the condenser. The area bounded between the curves for boiler and condenser is the resultant difference between the added and rejected heat quantities and represents the power output for the turbine driven by the Rankine cycle.

The curve presenting the boiler progresses through sensible heating of liquid, latent heating of the liquid – vapor composition and the subsequent sensible heating of the vapor. The Rankine cycle shown in the figure is a superheat Rankine cycle. Pure water is considered as working fluid. The heat addition process in the boiler is followed by an isentropic expansion and provides power output for the cycle. The design point for the turbine operation is considered as the inlet and exit pressures of 2 MPa and 30 KPa, respectively generating a power output of 0.77 MW and steam delivery of 3536 kg/hr. The turbine expansion process is isentropic as evident by the straight line. The process demonstrates a change in temperature of the working fluid while keeping the entropy constant. The steam from the turbine exits enters the condenser and exits it as saturated water at turbine exit pressure 30KPa.
Figure 16 - Rankine Cycle T-S Diagram
5.2 Startup Stage

The boiler startup simulation has been carried out to understand the maximum delay in response to a change in heat input. The variation of dryness fraction, prevalent pressures and mass in the individual control volumes of the boiler are of particular interest. The heat transfer to the boiler causes fluid heating and subsequent phase change. The dryness fraction at the exit of the boiler is shown in Figure 17 and the corresponding change in boiler pressure is depicted in Fig. 18. Also, boiler exit temperature and mass flow rate are plotted in figures 18 and 19, respectively. Initially, the entire boiler volume is filled with liquid water and the pressure is low. Because change of phase decreases the overall density of the fluid, to fit the fluid in the boiler volume, its pressure increases. This trend is seen in Fig. 20.

The dry steam is first predicted around 15 minutes (Figure 17 - Boiler Exit Dryness Fraction – Startup). However, consistent delivery of dry steam and a uniform delivery rate at the required pressure is achieved at about 18 min. The boiler startup time thus subject to above conditions is about 18 minutes.

The steam delivery rate of the boiler a steady state is 3536 kg/hr. The steam delivery rate at pressure of 2 MPa is achieved gradually following a specific trend (Figure 20 - Boiler Delivery Rate – Startup). The mass flow rate out of the boiler at pressure of 2 MPa and dryness fraction of 1 is fixed as the boiler delivery rate. The boiler delivery rate stabilizes at the rate of 3536 kg/hr a little after the initial delivery of dry steam.
Figure 17 - Boiler Exit Dryness Fraction – Startup
Figure 18 - Boiler Pressure – Startup

Boiler Delivery Pressure
2000 KPa at 17.38 mins
Figure 19 - Boiler Outlet Temperature
Figure 20 - Boiler Delivery Rate – Startup

Boiler Delivery: 3536 Kg/hr

Dry steam at 14.67 mins
The mathematical model requires discretization of the boiler tube geometry. The discrete elements in the boiler tube contain masses that are functions of the prevalent pressure, enthalpy and the subsequent dryness fractions. The resulting amounts of mass present in the discrete volumes change with time and prevalent conditions. (Figure 21 - Boiler Mass Elements – Startup). A considerable difference is observed in the masses present along the length of the boiler tube geometry from entry to exit. The masses present in the exit control volume determine the mass flow rate of the boiler at the particular time instant. It is to be noted that the prevalent pressure inside the boiler tube is increasing throughout the startup process till it reaches the delivery pressure of 2MPa.

The composition of the boiler tube (Figure 22 - Boiler Dryness Fraction – Startup) can be represented in terms of the dryness fractions of each discretized control volumes. The composition of the boiler during the startup cycle at different time instants is consistent with the exit dryness fractions shown in the earlier figure. The curve representing the dryness fractions at 15 minutes into the startup shows the fluid to have achieved a dryness fraction of 1.
Figure 21 - Boiler Mass Elements – Startup
Figure 22 - Boiler Dryness Fraction – Startup
Figure 23 - Condenser Dryness Fraction – Startup
Figure 24 - Condenser Pressure Steady State
The composition of the condenser tube (Figure 23 - Condenser Dryness Fraction – Startup) can be represented in terms of the dryness fractions of each discretized control volumes as in the case of the boiler. The composition of the condenser during the startup cycle at different time instants is consistent with the boiler exit dryness fractions shown in the earlier figure. The curves represent increasing dryness fractions of entering fluid as the boiler progresses through the startup stage. The curve of the dryness fraction at 15th minute of the operation shows the entering dryness fraction to be 1. The properties of steam received by the condenser from the turbine at 15th minute into the cycle are consistent with the saturation properties of the steam at 30 KPa.

The steady state of the condenser is achieved at the stabilization of internal pressure. The influence of the ambient temperature functioning as a sink is considered to be the sole influencer of the condenser pressure. Depending on the ambient temperature, the condenser achieves a certain pressure level and a corresponding temperature variation (Figure 25 - Condenser Temperature Dynamic). (Figure 24 - Condenser Pressure Steady State) illustrates the trend of pressure increase in the condenser at different ambient temperatures. The horizontal straight line denotes the stabilized condenser.
Figure 25 - Condenser Temperature Dynamic
5.3 Cycle Response to Discrete Temperature Variation

The behavior and performance of the Rankine cycle power plant at different ambient temperatures is crucial in understanding its intrinsic characteristics. To understand how the condenser temperature and pressure responds to various ambient air temperature, simulations were carried out at ambient temperatures of 299 K, 308 K, 314 K, 320 K. The corresponding changes in the condenser temperature are shown in Fig. 26.
Figure 26 - Condenser Steady State - Discrete Temperature Variation
Pressure adjusts itself in the condenser until it reaches the steady state and its variation is directly influenced by the ambient temperature. Variation in the ambient temperature also leads to different stabilization time intervals. The stability time required and the corresponding ambient temperatures are listed in Table 7. While the overall trend in temperature change in the condenser is similar at different ambient air temperatures, the time required for condenser to reach steady state are different.

<table>
<thead>
<tr>
<th>Ambient Temperature (Kelvin)</th>
<th>Steady State Pressure (KPa)</th>
<th>Condenser Stabilization Time (Minutes)</th>
</tr>
</thead>
<tbody>
<tr>
<td>305</td>
<td>28.845</td>
<td>13.3</td>
</tr>
<tr>
<td>308</td>
<td>30.234</td>
<td>16.64</td>
</tr>
<tr>
<td>314</td>
<td>33.102</td>
<td>18.53</td>
</tr>
<tr>
<td>320</td>
<td>35.456</td>
<td>19.90</td>
</tr>
</tbody>
</table>
5.4 Cycle Response to Dynamic Temperature Fluctuation

The diurnal temperature variation affects heat transfer from the condenser and changes the condenser operating pressure and temperature. As such, once the system has reached a steady state operation, it is essential to inspect its response to the changing atmospheric temperature. This is especially relevant during hot summer months. Average temperatures for the month of July at four distinct locations in the United States have been considered (wunderground.com). These locations are listed below:

1. Northeast United States - Providence, RI
2. Southwest United States – Las Vegas, NV
3. Northwest United States - Seattle, WA
4. Southwest United States - Tampa, FL

In each location, two types of air temperature variations are considered. First, the condenser response is determined to the actual ambient temperature variation. Second, the air is considered to first flow through a pre-cooler such that its temperature does not exceed a set design point. Both temperature variations match when the temperature is below the pre-cooler set temperature. Variation of temperature through a time period of 24 hours is considered for the study. A sinusoidal temperature variation between the maximum and minimum daily temperatures is considered at each location.
Figure 27 - Temperature Fluctuation for Providence, RI
The ambient temperature variation in Providence, RI and corresponding condenser pressure variation is shown in Fig. 27. The ambient temperature varies between a low of 287 K and 301 K for a typical day in July. The condenser pressure is seen to change with the variation in temperature. Pressure in the condenser varies between 22 kPa and 28 kPa. A phase difference is observed between the condenser pressure and the ambient temperature in the plot above. Effect of condenser pressure is felt directly at the turbine exit and affects the power output and the operations dependent on that. The change in power output portrayed through graphs mentioned later helping in tracking the performance of the Rankine cycle.

Figure 28 shows a fluctuation in temperature for Providence, RI with a pre-cooler induced bound at 298 K. The temperature bound in terms of time lasts for about nine hours. A pre-cooler utilizing the thermal energy storage must be sized to ensure the supply of air at this preset temperature during the nine hours. As discussed above, there exists a phase difference in the condenser pressure and the ambient temperature. Because of a lower maximum ambient temperature, the pressure stabilizes a little above 28 kPa. Condenser pressure stays unchanged until the ambient temperature begins to decrease. The maximum pressure observed in this case at the stable temperature of 298 K is identical to the pressure observed when 298 K ambient temperature was applied to the condenser in a static manner. Similar behavior is observed at other locations considered here as well. The temperature variations and the corresponding condenser pressure variations are shown in Figs. 29 and 30 for Las Vegas, NV, Figs. 31 and 32 for Seattle, WA, and Figs. 33 and 34 for Tampa, FL. and Graphs below portray the aforementioned twin cases of temperature fluctuations for other locations in the United States. Figure 35 shows the variation of power output throughout the day and figure 36 provides the
corresponding cycle efficiencies for the four locations. The efficiency variation throughout the day when the maximum air temperature is capped is given in Fig. 37.

![Figure 28 - Temperature Fluctuation for Providence, RI (Bound - 298 K)](image-url)
Figure 29 - Temperature Fluctuation for Las Vegas, NV
Figure 30 - Temperature Fluctuation & Condenser Pressure for Las Vegas, NV

Location - Las Vegas, NV
Temperature in July

T = 308 K

Temperature (K)

Pressure (kPa)

Time

20:00 AM  6:00 AM  12:00 PM  6:00 PM  12:00 AM
Figure 31 - Temperature Fluctuation for Seattle, WA
Figure 32 - Temperature & Condenser Pressure Fluctuation for Seattle, WA

Location - Seattle, WA
Temperature in July
Condenser Pressure (kPa)

Temperature(K)

Pressure

$T = 304 \text{ K}$
Figure 33 - Temperature and Condenser Pressure Fluctuation for Tampa, FL
Figure 34 - Temperature and Condenser Pressure Fluctuation for Tampa, FL
Figure 35 - Power Output Fluctuation

Seattle, WA; Net Output = 18,000 kWh

Providence, RI; Net Output = 18,350 kWh

Tampa, FL; Net Output = 17,300 kWh

Las Vegas, NV; Net Output = 15,000 kWh
Figure 36 – Efficiency Fluctuation
Figure 37 - Efficiency Fluctuation (Bound)
Figure 38 - Sensitivity Test for Temperature Fluctuations with bounds at Las Vegas, NV
Sensitivity analysis was performed for the temperature variation for the location of Las Vegas, NV. The temperature variation was capped at three different temperatures of 308 K, 310 K and 312 K and their effects on the condenser pressure were studied. The lines in black color correspond to the temperature variations with bounds distinctly evident. The red lines towards the lower part of the graph correspond to the condenser pressures observed. The bounds last for different amounts of time for the three different temperatures. The bound lasts the longest for 308 K and is the shortest for 312 K. The bounds last for 10, 8 and 6 hours for 308 K, 310 K, 312 K respectively.

The pressures can be seen rising uniformly for the temperature variations till the bounds are reached. When the temperature bounds get initiated, the pressure stabilizes to an appropriate value. The stabilization values for the condenser pressure at three different temperatures are seen in the graph. As seen in the condenser plots preceding the current one, the pressure initially drops due to lowering of the ambient temperature and then proceeds to increase. Because of the long duration over which the maximum air temperature remains constant, the operating pressure of the condenser is the same as what was predicted in steady state simulations with a constant ambient temperature. The change in the power output corresponding to these temperatures should be balanced with the cost of pre-cooling in each case.
6 Conclusions

6.1 Conclusions

A computational model based on the Arbitrary-Lagrangian-Eulerian method was developed to simulate a Rankine cycle comprising of a pump, a boiler, a turbine, and a condenser. The method divides each time step in two parts. In the Lagrangian part, the mesh moves with the flow. In the Eulerian re-meshing of the domain, mass can enter or leave a component. Because the mesh returns to its original location after each time step, the cumbersome tracking of heat transfer interfaces between moving heating and cooling fluids in case of a purely Lagrangian method is avoided. The model was first modified to simulate a vapor compression cycle for which data were readily available for model validation. The model was then used to simulate the dynamic behavior of the Rankine cycle with an air-cooled condenser. The goal was to understand the effects of diurnal temperature variations on the performance of the air-cooled condenser and the Rankine cycle. Air temperatures during the month of July from four locations in the United States were considered. The simulations showed that the condenser operating temperature and the corresponding condenser operating pressure increases as the ambient temperature reaches a maximum early afternoon. This leads to a reduced power output from the turbine and lowering of the cycle efficiency. To overcome this shortcoming of an air-cooled condenser based Rankine cycle, a system with a thermal energy storage based air pre-cooler was considered. In such system, the air going to the air-cooled condenser is cooled so that the air temperature does not exceed a fixed maximum temperature. The simulations showed that such a cycle with air pre-cooling provides higher cycle efficiency. It allows the condenser to operate at a lower temperature and pressure and increases power output from the turbine. It is seen that the developed model can be used to predict the dynamic behavior of a Rankine cycle and determine
its performance in response to changing atmospheric conditions. Future work on organic Rankine cycles is recommended to evaluate their dynamic performance for utilizing low-grade waste heat. Also, some air-cooled Rankine cycles use a water cooled condenser with an air-water heat exchanger. The present model can be modified to simulate such a system as well.
7 Appendix

7.1 Variation of Condenser Initial Conditions

The study considered the initial starting condition of the condenser internal pressure to be 20 kPa. Stabilization of the condenser to different internal pressures were then simulated portraying different stabilization times. Stabilization times for the condenser assuming an initial condition different from 20 kPa have been addressed in the current section. At an unchanged ambient temperature of 308 K, the condenser stabilized to 30 kPa as in the previous cases. Stabilization times were observed to extend longer in the current cases.
Figure 39 - Condenser Stabilization Time for different initial conditions

Initial Pressure = 80 kPa; Stabilization Time = 58 Minutes

Initial Pressure = 60 kPa; Stabilization Time = 41 Minutes

Ambient Temperature = 308 K
Condenser Stabilization Pressure = 30 kPa
References


